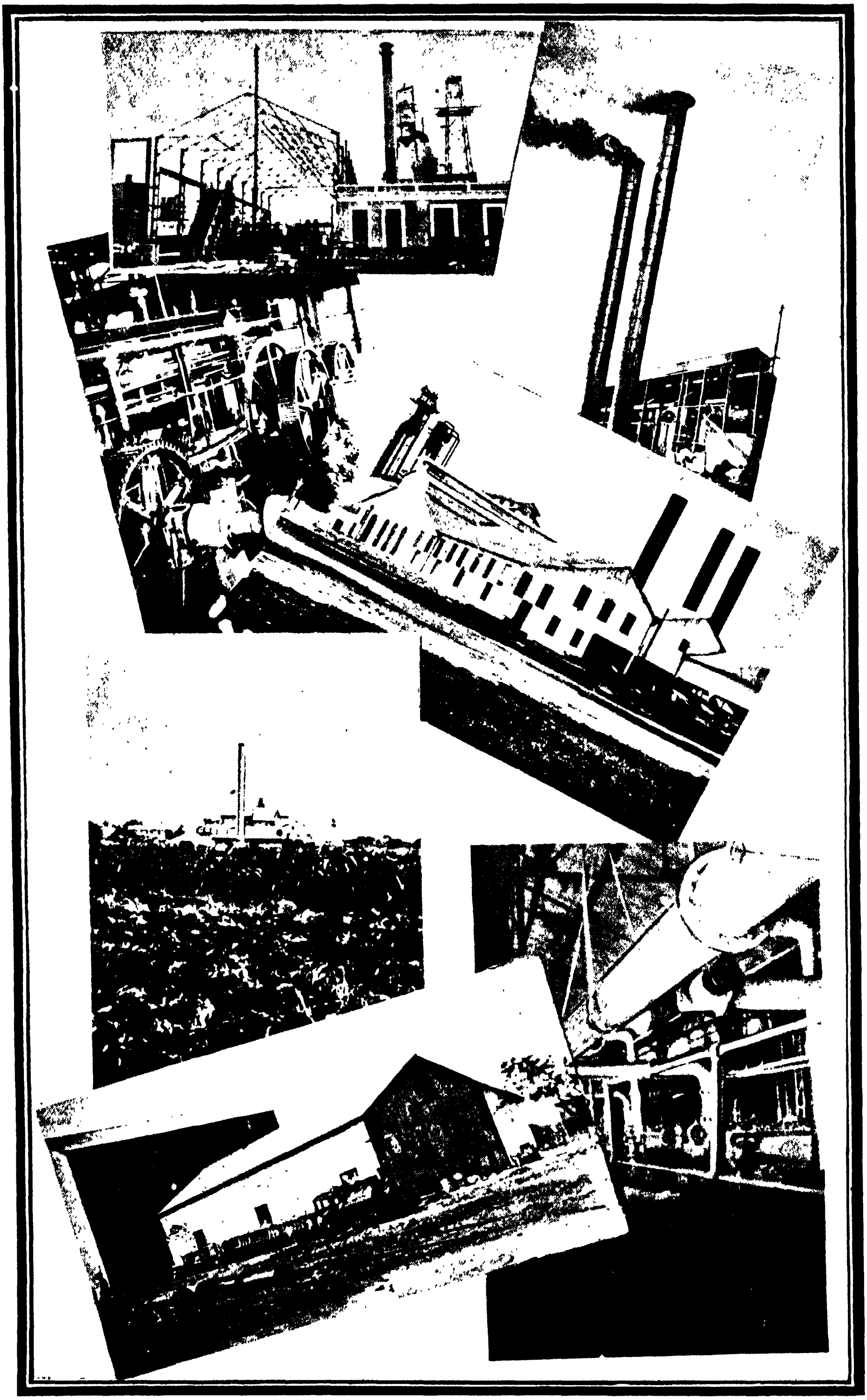

SNAPSHOPS OF THE ENGINEERS' FIELD OF ACTIVITY AT A CANE SUGAR FACTORY.

PLATE I.

TYPES OF CANE CARTS WITH PNEUMATIC TYRES.
(*Dunlop Rubber Co., Ltd.*)

MACHINERY AND EQUIPMENT

OF THE

CANE SUGAR FACTORY

A Textbook on Machinery
for the Cane Sugar Industry

BY

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7 & 8, IDOL LANE
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PREFACE

To meet the need for an adequate textbook, covering all the machinery and equipment used in a cane sugar factory, the present volume is offered the sugar world; hitherto the engineering side of sugar manufacture has not been the subject of nearly so many publications as have the agricultural and manufacturing sides of the industry, so an attempt has now been made to meet the deficiency.

A synopsis of up-to-date equipment, buildings, machinery and apparatus has therefore been collected from the author's private files, compiled during nearly 20 years of consulting, designing and operating practice abroad. Reference has been made to the theoretical as well as to the practical aspects of design and operation, and useful information is offered which the author trusts will prove of benefit not only to engineers operating factories or designing the machinery for them, but will also add to the overall knowledge required by the managing and manufacturing staffs of a modern sugar factory.

Many data contained in this book will be equally useful for the engineer and superintendent of a beet sugar factory or refinery.

The engineer designing cane sugar machinery does not invariably get the chance to put into practice and obtain operating results from the equipment devised by him; and useful information for this group may, it is hoped, be gleaned in these pages.

The 616 drawings to be found in the book have all been prepared by the author himself, and exclusive use has been made of these to explain the text; a drawing being considered the most effective means of conveying technical ideas.

To those persons and firms who have supplied interesting data, the author feels greatly indebted and will always appreciate receiving such data in future, or any comments or constructive criticisms.

He has also to thank those numerous firms who have helped to supply the varied collection of Plate Illustrations to be found in the book, which plates may be deemed to offer a useful contrast in perspective to the main illustrations.

PREFACE

A few brief sections of the book have already been published by the author in the *International Sugar Journal* and are here reproduced with the consent of the proprietors of that periodical.

Sources have been accredited so far as it has been in the author's power to do so, or they have come to his knowledge. But the data when filed originally were not meant for publication, so sources were omitted by the author in many instances ; and he must now rely on the courtesy of his readers if some omissions to credit come to light.

To Mr. NORMAN RODGER due acknowledgment must be given for his personal efforts in helping to prepare the Mss. for press. Here the author has been fortunate to secure a high standard of co-operation.

Finally, the book is dedicated to those able executives—managing, supervising and operating—abroad, with whom the author has shared so many a difficulty and has helped to solve them ; and in this group the designing engineers at home are also included.

L. A. TROMP.

Lansinkweg 49,
Hengelo (O), Holland.
September, 1936

NOTE TO THE SECOND IMPRESSION

The reception of my book by technologists, technical and managing factory staffs, as well as by designing engineers in practically every sugar-producing country of the world, has exceeded anticipation. In response to demands for it, a Second Impression is now issued.

At present the author finds it unnecessary to make any important alterations in the text; and it is hoped that the book will remain to prove a reliable guide on all matters concerning the machinery and equipment of the cane sugar factory, especially under the prevailing condition of scarcity of material.

L. A. TROMP.

Karthuizerstraat, 62,
Arnhem. Holland.
January, 1946.

CONTENTS

List of Plates

TROMP'S "MACHINERY AND EQUIPMENT."

ERRATA

Page 58, line 26. "Torsion" should be "torsional."

,, 83, 9th line from bottom. "42 lbs. per square inch" should be "per square foot."

,, 167, under Fig. 174. "37 ft. × 84 ft." should read "37 *in.* × 84 *in.*"

,, 168, under Fig. 175. "34 ft. × 72 ft." should read "34 *in.* × 72 *in.*"

,, 230, under Fig. 250. For "Diagram of Releasing Corliss Indicator Card" read "Indicator Card of Releasing Corliss Engine."

,, 231, under Fig. 252. "Lenz" should be "Lentz."

,, 249, line 14th from bottom. "on dry fibre weight" add "by 10% fibre in cane or by 12% 83, 5 to 292% on dry fibre."

,, 272, line 10th from bottom. After "analysis" add "of dry matter in the different fuels."

,, 342, line 14th from bottom. "1·61 cub.m. per kg." should be "1·61 kilogram per cubic metre."

,, 359, line 21st from top. Add "12-density of the juice."

,, 360, from line 10, recast:

"$\Delta_t = 232 - 82 = 150°F.$

$$\Delta_m = \frac{150 - 20}{\log_e \frac{150}{20}} = 64{\cdot}5°F.$$

"The arithmetic average $(150 - 20) \div 2 = 65°F.$, thus showing a small difference as compared with 64·5°F."

and at line 28:

". . . and with K = 120 B.Th.U./sq. ft./hr./1°F. and $\Delta_m = 64{\cdot}5$, the heating surface *A* amounts to 32 sq. ft.

"Thus for each ton of cane ground per hour, about 32 sq. ft. . . . ."

,, 369, 5th line from bottom. "0·007 per cent." should read "0·7 per cent."

[p.t.o.

Page 400, 14th line from top. "0·15 tons cake" should be "0·015 short tons cake."

,, 401, line 7th from bottom. Instead of "from 1 to 3 tons of cane" read "1 to 3 tons of cane per 24 hrs. per sq. ft."

,, 402, line 9th from bottom. In formula (104) read:

$$S = W_j \times B \div 100 \ldots\ldots (104).$$

and

$$100\,S = W_j \times B = W_{ji} \times B_i \ldots\ldots (104a).$$

,, 403, $W_e = W^r \times (1 - B \div B_i) \div 100 \ldots\ldots (105).$

,, 418, line 12th from top. Instead of "10%" read "6%" and then: $0{\cdot}88 \times \pi \times D^2 \div 4 = 0{\cdot}691\ D^2 \simeq 0.7\ D^2$.

,, 430, 4th line from top. For "chock graining" read "shock graining."

,, 434, line 15th from bottom. c.f. recast:

$(60 - 50) \div (85 - 50) \times 100 = 28{\cdot}5$ per cent. syrup
and 71·5 per cent.
second molasses.

,, 434, line 6th from bottom. "which require
$28{\cdot}3 \times (28{\cdot}5 \div 71{\cdot}5) = 11{\cdot}3$ parts
syrup, total .... 39·6 massecuite.
Total 130·3 syrup + 35·5 first
mol. + 28·3 second mol. .... 194·1 massecuite.
As final molasses, $39{\cdot}6 \times (50 \div 100) = 19{\cdot}8$ parts,
etc.'

,, 435, line 2nd from top.
First molasses .... 27·2 parts
Second molasses .... 21·7 parts

Total .... 148·9 parts massecuite.

Final molasses .... 15·2 parts

CHAPTER I.

THE FACTORY SITE.

WATER SUPPLY—RAILWAYS AND BRIDGES—
DWELLINGS—FACTORY YARD.

The determination where a cane sugar factory is to be located is purely an agricultural one, so far as it refers to the tract of land, which has to be adequate for growing sugar cane for a period of years and be situated where climatic conditions and rainfall are such that successful crops are to be expected.

The agricultural technologist, therefore, will make his investigations into the fertility and filtrability of the soil or the different kinds of soil, and by planting tests come to a final conclusion.

Once determined which tract of land to select, there are several engineering requirements necessary for the proper operation of the process of sugar manufacture.

The factory should be located in close proximity to the main railway line or to the coast, so as to have easy shipping and receiving facilities, and as centrally as possible within the area of canefields, as this will facilitate the transportation of cane with less rolling stock and greatly reduce maintenance, interest on capital and other fixed charges for this department. Moreover, an easy cane supply, so necessary for economical grinding, will thus be secured.

Many plots of a property are less fertile than others and the factory should be laid out on one of these less fertile areas. Often these plots are sandy or have a rock substratum, which gives a good sub-foundation for machinery and buildings.

For the living quarters of the employees and workmen, healthy surroundings are of paramount importance. Scrub jungle and medium size plants should be removed. Savannahs and ponds of standing or deteriorated water must not be allowed in the near vicinity of the factory.

The direction of the prevailing wind and its accessibility to the factory and dwellings should be ascertained in advance, to avoid dust, ashes, smoke and the odour of refuse going in the wrong direction.

The *Water Supply* is of very great importance. For condenser work exigencies are not many. River water containing floating trash and impurities can be used when coarsely strained. Even sea water is used in many instances, but it should be recollected that salt water must not mix with sugar or sugar solutions, as this would have a detrimental effect on the latter. For building purposes sea water is unfit, as it will hinder the binding of the cement.

A sufficient supply of fresh water, therefore, has to be at hand for general use, but factories close to the sea shore will often pump brackish water, and fresh water in some instances is brought from wells or springs miles away from the factory, this resulting in very costly pipelines.

Typical equipment belonging to these outlined essentials is described below under separate sub-headings.

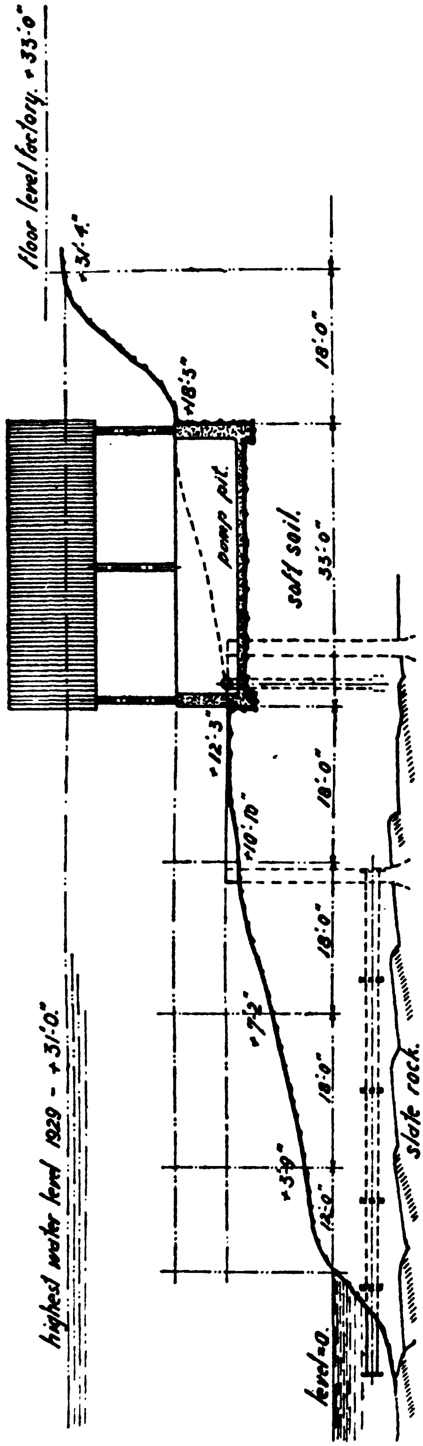


Fig. 1.—Pumping Station on River.

1.—River Pumping Stations.

Most rivers in tropical or sub-tropical countries have a flow of water which varies greatly between the wet and dry seasons, and the illustration of a *Pumping Station, Fig.* 1, shows a difference of 31 ft. above normal dry season level. The floor of the pump pit is well over 10 ft. above low level and about 300 ft. away from the factory buildings.

A triple expansion duplex pump, having steam cylinders of $11\frac{1}{2}$ in., 18 in., and 30 in. and pump cylinders of 26 in. diameter, with a common stroke of 24 in., supplies about 3000 Imp. gals. of water per minute and 100 ft. total manometric head.

Such a pump is of ample size for this output. On account of the distance from the factory, the exhaust steam is not returned, but a wet vacuum condensing equipment is installed on the same pump floor.

The live steam line is laid underground without any special insulating material. Heavy condensation, therefore, occurs and drainage of the pipeline is necessary before starting the pump. Through corrosion such a pipeline has been known to burst, causing inconvenience to the occupants of adjacent houses. The pump discharge is also a subterranean one. In case overhead equipment cannot be used, proper channels with concrete or masonry walls should be provided for subterranean pipelines.

The pump suction is from a subsider pit or sump next to the pump pit. At flood water the whole pump station is covered and after the water level has receded, both pits are filled with silt to a large extent, and expensive cleaning work has to be done ; also, one has generally to remove the silt from the horizontal pipeline between the river and the subsider pit, by means of a chain with scrapers.

A more efficient solution of the problem is a concrete vault, which can be closed hermetically at high river level. In it an electrically-driven centrifugal pump, which occupies a much smaller space, should be fitted. To instal the electric motor in the open pump pit has the inconvenience of its needing removal at times of high water. As the pump supplies water for general use during the dead season, it should remain in operating order all the time or else a small auxiliary pump be installed.

To raise the walls of the open pump pit is not feasible on account of the big difference between zero and highest level. The maximum rise of the river may last for a day only, but smaller rises, covering the existing pump pit, will occur more frequently.

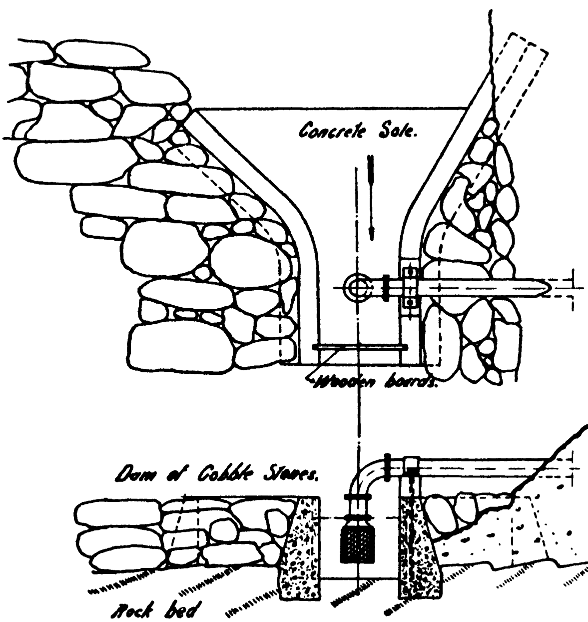


Fig. 2.—Pump Intake.

During the dry season, the river does not invariably carry sufficient water to cover the pump intake. Sand bags are then placed across the stream to form a dam and have to be removed when the water rises again, so as not to be swept away. For this case the following construction is provided.

A concrete lined narrow channel, as shown in *Fig.* 2, is built, having its bottom on the slate rock of the river bed. The walls of the channel are about 5 ft. high and a dam of heavy cobble stones is built across the river, having some fasteners sunk into the slate rock. During high water the whole dam will cover up with silt and the channel is blocked. On the water receding, the force of the current is sufficient to clean the channel.

At very low river levels, a few wooden boards are inserted in appropriate grooves in the channel walls, so to raise the water level until the foot valve is well covered.

The foot valve and strainer have to be secured by tie-rods, so as not to be dragged along with the current, which sometimes has a speed of 20 miles per hour.

2.—Artesian Wells.

When a water supply from a river is not at hand, or the water is not sufficiently clean to be used for the sugar manufacturing process and for household service, a solution must be found by drilling an artesian well.

Upon *Rain Water* one cannot depend, as the crop time is during the dry season, when the biggest amount of water is required. Cisterns or tanks are in use at small factories, but the water from the roofs as a rule is soiled by contagious faecal mattor from birds such as vultures and similar unclean creatures. This infected water should not be used at all, as it might propagate contagious diseases like typhoid fever. Boiling will make this water sterile again, but is very costly.

Generally, a trial well is drilled, which will give information about the sub-strata and the presence of water, and at what depth. In case there are open wells existing in the vicinity, useful information can be derived therefrom. In difficult cases a hydrologist can determine by galvanostatic measurements the presence of water veins. A more empirical method, using the so-called "diviner's wand," has given good results when in tho hands of skilled persons. Local well drillers have an overall knowledge of the subsoil and generally will find water at reasonable depths.

The diameter of the artesian well must be determined by the abundance of water issuing from the trial drilling, and the amount needed.

If the factory is located at a lower level than the surroundings, as might be the case at the side of a hill, it sometimes happens that the well water comes to the surface by hydrostatic pressure ; but unfortunately in most cases it has to be pumped, and it should be recollected that low pressure centrifugal pumps are made for about 60 feet max. manometric head.

The different types of equipment for bringing up the water from the depths are treated below.

Incidentally, attention may be called to the fact that well water generally is of a certain hardness, due to the presence of lime or magnesia, and special water-softening apparatus may be necessary.

3.—Ejectors.

The operating agents for ejectors are compressed air, and water under pressure. The disadvantage of both types is a low efficiency, decreasing with greater depth. There is, nevertheless, one advantage which should not be overlooked, namely, that there are no moving parts within the well casing, which will make the apparatus very dependable.

Of the first type, operated by compressed air, should be mentioned the Mammoth pump. To install this pump a considerable depth is needed, as the apparatus has to be submerged to about 100 to 150 per cent. of the manometric head. The air consumption for depths down to 200 ft. is about 4 to 5 gallons air of atmospheric pressure per gallon of water brought to the surface. The mechanical efficiency is about 45 per cent.

The efficiency of ejectors using water under pressure is still lower than that of the Mammoth pump. The advantages are that a compressor is not needed, but a part of the discharge of the pump is returned to the ejector or aspirator at the bottom of the well. The well only has its normal depth, and the limit of depth has already passed the 200-foot mark. These aspirators are built for quantities up to 100 Imp. gals. per minute.

4.—Deep Well Piston Pumps.

For artesian wells there are on the market *Piston Pumps* driven from the surface. Some are steam driven and others electrically by means of a belt-drive and reciprocal mechanism.

The maximum depth for pumping with this kind of equipment has reached 500 ft. and for capacities up to 30 gals. per minute. Such a pump is of a very simple construction with cup leathers on the piston. This piston contains also the discharge valve and can be brought to the surface without dismantling the casing. In the event of a leaky suction valve, the pump casing has to be hoisted to the surface. With clean well water, without sand or grit, these piston pumps have given good service.

5.—Centrifugal Deep Well Pumps.

The *Centrifugal Pump* is also used for artesian wells having an inside diameter of 18 in. or larger. The pump (*Fig.* 3) has one or more impellers according to the manometric pump head. These impellers are usually of the Francis type, the discharge being inclined upwards to shorten the path to the next impeller on top and to reduce the outside diameter.

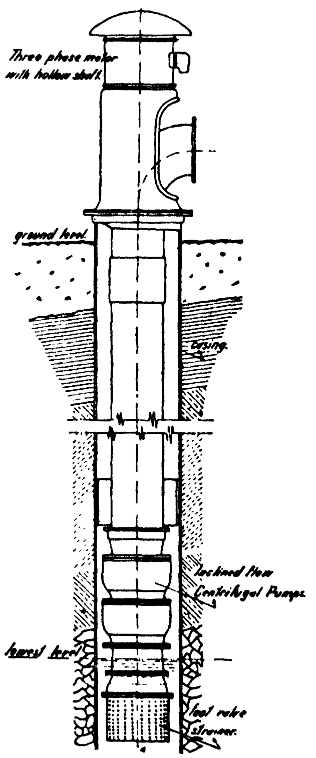


Fig. 3.—Deep Well Pump for Artesian Wells

The driving electric motor is mounted on a pedestal at ground level and direct-coupled to the pump shaft. The shaft is guided through the discharge pipe and surrounded by a casing of small diameter. The space between the shaft and casing is filled with grease or heavy oil, giving a proper lubrication to the shaft bearings, which are placed at intervals depending on the length of the pipe sections. The writer has seen these pumps in good service for capacities up to 5000 gals. per minute and 90 ft. head. The bigger capacities are used for irrigation.

When the well casing is of sufficient diameter, these pumps can be constructed with normal Francis impellers.

6.—Screw Pumps for Deep Wells.

For low lifts the water screw or vice has developed into the screw or helicoidal pump. The impeller acts like a ship's propeller, having ample screw surface. To prevent the whirling effect of the impelled water, baffles are placed to rectify the water current.

On account of their very small outside diameter, these pumps are suitable for deep well pumping, and they can be introduced in casings of 6 in. and upwards.

The manometric head of each impeller is low, and generally more impellers are mounted on the pumpshaft. This shaft is designed in the same manner as for the deep well centrifugal pump.

The efficiency is somewhat lower than that obtained with centrifugal pumps. For all high-speed pumps sandy water is detrimental and will cause the pump to wear out in a comparatively short time.

7.—Pumps for Open Wells.

Where water is to be found at a depth of not over 60 ft., the digging of an open well should be considered, as the artesian well is specially designed

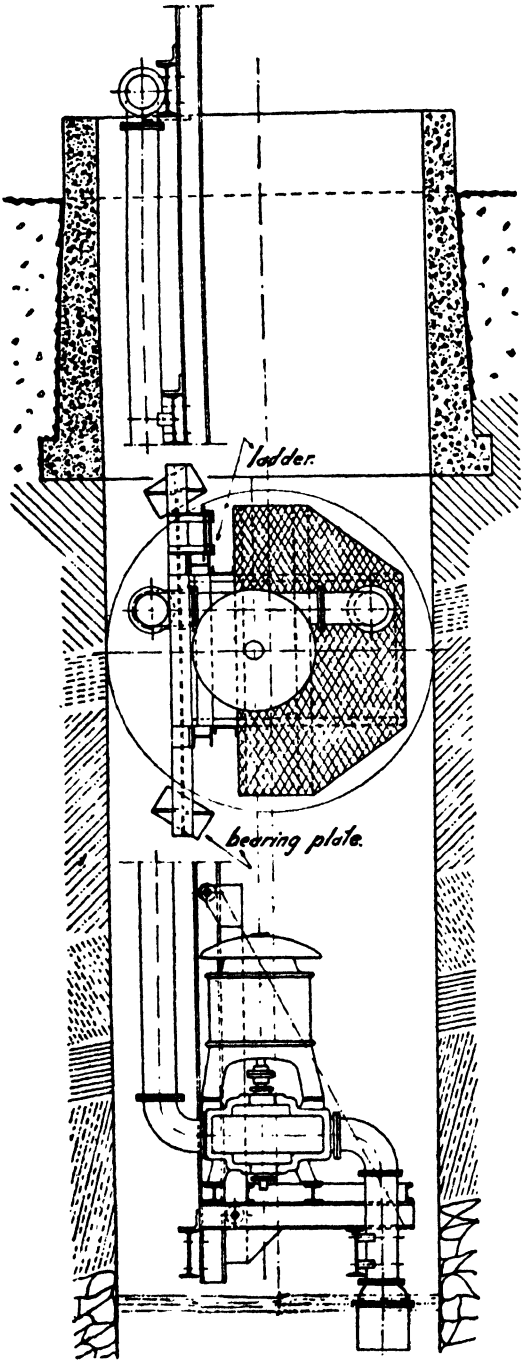


Fig. 4.—Pump for Open Well.

for greater depths. In *Fig.* 4 a concrete socket is provided at the mouth, being sunk until hard soil is reached, where there is no danger of earth caving in.

In an open well, nearly every kind of pumping equipment can be placed, so far as the diameter of the well will allow. The most important ones are mentioned below.

The *Pulsometer* is a two-chamber apparatus in which the water is impulsed by steam, which thereafter is condensed to cause a vacuum for suction. The two chambers work by turns and the automatic operation is achieved by the use of spherical valves. For sandy water the pulsometer has no equal and renders a very dependable service. A suction lift of 26. ft can be reached when the well water is cold and 160 ft. manometric head can be considered as the limit. For higher lifts pulsometers can be placed in series.

The steam consumption is high, being about 1 per cent. of the weight of the water delivered. Per lb. of steam about 15,000 ft. lbs. can be obtained and in some instances more.

The *Steam Injector* has a high steam consumption and is more subject to wear or clogging when dirty water has to be dealt with. For pumping purposes other than boiler-feeding this apparatus is little used. The temperature of the water will rise some 10°F., as is the case with the pulsometer.

A *Steam-driven Duplex Pump* has been used in many instances. Steam consumption is not very favourable, but its record of reliable service is such that this inconvenience is ignored. The live steam and exhaust steam lines down the well are a source of condensation, and the condensate has to be collected in separators, as otherwise water-hammer will occur.

The writer knows a case where a *Turbine Driven Horizontal Centrifugal Pump* is installed about 60 ft. below the surface. During the dead season this outfit is hoisted above the mouth of the well for repair and overhaul. Wells are generally very damp, and corrosion will take place very rapidly, when protective coatings are not applied.

Electrically Driven Horizontal Centrifugal Pumps have been taken down 120 ft. deep in open wells for capacities of 6000 U.S. gals. for irrigation purposes.

The pump shown in *Fig.* 4 is of the *Vertical Electrically Driven* type, being a convenient design for wells of smaller diameter with a considerable water supply.

This pump is mounted on a frame or platform, which can slide up or down on a pair of rails. The equipment therefore can be used as a sinking pump, when digging the well, and in case of repair can be easily hoisted to the surface by only detaching the suction and discharge pipelines.

Also for those wells where there exists a big difference of level between the dry and the wet season, the pump can be placed at different heights.

Care should be taken to have ample bearing plates under the supporting joists of the rails. It is an unpleasant experience when such a joist gives way.

Again, the precaution should not be overlooked to ascertain whether a well is free of *Carbon Dioxide*, ere descending into it. This noxious gas has caused many a casualty. It is released by the water or substrata and, being heavier than the air, accumulates at the bottom of the well. The presence of the gas can be found out by lowering an open lamp or lantern attached to a rope.

For protecting the pumping equipment and for mounting the hoisting gear, the wells should have a roof.

It is hardly necessary to add that water 24 ft. below the pump centre can be aspirated and does not require the pump to be placed down the well.

8.—Water Towers.

There are automatic systems of water supply, called *Hydrophores* on the European Continent, composed of a closed tank, placed on the floor level and having an air space on top of the water. The air is under pressure and replaces the hydrostatic pressure of a high level tank. This equipment can be used to advantage where the water consumption does not show wide fluctuations and where electrical current is supplied 24 hours per day. For plantation houses having electrical current but not connected to the main water line from the factory, the hydrophores are very useful.

In other cases, and especially for the needs of the factory and the surrounding dwellings, a Water Tower is to be preferred, having sufficient capacity not only to cover the water consumption at night-time when the pumping plant is at rest, but also for fire protection. For fighting a fire from the floor level a pressure of from 60 to 75 lbs. per sq. inch at the nozzle is generally needed. This will be almost impossible to get from a high level tank, and to overcome this difficulty, the main fire lines should be laid close under the ridges of the roofs with offshoots to the different floors. A head of 20 ft. above the highest ridge and an abundant supply of water will be a good solution.

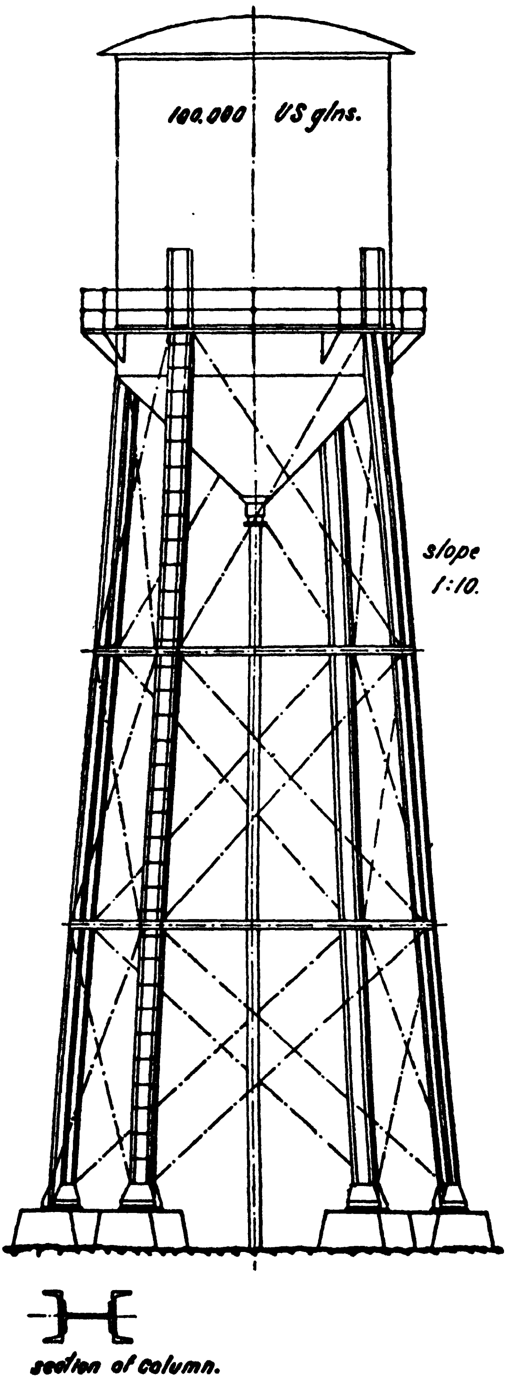


Fig. 5.—Water Tower.

If a hill is located close to the factory grounds a concrete or steel tank should be erected on top of it. Otherwise, a *Steel Tower* is advisable. *Fig.* 5 shows such a tower having on top a tank of 100,000 U.S. gals. capacity suitable for a large sugar factory. The usual roof has a conical shape without reinforcements, and as these roofs are easily blown off in countries affected by cyclones or hurricanes, a spherical roof as shown in the illustration offers a smaller surface to the attacks of the wind and can be adapted to have a framework of rolled sections, that can be fastened to the top angle of the tank, which should be of heavy material.

The supporting beams bear eccentric loads and are generally made of two heavy channels with lattice work. Nowadays, when electrical welding is universally employed, a welded construction of three channels, as shown, can be made at a reduced cost. Moreover, there will be a saving of paint, and painting can be done more thoroughly with less effort.

The tank bottom of conical or spherical shape should have an expansion joint at the lowest point for the discharge pipe.

A ladder of bars welded to one of the columns gives access to the walking platform, which is useful as a look-out to locate cane fires, etc.

A second ladder should go to the top, and an inside ladder in the tank should be provided. The best protection against cyclone damage is to fill the tank completely. A half-filled tank might be smashed just above the water line.

Where the sugar factory is built according to the gravity system, a rectangular or circular "*High Level*" *Tank* can be placed on a platform built on columns projecting through the roof, as shown in *Fig.* 6. Soil bearing and strength of columns and bracings should be checked.

The writer has known a case where a huge tank was built up from flanged cast iron plates, bolted together. The plates were 3 ft. × 3 ft., and the tank had been gradually increased by adding additional plates, but the maker had neglected to provide sufficient tie-rods and reinforcements. One day, when the tank was filled to the brim, one of the sides collapsed and about 60 tons of water poured down on the factory yard.

9.—Quantities of Factory Water.

When the water supply is limited, the factory should have spray cooling equipment for the waste-water of the condensers, so that it can enter the cycle again. During the grinding season the condensed vapours which have entered the condensers increase the amount of water. Water for the make-up is therefore scarcely needed.

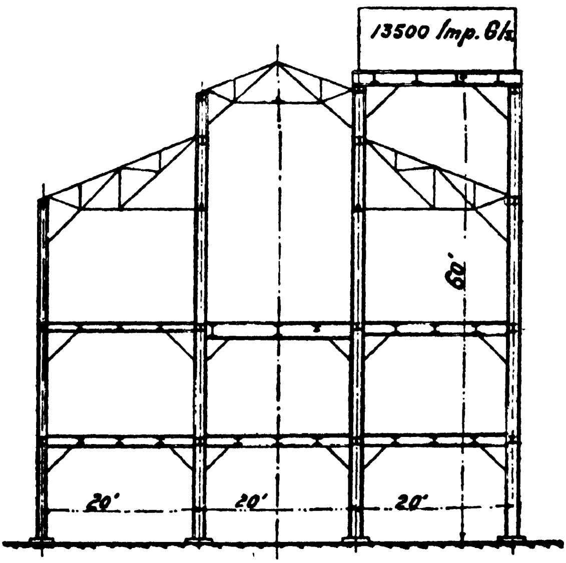


Fig. 6.—High Level Tank on Roof.

Boiler feed-water will be supplied by the condensate from the coils and calandrias of the evaporators, vacuum pans and heaters.

For cooling water in rapid cooling crystallizers and for washing white or centrifugalled sugars, about 250 per cent. on the weight of produced sugars is needed in all.

For sanitary and household requirements, including waste, 150 gals. per head of the factory population per 24 hours, will cover amply all that is needed. Where factories have a possibility of increasing their output in the future, this should be considered in advance.

The capacity of water towers varies between 10 and 20 Imp. gals. per ton of cane ground per 24 hours.

10.—Plantation Railways.

Only the very large sugar factories have a special department of civil engineering, and the mechanical engineer is in such cases called on to assist in making constructions for bridges and other railway appliances. For the medium size and small factory the railway department is frequently supervised by the factory mechanical engineer. A few typical problems of railway engineering, therefore, should not be omitted here.

The question whether *narrow* or *standard gauge* lines should be used is a matter which should be carefully considered. Any factory receiving cane, or shipping sugars, over a public railway, should use the same gauge as this public carrier.

If the canefields give a high yield in cane and are located close to the factory, so that many trains can be hauled per day and the capacity of the factory is not too large, a narrow gauge layout should be chosen. Furthermore, it is of paramount importance whether the railway is laid out in a flat country or there are gradients which will greatly reduce the hauling capacity of a given locomotive.

A larger factory, having considerable distances to cover for its cane supply, should use *standard gauge* exclusively.

A comparative cost calculation of transportation charges, inclusive of maintenance charges of equipment, should be carefully estimated before any decision is taken.

For general information the capacities of narrow and standard gauge cars should be taken at 10 and 25 tons of cane respectively, although there are cane cars for 36 in. gauge having 20 tons capacity and 40-ton cars for standard gauge.

One locomotive being deemed to pull one cane train, the amount $A_{loc.}$ is derived from :—

$$A_{loc.} = \frac{G}{R \times L \times C} \quad \ldots\ldots\ldots\ldots \quad (1)$$

where : $G =$ Grinding capacity per 24 hours in tons of cane.
$R =$ Number of round trips in each direction per working day of the train crew.
$L =$ Normal average loading capacity of each car in tons.
$C =$ Number of cars per train.

When there are railway branches of different lengths, then G should be taken as the amount of cane hauled per working day from each branch.

It is obvious that, for economical operation, the following balance should be approached :—

$$R \times T = W \quad \ldots\ldots\ldots\ldots \quad (2)$$

where : $T =$ Time required to complete a full round trip, including collection and delivery of loaded cars and distribution of empty ones.
$W =$ Working hours of railway crew per day.

This being the case, the formula (1) can also be read :—

$$A_{loc.} = \frac{G \times T}{W \times L \times C} \quad \ldots\ldots\ldots\ldots \quad (3)$$

Applying formula (1) to a few cases from practice :—

Case I : G — 960 tons. L — 10 tons per car.
R — 2 round trips. C — 20 cars per train.
T — 3 hours. W — 8 hours per day.

$$A_{loc.} = 960 \div (2 \times 10 \times 20) = 3 \text{ locomotives.}$$

This factory has a 36 in. narrow gauge and 120 cane cars. Three locomotives are in use for cane hauling and also do shunting in the yard. The average cane haul is about 8 miles (13 km.).

Case II : G — 5400 tons. L — 30 tons per car.
R — 2 round trips. C — 20 cars per train.
T — 5 hours. W — 12 hours per day.

$$A_{loc.} = 5400 \div (2 \times 30 \times 20) = 5 \text{ locomotives.}$$

The factory has standard gauge (4 ft. $8\frac{1}{2}$ in.) and 480 cars. Seven heavy locomotives are in use, of which one is for shunting. Sugar hauling is also done with this same equipment. Average haul about 25 miles (40 km.).

Case III : G_s — 1800 tons by standard gauge. L_s — 30 tons.
G_n — 8200 tons by narrow gauge. L_n — 20 tons.
R_s — 1 round trip. R_n — 2 round trips.
T_s — 6 hours. T_n — 5 hours.
C_s — 20 cars. C_n — 10 cars. W — 12 hours.

$$A_{loc.s} = 1800 \div (1 \times 30 \times 20) = 3.$$

Four heavy standard gauge locomotives are in use and 80 cars. Shunting is done by cable haulage.

$$A_{loc.n} = 8200 \div (2 \times 10 \times 20) = 21.$$

This factory has also 27 narrow gauge locomotives and has grown too large for narrow gauge cane supply. Six hundred cane cars of 20 tons are available. With 450 standard gauge cane cars and 12 locomotives of the same gauge, the calculation would well cover the exigencies, where now 31 locomotives and 680 cars are in service.

11.—Plantation Railway Bridges.

On soft soil the bridge heads of many a plantation railway have collapsed. There are generally several reasons, but one of the principal ones is the eccentric load, whereby the bridge head tends to fall out of plumb, towards the river or creek.

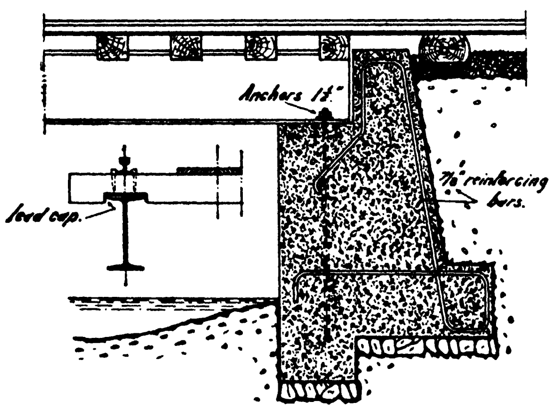


Fig. 7.—Plantation Railway Bridge Head.

Even the customary slope will not prevent this in all cases.

The construction in *Fig.* 7 is very reliable. The projection below the support of the load will prevent lateral sliding and the heel protruding under the abutment will prevent overturning. Reinforcements one foot apart are provided for tensile strength.

The bridge beams are firmly tightened to the bridge head. The mid-stream pilaster has oblong holes for expansion. The bearing on the bridge head is done by means of three pieces of heavy rail, about 8 ft. long and inserted in the concrete.

The cross ties of hard wood are trimmed to clamp the bridge beams. As corrosion will take place especially underneath the ties, where moisture is retained by the wood fibres, a lead cap is laid on top of the bridge beams, which gives good protection. 150 to 200 lbs. per sq. inch is a safe bearing load for lead. For heavier specific bearing a copper plate of $\frac{1}{16}$ in. to $\frac{1}{8}$ in. thickness should be preferred.

Between the rails on the cross ties are laid a double line of boards, for pedestrians crossing the bridge. For high bridges this is very convenient.

Bearing Capacity of the soil should be determined by testing with 2·5 times the specific load when under traffic. Soft soil will have a bearing value of about one ton per square foot allowable, whereas medium rock strata will bear up to 20 tons per square foot.

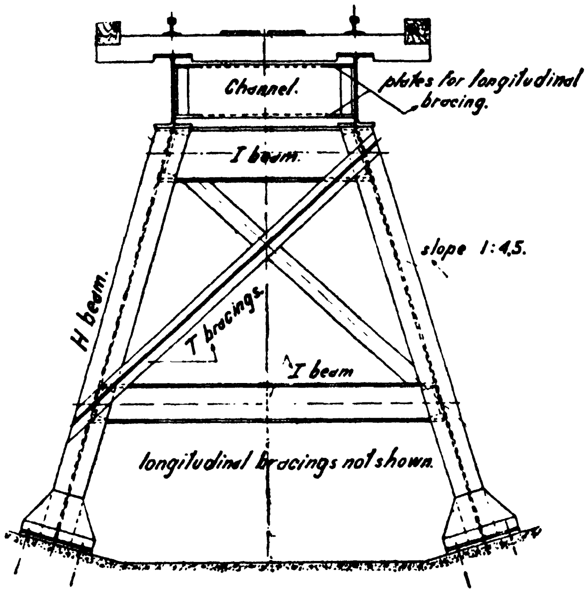


Fig. 8.—Midstream Bridge Support for Lateral Forces.

Pile driving should be done only in very exceptional cases.

The safe bearing load P of a pile in lbs. is determined by :—

$$P = \frac{W \times H}{Y \times S} \quad ..(4)$$

where : W = Hammer weight in lbs.

H = Fall of hammer in inches.

Y = Factor of safety = 10.

S = Sinking in inches under last blow.

The mid-stream bridge support shown in *Fig.* 8 is placed in a river nearly dry in summer time and having a hard rock bottom, the bridge being about 40ft. high and 300 ft. long. This particular construction was applied, as the river has a rise of about 30 ft. and many tree trunks are dragged along in the stream. The supporting columns have a slope of 1 in 4·5. The adjacent columns are braced together in pairs so as to form a tower, having two parallel and two inclined faces.

This bridge has withstood heavy currents accompanied by tree ramming. In the drawing the height has been reduced.

The cross ties are held in place by lateral wooden beams, trimmed at the cross tie spacing of 24 inches. There is no bolting of the cross ties to the bridge beams. Traffic is composed of 70-ton locomotives and 30-ton cars.

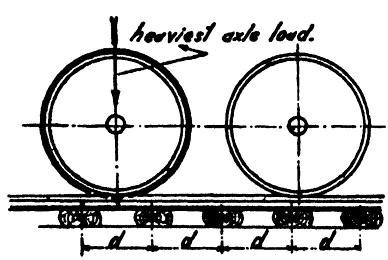


Fig. 10.—Rail Deflection.

The question is sometimes discussed whether concrete or constructional iron should be used for bridge supports. Careful cost estimates will determine which construction has to be followed. For tall supports constructional steel has the advantage not only in respect to reduced cost but also to reduced weight, especially when the river bed has but medium bearing qualities.

The writer knows a case where a concrete bridge support about 60 ft. high fell out of plumb apparently by reason of its own weight and had to be demolished completely and then be re-moulded and concreted on fresh

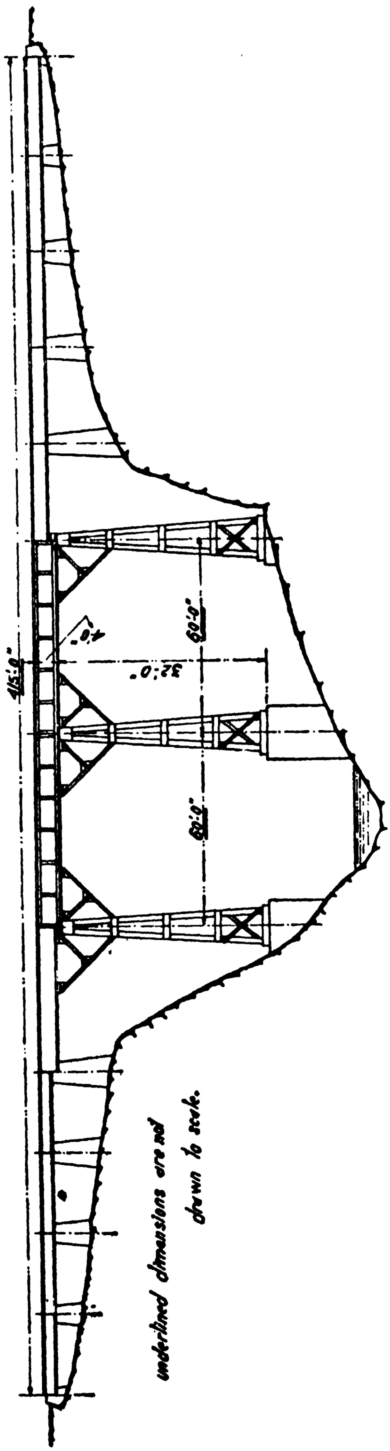


Fig. 9.—Narrow Gauge Plantation Railway Bridge.

foundations laid deeper in the river bed.

A typical *River Profile* is shown in *Fig.* 9, the centre part forming a deep ravine and the extremities being on a much higher level. The centre supports, therefore, were made of constructional steel, 60 feet apart. The bridge beams of the centre part have a made-up section of plates and angles. The remaining sections are of rolled shapes at centre distances of 30 to 45 feet.

After the bridge was in place, a few new locomotives were bought, having each a total weight of 65 tons, compared with the 33 tons of the existing ones. Reinforcements were applied to the centre spans, while the outer spans were reduced to half the length by placing concrete supports between the existing ones.

12.—Rail Deflection.

According to BALDWIN'S formula, the rail weight per yard with a given *wheel load* is determined by :—

$$W_r = \frac{L_w}{300} \quad \ldots\ldots (5)$$

where : W_r = Weight of rail in lbs. per yard.
L_w = Max. wheel load in lbs.

It should be recollected that the wheel load is half of the axle load.

The Baldwin formula has value for a cross tie or sleeper spacing of 24 in.

The deflection or bending of the rail only exists as a matter to be considered when certain specific stresses are reached. A heavier rail section than that absolutely required has no detrimental effect in respect to cost of purchase and laying down, but might reduce maintenance and repairs to a large extent, especially if heavier rolling stock were to be used in the future.

The heaviest axle load of the rolling stock to move on the rail in question —viz., the load on one pair of wheels—should satisfy the following formula :—

$$\frac{L_a \times D}{4} = 2S_r \times K \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (6)$$

being : L_a = Max axle load in lbs. (2 $\times$ wheel load).
D = Spacing, centre to centre of cross-ties in inches.
S_r = Section Modulus of rail section i· cubic inches.
K = Allowable stresses in lbs. per sq. inch.

Although rail material will stand allowable stresses up to 18,000 lbs. per sq. inch (1260 kg./cm.²) it is not advisable to exceed 9000 lbs. per sq. inch, as the life of the rail will be shortened by resorting to maximum allowable stresses.

13.—Tie Bearing.

The spacing between ties should be 24 in. centre to centre for plantation work.

Cross-ties or sleepers in Europe and U.S.A. are from soft woods, generally creosoted. Northern oak is considered as soft wood in this case. Most cane plantations have their own lumber supply or one very close at hand, and the excellent tropical hard woods should be exclusively used.

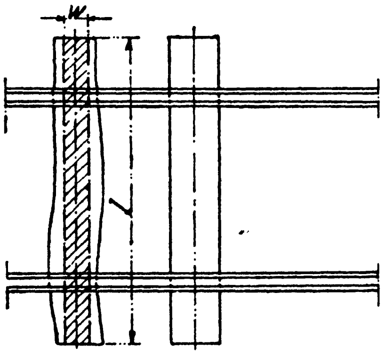


Fig. 11.—Tie or Sleeper Bearing.

The writer knows cases where imported creosoted cross-ties from northern countries had completely rotted away within three years.

Tropical hardwoods will have a life of eight to ten years or longer and do not need to be creosoted, as they are not attacked by insect pests such as white ants and the like.

The tie length should be at least 8 feet for standard gauge.

Sawn ties are not essential for good rail support. Straight poles, flattened on two sides, 6 to 8 in. high and having 8 in. bearing width, will give excellent service.

Steel tie plates or chairs are not used on plantation railways, as the hard wood has very good bearing, and the rails are held in position by dogspikes.

For expansion, $\frac{3}{16}$ in. between rails of 33 ft. length should be allowed.

For a known axle load, according to *Fig.* 11 :—

$$L_a = W \times L \times B_t \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (7)$$

where : L_a = Max. axle load in lbs. (2 $\times$ wheel load).
W = Mean width of cross-tie in inches.
L = Length of same in inches.
B_t = Bearing coefficient in lbs. per sq. inch.

For standard gauge, 50 to 90 lbs. per sq. inch is allowed with proper ballasting.

Plantation railways are generally laid in an earthen road bed and 30 lbs. to the sq. inch (2 kg./cm.²) bearing coefficient can be taken as a safe figure.

For a cross-tie of 8 in. × 96 in. bearing surface the allowable axle load for an unballasted road would therefore amount to about 23,000 lbs. For heavier axle loads, and also for soft soils, ballasting will be necessary.

Sundries, like *Rail Frogs*, *Guard Rails*, *Switches*, *Crossings* and *Railbraces* can be bought direct from the manufacturers, and good reliable constructions should be selected.

14.—Dwelling Houses.

As the writer has been called on many a time to build or design dwellings for sugar factories, a few general constructional details may be given. Only general indications, as mentioned at the beginning of this chapter, will be considered.

Sanitary Sewers should be enclosed throughout and septic tanks of sufficient capacity be provided. These septic tanks should be well sealed to avoid the entrance of air and so allow the anaerobic bacteria to destroy the organic matter contained in the sewage, thus reducing the sludge.

Per million gallons of sewage, about 2 cubic yards of sludge are obtained.

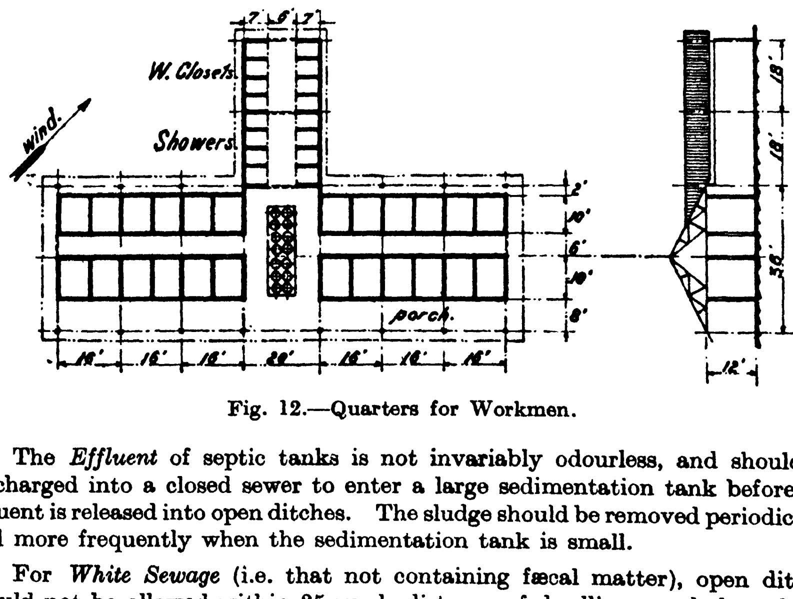


Fig. 12.—Quarters for Workmen.

The *Effluent* of septic tanks is not invariably odourless, and should be discharged into a closed sewer to enter a large sedimentation tank before the effluent is released into open ditches. The sludge should be removed periodically, and more frequently when the sedimentation tank is small.

For *White Sewage* (i.e. that not containing fæcal matter), open ditches should not be allowed within 25 yards distance of dwellings, and they should have sufficient fall to maintain a steady flow. If spill water from a weir or other source can be led into an open sewer it will greatly increase the amenities of the place. Sewer effluent can be used for fertilizing irrigation.

A few factory dwellings are here treated in detail.

The type of *Quarters* built should be determined by the class of labourers the factory employs. *Fig.* 12 shows an up-to-date construction, having 24 rooms, 8 ft. × 10 ft. These rooms can be occupied by one or two persons.

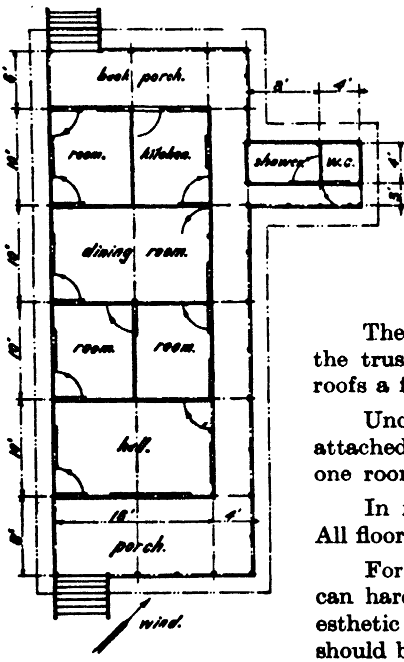
All windows are outside and grated with iron bars for protecting the occupant's property, being six feet above ground level.

The doors are in a six-foot corridor in the longitudinal centre of the building, whereas the washstands are in the lateral centre.

An annexe contains the eight shower baths, each 7 ft. × 4 ft. 6 in., and at the end of it there are eight water-closets of the same size.

At some places water-closets of the bowl type have given rise to complaints, as they were clogged by newspapers and extraneous matter. The writer has installed in such cases open Turkish closets, having a 6 in. discharge sunk in the centre of an enamelled plate, about 4 ft. × 4 ft., where two corrugated foot marks indicate the place where the occupant has to locate himself. These closets are easily cleaned by a water hose.

Doors should be fitted at the ends of the corridors to avoid draughts during cold nights. Lateral walls are built up to the roof, whereas section walls only reach to the roof trusses.



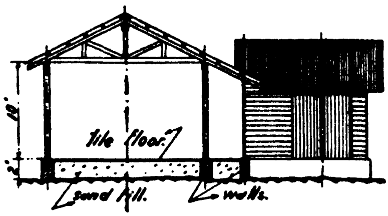


Fig. 13.—Living House.

The bath house has walls not reaching to the trusses to provide ample ventilation. In the roofs a few rotary ventilators should be placed.

Underneath the trusses coarse wire gauze is attached to prevent anyone climbing over from one room to another.

In front a wide covered porch is provided. All floors are of concrete.

For *Living Houses* at the factory a standard can hardly be applied and for architectural and esthetic reasons some variety in building design should be encouraged. Also, the size of a family will determine the size of the house they have to live in.

The illustration (*Fig.* 13) shows a *Family Living House* for a skilled workman's family.

The floors are about two feet above the ground level and covered with Spanish tiles laid in cement.

The porch, also with a tiled floor, surrounds the house on three sides. Shower and water closet are located in a small annexe, to be reached from the side porch.

The doors are indicated by arrows. Windows are provided with iron grating and shutters. The porch and outside windows should be provided with mosquito gauze preferably of copper as this will outlast iron gauze several times. Painted iron gauze should be re-painted regularly.

The construction is completely of wood. Ceilings can be made in the hall and the dining room, but often are left out for better ventilation. The kitchen walls should go up to the roof to prevent smoke and odours from entering the rest of the house. The kitchen can also be combined with the bath house annexe, which makes a good arrangement.

Roof covering is generally of galvanized corrugated sheets or French tiles on wooden board covering.

For outer walls and room partitions dove-tailed boards are used.

The walls are single-boarded, but for more expensive houses double-boarded walls make the inside appearance look better.

The writer has used "Celotex" and similar materials for partitions, ceilings and inside wall coverings. These materials are made fire-proof and are specially adapted for plantation houses.

When wooden floors are to be fitted, it is essential that they be laid on square columns of masonry or concrete and well anchored to these, about 6 feet above the ground level, as shown in *Fig.* 14. This makes the floors accessible underneath, where they should be white-washed.

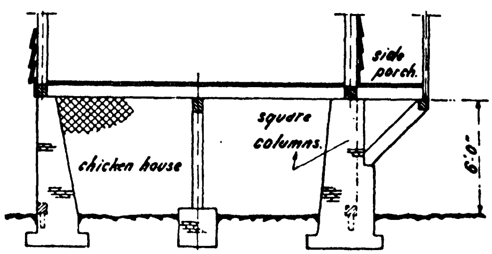


Fig. 14.—Wooden Floor Construction.

It is very convenient to have the chicken-house in this space, as many insects will then be devoured by these useful domestic birds. An ant eater, tapir or armadillo could be kept to advantage also.

Overlapping wall boards are used where the lumber comes from the factory saw mill or a local one. Stairs to the porch should have steps not more than 8 inches high and preferably 12 inches wide.

In those countries where bricks are cheap and plentiful, bungalows are generally made of bricks, having cement floors. A brick house does not need the walls painting and has more lasting qualities than a wooden structure.

15.—The Factory Yard.

The location of the factory yard depends on local or topographical conditions and the situation of the cane fields, as well as the direction of the prevailing wind. In *Fig.* 15 is shown the *Yard* of a tropical sugar factory, having about 800 tons cane grinding capacity when constructed, but this capacity has been gradually increased to about 1200 tons. For the sake of effectiveness a few additions have been included in the drawing and a few superfluous features left out.

The factory is of the straight line type, having the boilers in line behind the mills. With increasing capacity a new boiler house has been added parallel to the existing one.

The cane is discharged from the cars by a cane rake, located perpendicularly to the mill axle and the main track therefore has to be laid parallel to it. This has the inconvenience that the cane cars, when empty, have to be switched backwards, as the main tracks are in front of the mill.

It should be recollected that cane cars, having *unilateral* discharge, as is the case for big factories, have to be returned to the fields the same way they came. Such cars are hence *non-reversing.*

The loaded cars have to pass the weighing scale, located flush with the track at the west entrance of the yard. Before the crop starts the *tare weights* should be checked especially for wooden cane cars.

As the cane is cut only in the day time, there has to be sufficient track length available for storing loaded cars for the night run of the factory. Some 800 tons can be stored in the yard in *Fig.* 15, which is sufficient for say 16 hours' grinding.

In general, the yard capacity should fulfil the following equation :—

$$Y_{loaded} = G \times \frac{24 - W + T}{24} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (8)$$

where : Y = Yard capacity.
G = Grinding capacity per 24 hours.
W = Daily working hours of train crew.
T = Time for round trip of cane train.

If this equation is not fulfilled, longer working hours for the train crew will be the result.

The available storage for empty cars can be less, viz. :—

$$Y_{empty} = G \times \frac{24 - W}{24} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (9)$$

A liberal excess should be allowed in both formulae.

Since, moreover, the entire stock of cars has to be stored, it is convenient to have the main storage place at the factory yard. Furthermore, it should be arranged that both empty and loaded cars can be switched on *the same tracks* in cases of emergency.

The necessary length of track can easily be determined by the number of loaded and empty cars, multiplied by their overall length.

The minimum number of cars should necessarily fulfil the equation :—

$$N_{cars} = \frac{Y_{loaded}}{L} + S \times Q \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (10)$$

where : N = Number.
L = Loading capacity of each car.
S = Number of loading stations in the fields.
Q = Average number of cars for each station.

The sugar is loaded on the N.E. side of the factory so that sugar haulage is not handicapped by the cane movements, nor *vice versa.* A special weighing scale for sugar cars is at hand. The number of cars needed for sugar transport can be easily derived in the same way as for cane cars.

Molasses is not shipped at this factory but treated in its own distillery. Alcohol is shipped by tank cars, which can be weighed on the sugar car scale.

As the cane enters from the E. as well as from the W. side, a *half circular curve* of track is laid for reversing the locomotives. For upward crosshead reaction, locomotives should run ahead, not backwards, when hauling heavy loads. Turntables are seldom at hand at sugar mills.

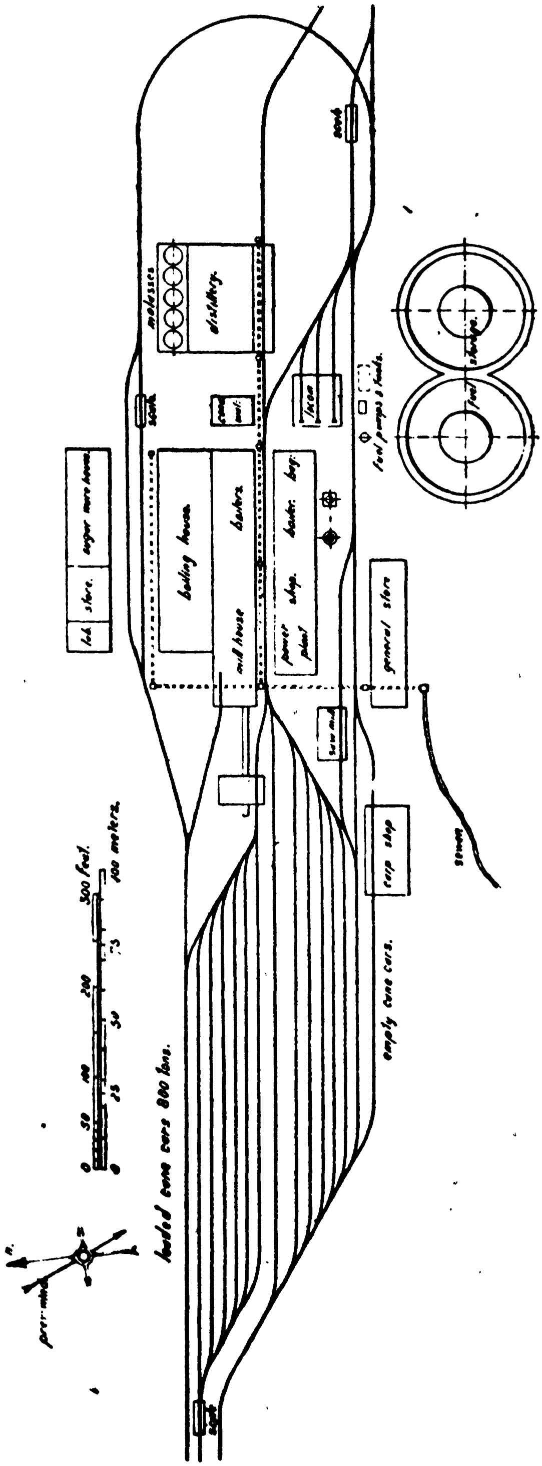


Fig. 15.—Yard for 1200-ton Cane Sugar Factory.

For *Fuel Oil* for factory and locomotive use, two large tanks are provided, surrounded by protecting earth walls, to be dimensioned in such a way that the annular space around the tank will be sufficient to contain the whole of the tank contents. This factory has a high fuel oil consumption, which should not be the case with a well-balanced sugar mill.

Oil consumption varies between wide limits and a storage capacity of 2 Imp. gals. of fuel oil per ton of cane ground in 30 days is a normal one, although the author knows instances where 7 to 12 Imp. gals. per ton of cane are burnt, and other instances where there is no fuel oil consumed.

For locomotive service, when these are fired with oil, 1 to 2 Imp. gals. are consumed per ton of cane, including shunting service in the yard and sugar transport to the main railway junction, and the necessary return service of empty cars. A storage tank of size equal to that mentioned above is required for this service, or both may be combined, and 3 Imp. gals. per ton of cane ground in 30 days will be the normal capacity.

For plantation traction a consumption of about 20 lbs. or 2 Imp.

gals. of fuel oil is required per 100 tons train-weight per mile distance covered. For coal firing about 40 lbs. per 100 ton/miles is the average figure.

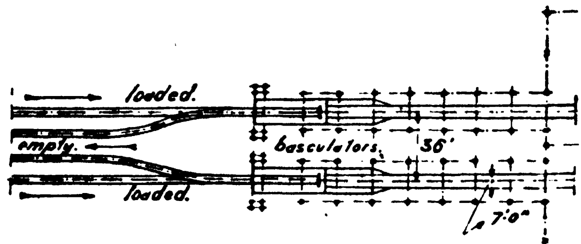


Fig. 16.—Cane Car Head Unloading for Two-Tandem House.

Oil is brought in tank cars, which are unloaded in a subterranean tank, and then it is pumped into the storage tanks.

The factory has, furthermore, its own *Sawmill*, and lumber is brought in by railway cars.

The *Carpenter's Shop* is located close to the storage yard for empty cars, as car repairs are done in this shop.

The *Locomotive Shed* has to be of sufficient capacity to hold all the locomotives and their repair shop. Round houses are not used at sugar factories, but it is sometimes preferable to have one track for each engine. Also, through-going tracks should be fitted, so that locomotives may enter from either direction.

The *Laboratory* is at the end of a separate warehouse building. This has the advantage that any vibrations from the heavy mill engines will not interfere with balances and other delicate instruments. The disadvantage is that samples have to be brought a longer distance and the control of sampling becomes therefore more difficult. The most convenient locus for the laboratory will, therefore, be adjacent to the sugar floor or sugar storage place, within the factory buildings, this being a quiet corner not subject to vibration from the heavy machinery, while it is close to the sugar making area.

The *Store House* for mill supplies is at a distance from the factory. A better place is inside the factory buildings, as workmen will lose a lot of time in coming and going.

The front of the mill house should have a connexion to the tracks, to allow for loading or unloading heavy pieces of machinery (as mill rollers and the like) with the mill house crane.

This particular factory has its water supply from a pumping station on a river and the waste-water is discharged, together with the filter mud and refuse, into a *sewer*, to be emptied into an open ditch leading to the cane fields for irrigation and manurial purposes. Closed factory sewers should have inspection pits at reasonable intervals. Sometimes drag chains are laid in these sewers for easier cleaning.

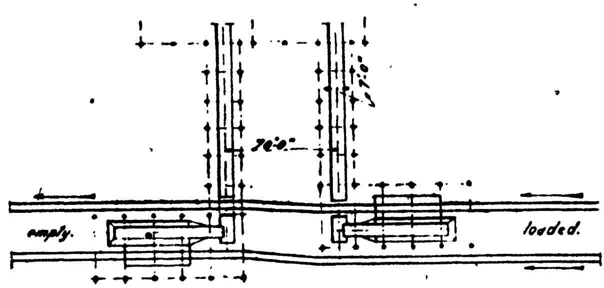


Fig. 17.—Lateral Cane Car Unloading for Two-Tandem House.

In the above factory, a *Cooling Pond* for cooling and return of the condenser waste-water is not required on account of the adjacent river supply. But in those factories having a scarcity of water, a spray cooling pond will be needed, and it should be placed in such a position that the moisture-laden air from the pond is not blown into the factory or houses and especially not into the sugar warehouse.

When cane car dumping came into practice for big sugar mills, the first installations were made for *end* dumping. Although this unloading is not as efficient as the *side* system, there is sufficient time interval between two consecutive unloadings to place a new car into position on the dumper or tipper.

Fig. 16 gives the arrangement for a two-tandem house, and the yards for loaded and empty cars can be located at a convenient place. Only double track has to be laid to each dumper. The cars are non-reversing.

Where a separate feeding carrier is used, cane cars can be emptied on a *Lateral Cane Car Dump*, as shown in *Fig.* 17 for a two-tandem house. The feeder carriers are arranged at right angles to the cane conductors. The space in front of the mill need not be large, and the car storage yards can be conveniently located at any spot, as traffic moves only in one direction.

As it sometimes happens that heavily loaded cane cars do not empty spontaneously, when on the dumper, the top cane has to be pulled off by hand with hooks. The workmen, engaged on this task, should not be hindered by the cars moving alongside the cane houses.

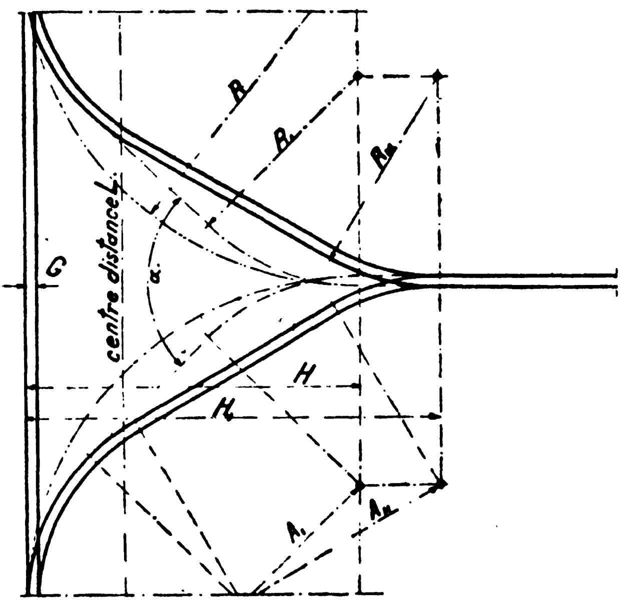


Fig. 18.—Y Switch.

The cane conductors are crossed by the railway tracks and for this purpose have to be sufficiently below ground level.

For reversing cars and locomotives, a *Triangle or Y-switch*, as shown in *Fig.* 18, can be used to advantage on some siding close to the factory or in the fields. With a frog angle of 9° 32′ (No. 6 American Standard) and a switch length of 15 ft. 0 in., the curve radius amounts to :—

56½ in. gauge	R =	282 ft. 10 in.
42 in. ,, ...	R =	193 ft. 8 in.
36 in. ,, ...	R =	160 ft. 10 in.
30 in. ,, ...	R =	123 ft. 3 in.

According to *Fig.* 18 the centre distance L is :—

$$L = 2\,R - G \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (11)$$

Curves for the same switch length and standard gauge can be used with smaller radius up to 122 ft. 2 in., giving a frog angle of 14° 15′, and a straight section can be placed between the curves. At 90° of the angle α the dimensions of the Y switch are :—

$$\left.\begin{aligned} H &= R_{,} + 0{\cdot}707\ A_{,} \\ L &= 2\ R_{,} + 1{\cdot}414\ A_{,} \end{aligned}\right\} \ldots. (12)$$

For α being 60°, these values are :—

$$\left.\begin{aligned} H_{,} &= R_{,,} + 0{\cdot}866\ A_{,,} \\ L &= 2\ R_{,,} + A_{,,} \end{aligned}\right\} \ldots. (13)$$

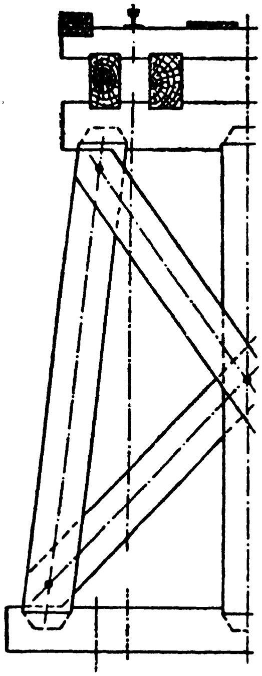


Fig. 19.—Wooden Bridge.

Where heavy hard woods are plentiful, wooden structures like railway bridges will prove cheaper than steel ones. It is nevertheless advisable that those wooden bridges be well within a reasonable radius of the factory yard, so as to be able to fight fire hazards, when these occur. A cross section of such a bridge is shown in *Fig.* 19, where hard woods like *Buchenavia capitata* or *Bucida buceras* are employed, having an ultimate tensile strength of about 12,000 lbs. per sq. inch, against 8000 lbs. for oak, stresses being parallel to grain. Shearing stresses are about 50 per cent. of these figures. The timbers of *Fig.* 19 are 12 in. × 12 in. for the columns and the bridge beams 14 in. high. The bridge carries standard gauge track and the supports are well fastened by heavy tie-rods. Dowels for vertical or diagonal timbers should not penetrate more than half-way the height of the top or lower beams, as through-going dowels, which have exposed surfaces of cross-grained timber, easily absorb moisture, and will rot in a comparatively short time.

In case of overhanging structures in the yard, care should be taken not to omit the *Clearance Gauge* for the rolling stock. This is of especial importance where rolling stock of a public railway can enter the factory yard. In *Fig.* 20 are shown clearance diagrams for standard gauge, the outer one for U.S. standard, and the other two for European continental railways. It will show the big differences existing in the sizes of rolling stock.

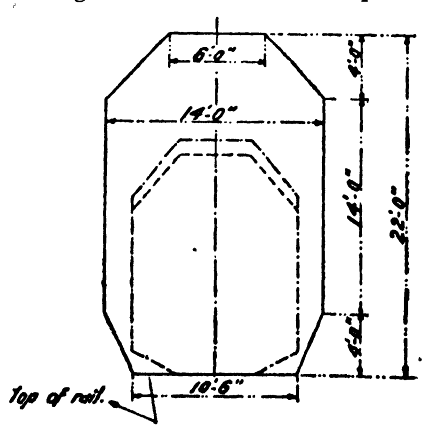


Fig. 20.—Railway Clearance Diagrams.

It should also be mentioned that pipelines, from one building to another, when carried over the tracks often prove very dangerous for the train men who walk on top of the cars or locomotives. The writer has known of several accidents due to these overhead obstructions. These pipelines should cross the tracks underneath in well-lined and properly drained channels.

CHAPTER II.

CANE TRANSPORTATION.

CANE CARTS—RAILWAY EQUIPMENT—CANE HOISTS—
CANE SCALES.

Of paramount importance is the even transportation of the cane from the fields to the factory, as regular grinding is not attainable if the cane supply falls behind schedule. Even a moderately sized sugar factory has to transport thousands of tons of cane each crop. In the previous chapter, the requisite minimum number of cars and locomotives is laid down in a few simple formulae.

For the average mill there are *two stages of transportation*, generally done by two different groups of workers, the one being through the fields to the loading station of the rail cars, and the other one from the loading stations to the factory.

Only small mills, up to a capacity of from 500 to 800 tons of cane per 24 hours, can handle their output by transportation on carts drawn by oxen, mules or horses, and these do not need transhipment. To handle larger

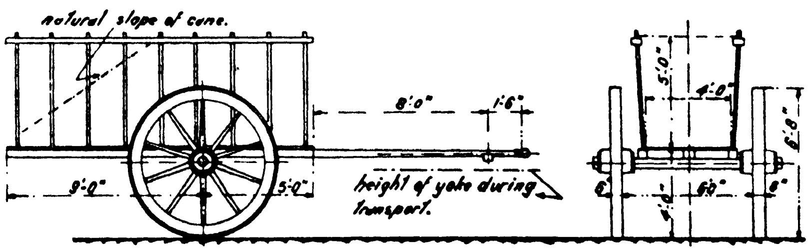


Fig. 21.—Cuban Ox Cart for Cane.

amounts of cane is very difficult with carts alone, not only on account of overcrowding of equipment in the factory yard, but also owing to the large number of animals used for traction, which have to be fed and given water during the time they wait for their turn to go to the cane carrier.

1.—Carts with Steel Wheel Tyres.

A very efficient piece of equipment for cane transportation is shown in *Fig.* 21, as used in the fields in Cuba. These *Carts* will load about 8000 lbs. of cane, cut in lengths of about 3 ft., under all weather conditions or conditions due to the soil; but when the fields or country roads are dry, up to 12,000 lbs. has been loaded.

The big wheels, having ten spokes and being 80 inches in diameter, are well fitted for fording a small river or for uneven roads with cobble stones and like obstacles, as well as for soft and muddy conditions of the fields. The wheel tyres are from 3 to 6 inches wide, and the wider ones should be preferred, as shallower tracks will result, and the cane stools will also suffer less. This of course does not matter in countries where the cane has to be replanted every year.

Care should be taken that the wooden rims have the same width as the wheel tyres, as otherwise the soil will tend to clog the overhanging part, and make riding heavy.

Hard wood is used throughout, not only for the rims, spokes and hubs, but also for the axles. This wood has an ultimate strength of about 12,000 lbs. per sq. inch. The platform has stakes of tough, almost unbreakable wood, which are held together by a longitudinal wooden lath. A twisted rope holds both sides together.

The load is nearly in balance and the yoke on the draw beam is tied to the heads of the last span of oxen. As the cane slopes down to the rear, this end of the cart platform is made longer.

The cart is pulled by six oxen, forming three spans (or pairs), the first pair called the leading span, walking at a certain distance. The resistance of an ox cart, having a total weight of 12,500 lbs., composed of 10,000 lbs. cane and 2,500 lbs. tare weight, is normally about 6 per cent., as the weight of each ox is around 1,250 lbs. and the pulling power not over 125 lbs. per ox for a day's work. At starting, an ox will pull up to 250 lbs. The advantage of the ox cart is certainly not on account of low friction, but lies in its sturdy construction, ideal for the very rough handling met with in the fields.

The cane is generally unloaded by dumping; the oxen yoked to the pole being released while the cart tumbles over to the rear, and its contents dump into the cane carrier, which has to be below the ground level for this operation.

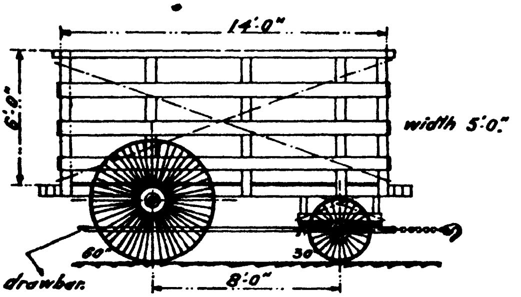


Fig. 22.—6-ton Four Wheeled Cane Cart.

A *Four Wheeled Cart* for 6 tons cane load is shown in *Fig.* 22, having a cage of 420 cub. ft. capacity. The weight of the stacked cane (cut in lengths of about 3 ft.) per cubic foot is not a constant figure. It depends on the average diameter of the cane stalks and the sugar content. For overall calculations *25 lbs. per cubic foot* will give good practical results. The cage of the cart of *Fig.* 22, therefore, will hold about 10,500 lbs. of cane, but on this kind of equipment a heavy top load is generally applied.

The wheels have flat steel tyres with riveted or welded round spokes, cast into a cast iron hub. For good roads and level surface these carts give good service and offer less friction than the two-wheeled cart of *Fig.* 21; but soft clay and mud will clog the inside of the rims and spokes, which not only makes riding very heavy, but causes breakages.

The small front wheels for the turning gear soon stick in the mud and cannot be used where cobble stones are present.

On good roads the writer has seen trains of these carts, pulled by a tractor, giving satisfactory service.

These carts are also pulled by oxen, mules or horses. The horse is not as strong and cannot stand as much hardship as the mule or the ox. *Mules* each weighing 1000 lbs. will pull 100 to 125 lbs. per mule for a day's work. And it should be borne in mind that not all oxen are fit for pulling carts and walking tied to a yoke in couples or for other team work. The teamsters need to have the skill to domesticate these animals, as otherwise they cannot be used for tractive purposes.

At several places tests have been made with *Dismountable Cane Cages* as in *Fig.* 23; these have wooden floors and the framework is made of rolled sections. The hinges of the four legs have to be fixed on *against* the direction of the pull of the truck.

The unloading of these cages has to be done by cane hoists or derricks, as they cannot be dumped. Instances have occurred where, by improper placing, the load has sunk the foot plates into the soil, so that the truck could not enter underneath, and the whole cargo had to be unloaded and re-loaded anew. Another inconvenience is that the cage sometimes falls with a jerk on the truck platform, breaking springs or axle gear. These cages, therefore, have to be handled carefully.

2.—Caterpillar Carts.

The same carts as in *Fig.* 21 can be provided with caterpillar gear. These caterpillars for *drawn carts* are different from those on tractors, as the movement is not derived from a driving axle, but only from the pulling force on the cart draw-bar. The cast steel beam, as shown in *Fig.* 26, has two trunnions of

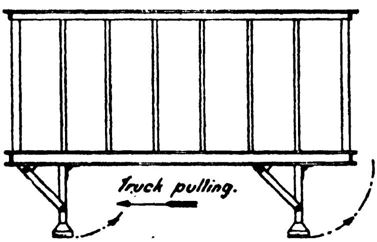


Fig. 23.—Dismountable Cane Cage.

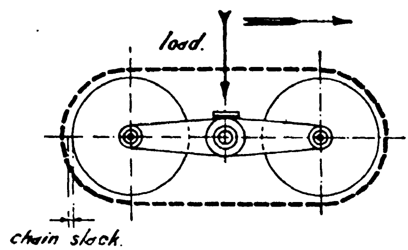


Fig. 24.—Caterpillar Mechanism.

3½ in. diameter, on which are mounted two rocking beams of malleable iron, which carry at each end two cast iron wheels. The wheels have flat rims and ride on the inside of the malleable iron caterpillar slats. During transport, therefore, a metal road, so to say, is laid in front of the wheels, as shown diagrammatically in *Fig.* 24.

The slack of the endless belt is at the rear and some arrangement is provided to take this up. The caterpillar slats have lugs in the centre, placed in the direction of movement and at right angles to the slat base. These lugs remain between the two wheels at each end of the rocking beams, so that the endless belt cannot run off in a lateral direction.

A very ingenious construction is shown in *Fig.* 25. The slats have pivoted lugs cast on, so that, with stretching, a truss beam is formed, and no intermediate support is needed against upward bending when under load.

Even the disadvantage that the slat connecting pins will be subject to wear through the intruding sand or soil will be offset by the fact that the bottom cord is *under compression* which is taken up by bearing surfaces at the sides of the slats, and a little play in the pin holes will not do any great harm.

The bearing surface on the road is about 4 sq. ft. for each belt, thus 8 sq. ft. per cart, and 5 tons of weight can be carried with a soil bearing of about 10 lbs. per sq. in. (0·7 kg./cm^2) which is a very low figure. Cane stools therefore will suffer little and for soft soils these caterpillars have great advantages over common wheels. But muddy soil will clog the caterpillar gear and inside pivots, and this is not a desirable lubricant, as it causes abrasive friction.

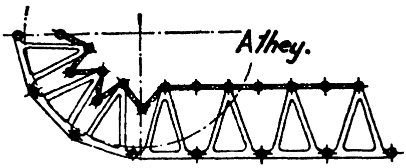


Fig. 25.—Truss Caterpillar.

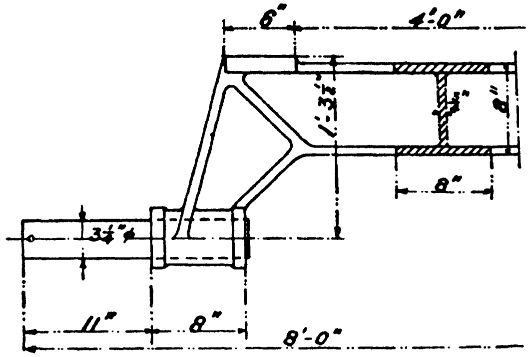


Fig. 26.—Axle for Caterpillar.

The centre trunnions have white metal bushings, and the wheel hubs are provided with ball or roller bearings.

3.—Cane Cars on Rail.

The number of cars for a given grinding capacity can be derived from the formulae given in the previous chapter. The chart in *Fig.* 27 shows the *Total Car Loading Capacity* of ten different factories. The figure mentioned at each dot is the total track length of the given factory. A dotted line is drawn, where the total car loading capacity equals the grinding capacity per 24 hours, and it will be seen that only one factory is below this line, another on it and the rest above. The shaded space covers the maximum and minimum car loading capacities of these factories and it will be readily seen that big variations occur.

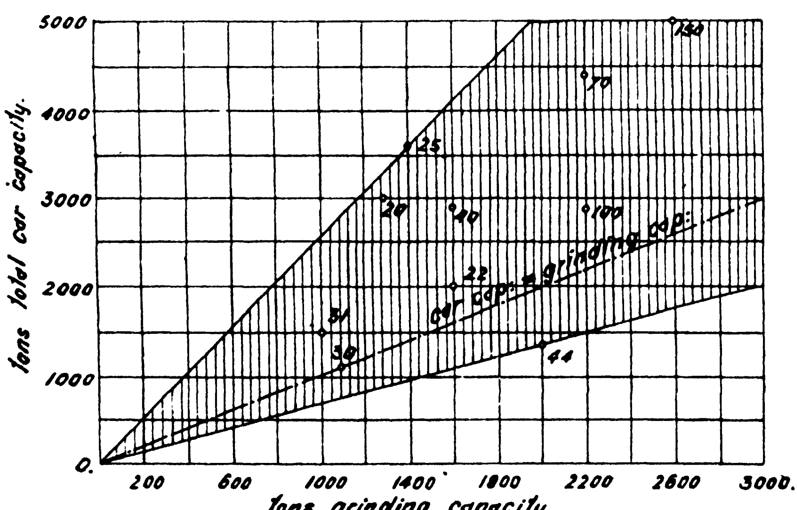


Fig. 27.—Total Car Loading Capacity Chart.

In *Fig.* 28 a *Narrow Gauge Cane Car* is given in detail. The car platform is made of 7 in. channels, well supported by tie rods and turnbuckles. The principal dimensions are :—

Fixed wheel base	3 ft. 2 in.
Centre to centre of wheel base of trucks	22 ft. 0 in.
Length of car platform	30 ft. 6 in.
Width of same	6 ft. 4 in.
Gauge	36 in.
Wheel diameter	20 in.

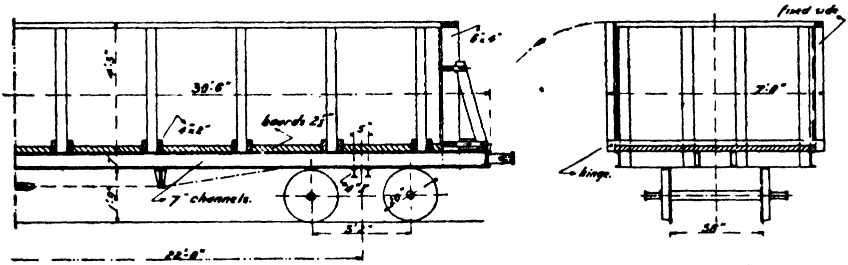


Fig. 28.—10 Tons Cane Car (Narrow Gauge).

The cage is made of wood at the factory and one side is hinged at the bottom, for unloading by a cane rake direct into the cane carrier. The hinged side is too heavy to be held by the workmen, and it is lowered and lifted by a simple hoisting gear. This side is also fully covered by boards, so the cane will slide down on a smooth inclined surface. The opposite fixed side has the boards spaced. The fronts should be well braced on the platform, as the *cane load is performing a jerking action against these fronts*, when the car is accelerated or the velocity of the train decreases. Frequently the cages have been damaged by this action. A diagonal set of bracings or ties should be placed on the fixed side of the cage. The wheel load will amount to about 3750 lbs. when loaded, and light rails can be used if the locomotives employed are not too heavy.

On the fronts of the platforms are attached cast iron bumpers having loose pins and links of round iron. This method of coupling the cars is objectionable because links and pins handled by careless labourers may fall into the cane and so pass into the crusher and mills, causing great damage to the grooving of the mill rollers. Magnetic or centrifugal tramp iron separators are not at hand at all factories and, moreover, are not invariably a 100 per cent. protection against this evil.

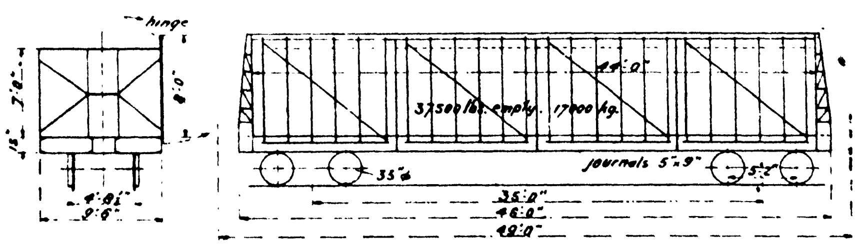


Fig. 29.—40 Tons Cane Car (Standard Gauge).

The heaviest car for cane transport in shown in *Fig.* 29. It is built for standard gauge, having a capacity of 40 short tons of loaded cane. The empty weight is about 50 per cent. of the load. These cars can be used only for very big factories and the car dumper has to be designed for the length of the cars.

Each of the four compartments will hold ten short tons of cane; and the cane, therefore, should be brought in carts having 5 or 3·3 tons load, so that an even number of cart loads can be hoisted as a whole into the cane car. Dividing the cart load is time robbing!

The wheel loads are about 15,000 lbs. and the journals of 5 in. × 9 in. give a specific journal bearing of 330 lbs. per sq. inch (about 23·5 kg./cm.²) which is an allowable figure.

The cage is made of rolled sections, angles and I beams for the horizontal and upright beams and T bars for the side spacing. The T bars are, on one side, hinged at the top end on the traversing heavy top angle, and four hinged doors are obtained this way.

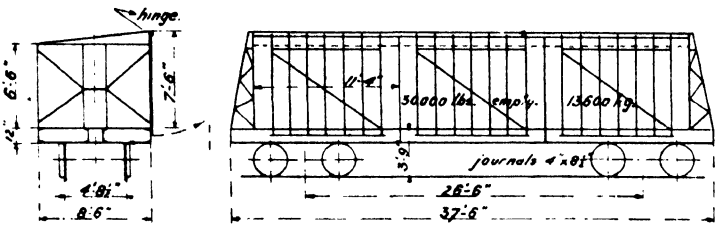


Fig. 30.—30 Tons Cane Car (Standard Gauge).

The fronts are well braced by latticed columns, to withstand the jerking action of the cane previously explained. For big cars these inertia forces are in the ascendant, especially when a top load is present.

The hinged side is 8 ft. high, whereas the fixed side is but 7 ft. The cars are unloaded by lateral dumping and it is assumed that the loose cane in the compartments will slide underneath the traversing top angle on the hinged side. This, nevertheless, is not always the case and due to shortage of cars or faulty distribution the estate hands are prone to load the cars in excess of the pre-determined height of the cage.

The same construction is applied to *Fig.* 30 for a *Three-compartment Cane Car*, having 30 short tons capacity. The wheel loads are about 11,500 lbs. and the journals of 4 in. × 8½ in. give a specific bearing of 340 lbs. per sq. inch. Overloading up to 20 per cent. is common practice in some countries. Good axle lubrication, therefore, has to be assured, as otherwise the journals soon become red hot and the axles will bend. The car then will slide on the wheels, as the writer has observed on several occasions, and the wheel treads get flattened at one spot to such an extent that turning will reduce the wheel rim beyond the safety limit.

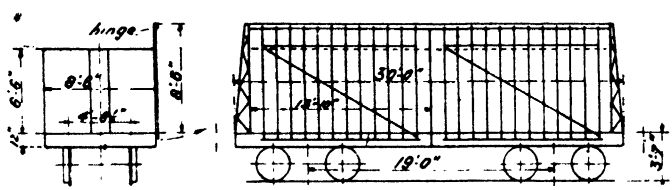


Fig. 31.—20 Tons Cane Car (Standard Gauge).

A *Two-compartment Cane Car* of 20 tons cane hauling capacity is shown in *Fig.* 31, also for standard gauge. The smaller distance between centres of the trucks makes this car suitable for curves of a smaller radius. Each compartment will hold 762·5 cubic feet of cane without applying a top load. At 25 lbs. per cubic foot, the cane weight will amount to 19,000 lbs. and thus be well within our figures.

In level cane fields a portable track can be laid and for smaller sized factories, where this is done, no transhipment is necessary. *Fig.* 32 shows a *Portable Track Cane Car* for 36 in. gauge and 5 tons holding capacity. These cars are brought from the portable track on to the main track direct to the cane rake at the factory. The wheels are 18 in. in diameter and the track should be laid as even as possible, as these cars are *easily derailed* by faulty track.

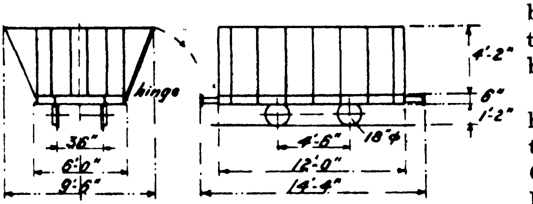


Fig. 32.—5 Tons Portable Car (N.G.).

The unloading side is hinged at the bottom, as is the case with the car in *Fig.* 28. Owing to their short wheel base, these portable cars hop or pitch at times and the front of one car may jam underneath the rear end of the next one, lifting it off the rails. The coupling links do not always prevent this happening.

Another *Five-ton Portable Car* for side unloading is shown in *Fig.* 33. It has a wooden cage and two compartments. Fifteen of these cars can be unloaded per hour. The factory concerned has two cane car dumps on the same cane carrier, one being used for larger cars.

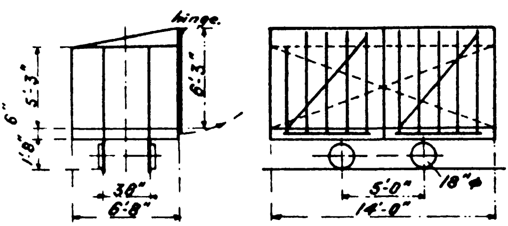


Fig. 33.—5 Tons Portable Car (N.G.).

It is obvious that these portable cars will need as much as twice the space occupied by 15 or 20-ton cars, loading the same amount of cane. So to avoid crowding the factory yard at night time, these cars are left in the fields overnight and are brought to the mills in the day time only, making one or two trips.

Fig. 34 shows a *Wooden Cane Car* for 36 in. gauge and 10 short tons cane holding capacity. The wheel load is about 4000 lbs. and the specific journal bearing about 265 lbs. per sq. inch (18·6 kg. per cm.^{2}). The two-truck arrangement gives light wheel loads and good journal bearing. The small wheel diameter makes these cars easily derailed.

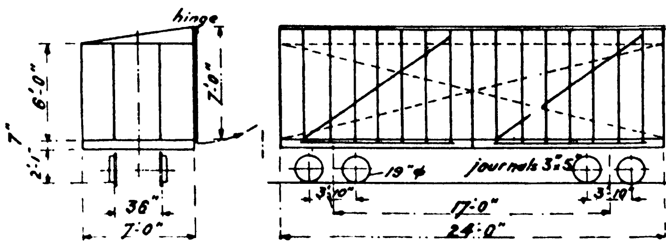


Fig. 34.—10 Tons Cane Car (Narrow Gauge).

In *Fig.* 35 is shown an *All-steel Car* for 30 in. gauge and 15 tons cane hauling capacity. The floor and sides are made from $\frac{1}{4}$ in. sheet iron. These steel cars will need fewer repairs than wooden ones, but they have to be thoroughly painted with good linseed oil paint after each crop. Fermented sugar juice has a detrimental effect on sheet iron. Sometimes $\frac{1}{8}$ in. sheets are used, but heavier sheets should be given the preference.

Narrow gauge cars for 36 in. gauge have been built up to 20 and 25 tons cane hauling capacity. Such heavy cars should not, however, be used on curves having a small radius.

Where the cane is cut in the full grown length, the cage form for cane cars is not practical, and the *Open End Railcar* as shown in *Fig.* 36 is used. The car platform is made of broad-flanged channels and bears direct on the journal boxes. No springs are used, as is usual on small cars.

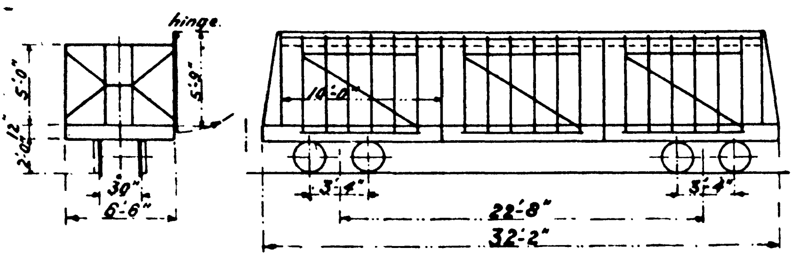


Fig. 35.—15 Tons Cane Car (Narrow Gauge).

The uprights are bent channels, or sometimes they are made from old rails, with diagonals of flat iron. The overall length of the platform and couplers has to be about 10 feet, in accordance with the length of the cut cane.

4.—Rail Car Details.

Several details of rail cars should be considered more closely, as they are essential for good operating performance and the selection of proper cane transport equipment.

For medium size cars, the *Journal Bearings* as shown in *Fig.* 37 are widely used. The cast iron grease box contains a cast iron wedge piece, which holds the bronze liner, and serves as a reinforcement of the latter. The bottom part

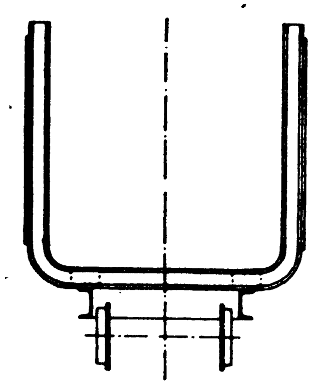


Fig. 36.—Open End Railcar.

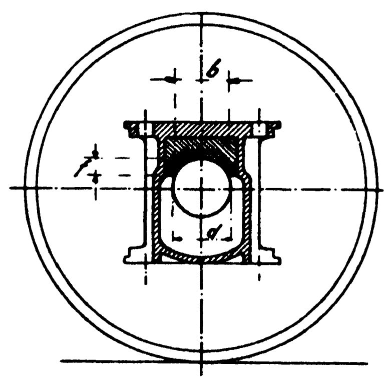


Fig. 37.—Brass Journal Bearing.

of the grease box is filled with cotton, drenched in heavy oil. The grease box is provided with a spring cover, so that it will remain closed during transportation and prevent dust entering. To the writer are known instances where the oil-saturated cotton was removed by the drivers to grease their ox carts with, and special locking devices had to be provided. The journal boxes are used in four-wheel trucks and have given excellent service even under adverse conditions.

The width b of the liner is about 95 per cent. of the journal diameter d; it does not cover the journal down to the centre line, but remains above it.

The sideward thrust, caused by the pulling force or the braking effort, is taken up by a projected surface $f \times l$, where l is the length of the journal. This sideward thrust is only a fraction of the wheel load.

In *Fig.* 38 the *Wheel Periphery* is not considered as a circle, but as a polygon with very small sides x, which are rolling upon the top face of the rail. A tipping momentum exists of the value :

$$M_r = Q \times x$$

where Q is the wheel load. The momentum is called the *rolling friction.*

The journal will cause a *sliding* friction, contrary to the above-mentioned one, equal to $Q \times \mu$, where μ is the journal friction coefficient. It will be recollected that this coefficient is different from the one for friction on a plane surface, as the load is acting on the projection $d \times l$, whereas the developed bearing surface is nearly $\frac{\pi}{2} \times d \times l$.

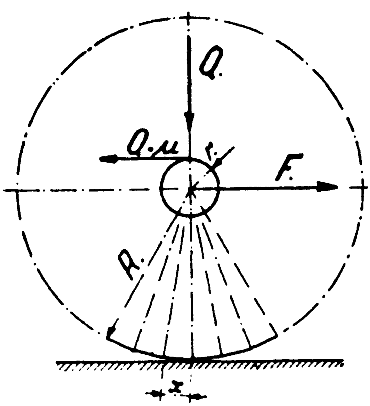


Fig. 38.—Wheel Friction.

As the journal friction is acting on a radius r of the journal, the momentum prevails : $M_j = Q \times \mu \times r$.

The pulling force F acts at the wheel centre on the wheel radius R, and amounts to :

$$F = \frac{Q\ (x + \mu \times r)}{R} \quad \ldots . (14)$$

For rail traction x is considered to be 0·02 inches, although exact investigations have not been made up to the present.

The coefficient μ depends on velocity, bearing pressure, temperature, kind of lubricant and kind of materials of journal and liner. For bronze liners and mild steel axles an average figure of 0·014 is taken for overall calculations. For a 33 in. wheel and a journal diameter of 4 in., a wheel load of 10,000 lbs. will have a friction resistance, viz., require a pulling force to move, of :

$$F = \frac{10{,}000 \times (0{\cdot}02 + 0{\cdot}014 \times 2)}{16{\cdot}5} = 29 \textit{ lbs.}$$

or about 6 lbs. per ton of 2000 lbs.

Wind pressure or load components due to track gradients are not considered in this calculation.

Special hard bronzes on case-hardened bushing around the journals will stand considerably higher bearing pressure than the commonly used material and the friction coefficient is then reduced to about two-thirds of the previously mentioned figure.

White metal has sufficient bearing resistance, as the maximum allowable limit of about 750 lbs. per sq. inch is seldom reached in car journals. For small cars white metal is extensively used and an average quality is given in the following analysis :—

Copper	8	per cent.
Tin	2	,,
Zinc	88	,,
Antimony	2	,,

There are a number of other alloys on the market, some using lead to a big percentage and others nickel.

The friction coefficient of white metal is lower than that of bronze and equals that of hard bronze. The inconvenience is the *low melting point*, generally around 440°F. (225°C.), whereas bronze has a melting point of about 1650°F. (900°C.).

When ordering white metal, the following data should be known :—

Melting point.
Friction coefficient under car lubrication.
Ultimate bearing resistance.

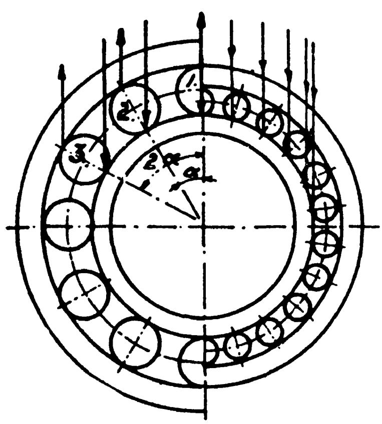


Fig. 39.—Ball Bearing.

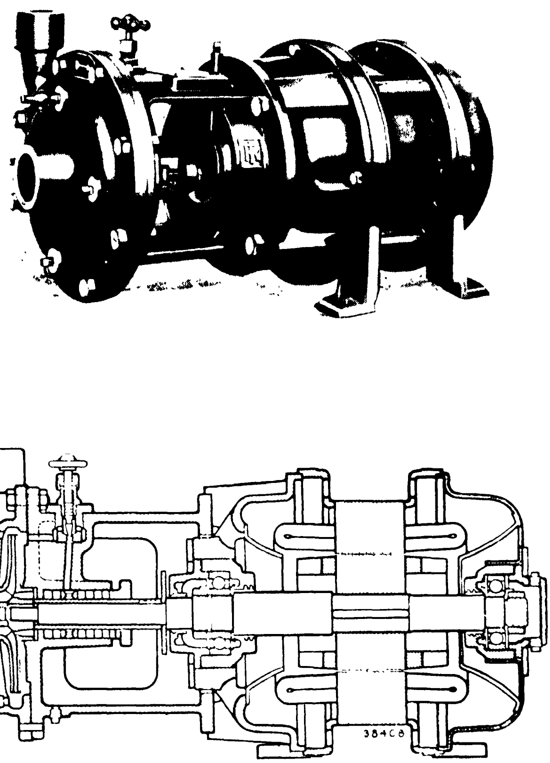
To reduce operating and especially starting friction, *Ball and Roller Bearings* have found an extensive field of application on journals of transportation equipment. The ball bearing had its rise with the early bicycle and the roller bearing is now extensively used in the automobile industry and its value has been amply tested on heavy road vehicles such as trucks, having wheel loads up to five tons. In the mining industry, as well as on contractors' trucks on rail, ball and roller bearings have withstood tests in excess of what is required for cane transportation.

In *Fig.* 39 the principle of a *Ball Bearing* is shown. Between case-hardened runners are placed a certain number of accurately ground and hardened balls. The runners have a rounded groove, in which the balls revolve.

The left side shows only five bearing balls, whereas the right side has eleven balls on the bearing half, but of smaller diameter. As the bearing takes place on small spots on the balls, caused by *micrometric surface flattening within the elastic limit* of the ball material and that of the runners, it should be obvious that the best roller bearing should have a large number of balls ; but this is only practicable when balls of a small diameter are used, and small balls will split more easily under load.

The design is guided by these rules and one is bound to adopt a certain ball diameter for a given load.

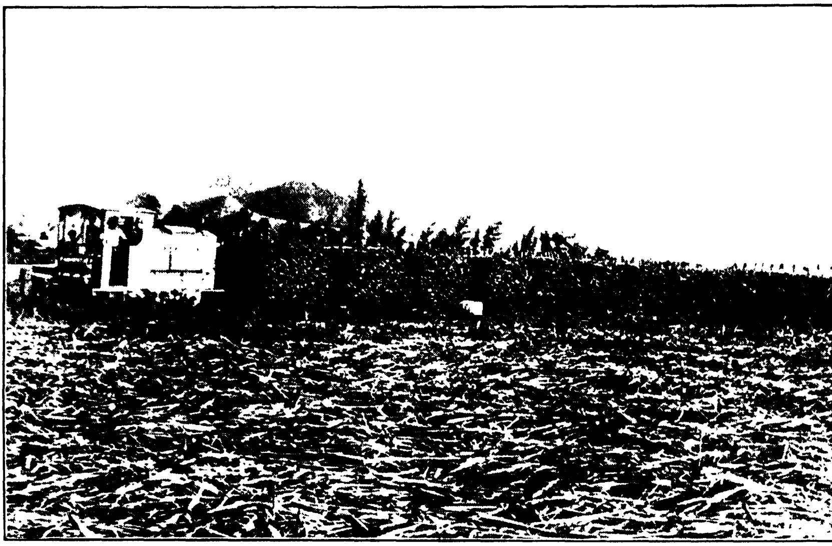
The top ball will bear the biggest share of the load, whereas the others in the top half will carry less, and the balls in the lower half are not bearing at all. The balls are spaced to avoid friction between them and therefore a ball cage, generally of stamped material, is provided, wherein every ball has its corresponding place.



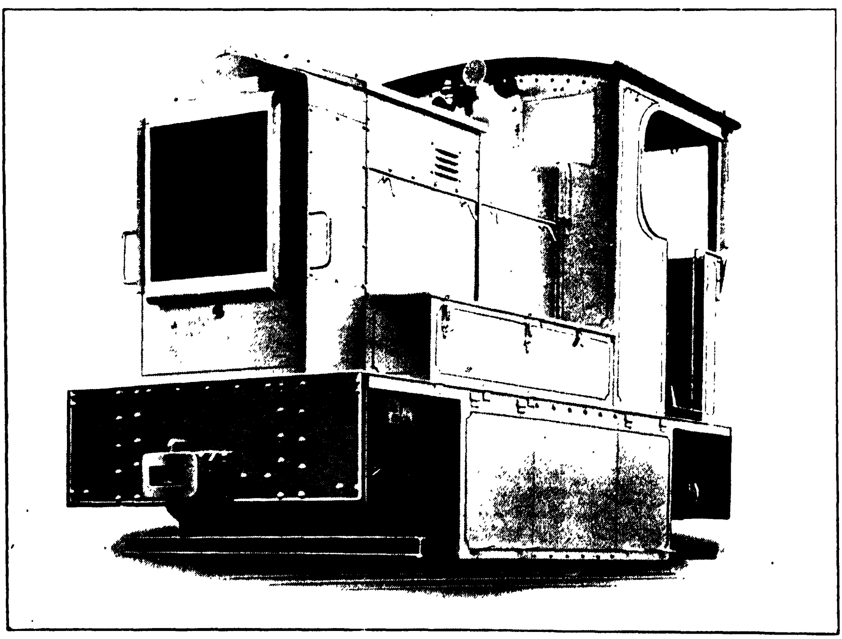
1-IN. MOTOR PUMP FOR COUNTRY HOUSE WATER SERVICE.

(Ingersoll-Rand Co., Ltd.)

PLATES 3 & 4.



Hauling Cane with Diesel-Electric Locomotive.
(Armstrong, Whitworth & Co., Ltd.)



Diesel-Electric Plantation Locomotive (12 Tons).
(Armstrong, Whitworth & Co., Ltd.)

At the left side, the angle α of the ball spacing is 30° and it is obvious that balls No. 2 will only carry *on the outside* :

$$P_2 = P_1 \times \cos \alpha$$

The load P_2 is transferred diametrically through the ball on the inner runner and the vertical reaction *on the inside* amounts to :

$$P_2 \times \cos \alpha = P_1 \times \cos^2 \alpha \quad \ldots\ldots\ldots\ldots \quad (15)$$

According to this calculation, the five balls on the top half will carry only 3 P_1.

P_1 is the maximum load on a ball and it is assumed that the ball is subject to a splitting force over the full diameter, as the load is transferred on to this same diameter. The ball diameter, therefore, has to fulfil the following equation:

$$P_1 = \pi \frac{d^2}{4} \times K \quad \ldots\ldots\ldots\ldots \quad (16)$$

where d is the ball diameter and K the allowable splitting stress in lbs. per sq. inch, when d is given in inches.

Very extensive tests by STRIBECK have yielded K = 3600 lbs./sq. in. max.

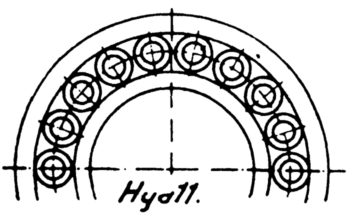


Fig. 40.—Straight Roller Bearing.

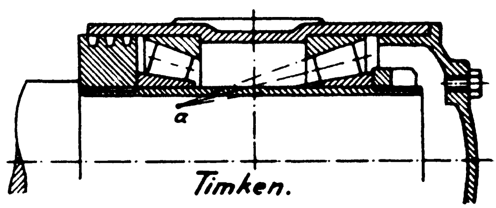


Fig. 41.—Conical Roller Bearing.

The bearing capacity of a ball bearing therefore lies within certain limits, and for loads beyond these limits the *Roller Bearing* has been designed as a logical consequence. Instead of a bearing spot, as on a ball, there is a bearing line or small bearing rectangle.

The first roller bearings of simple design had no roller cage and the rollers were laid against each other between the runners. This implied, of course, friction between the rollers when they rotated, but a simple construction has allowed this type of bearing to be used on many a small motor car, also on light cane cars.

Formula (16) for roller bearings runs thus :

$$P_1 = d \times L \times K \quad \ldots\ldots\ldots\ldots \quad (17)$$

Where no case-hardened runners are used, but the rollers bear directly on the journal or inside the cast steel bearing block, K = 850 lbs. For hardened rollers and runners K = 2100 lbs./sq. in. is accepted as a very safe figure.

Fig. 40 shows an interesting *Roller Bearing*, having hollow rollers, held in a cage and spaced. It is obvious that here the microscopic deflection of the roller material will take place more efficiently than with solid balls or rollers.

Ordinary ball and roller bearings are not adjustable to take up wear and they have to be replaced when this wear has reached a certain limit. These bearings are generally low-priced and for light service there is no inconvenience in their use. For heavy loads the *Conical Roller Bearing* has been designed as shown in *Fig.* 41. A split bushing with a split tightening nut is pulled over tho journal. The inner runners or cones are placed over this bushing. The conical rollers are located and spaced in a cage and to neutralize the axial thrust two sets of conical rollers are applied. The caps or outer runners are held tightly in the cast iron grease box. By tightening the nut, wear is taken up.

From *Fig.* 42 it will be seen that the rollers on account of the wedging action will be thrown out of the runners, if not prevented by shrouds. These shrouds are placed on the adjusting runners, which can be the cones as well as the caps. The other runners are flush.

The velocities on the runners fulfil the equation :

$$v_{I} = \frac{V_{I}}{2} \text{ and } v_{II} = \frac{V_{II}}{2}$$

when V_I and V_{II} are the rolling velocities on the *inner* runner. Moreover, there has to exist the proportion :

$$\frac{v_{I}}{v_{II}} = \frac{V_{I}}{V_{II}} = \frac{d_{I}}{d_{II}}$$

and this is possible, when the intersection *a* in *Fig.* 41 of the projecting lines *x*, *y* and *z* of *Fig.* 42 falls in the centre of rotation, i.e. in the journal centre. In *Fig.* 41 the intersection is intentionally shown in the opposite direction.

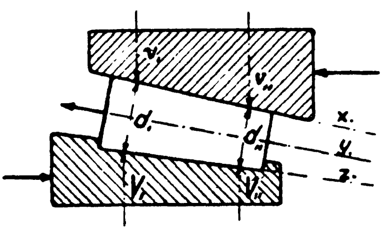


Fig. 42.—Adjustable Rollers.

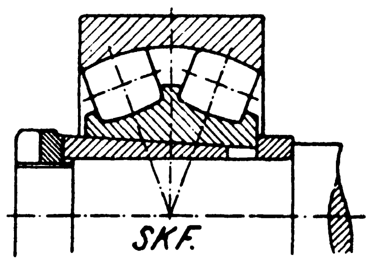


Fig. 43.—Barrel Shaped Rollers.

The conicity of the rollers has to be made very exact, as has that of the runners, since even very small differences will impair the work to be performed by the bearing. This applies to ball and straight roller bearings as well, and manufacturers maintain the highest class of workmanship for their construction. The balls, rollers and runners are case-hardened and ground to exact dimensions.

To improve upon the construction of ball bearings, *Barrel-shaped Rollers* have been designed, as shown in *Fig.* 43. The curvature of the rollers coincides with the periphery of a circle, having its centre in the journal axle. Any bending of the axle is taken up by the journal easily. The inner runner is placed on a split conical bushing, and the rollers are spaced and located within their cages. Left and right hand rollers are placed staggered to the extent of half the roller pitch.

These bearings have been used for many years in railroad journals and for wheel loads up to 7·5 tons.

The roller bearings in *Figs.* 41 and 43 will take up a lateral thrust through the inclined position of the rollers. For *Fig.* 40 special shrouds are provided with conical bearing surfaces. Ball bearings are made with runners, having an L section to take up axial thrust.

As already mentioned, the resistance of ball and roller bearings *at rest* is considerably less than with bronze or white metal bearings. The friction coefficient at rest is about 10 per cent. of the latter.

Ball bearings will not withstand heavy shocks, as the balls are liable to split, and a split ball will destroy the rolling surface of the runners. As these bearings are unimpaired by dust or grit they are still used extensively on small and medium-size cars. A wheel load of 1·5 long tons can be easily sustained.

By the use of ball and roller bearings on rail cars, the jerking effect, when the train starts, is greatly reduced; this jerking is of course detrimental to the long life and maintenance of the equipment.

For cane cars of medium and heavy loading capacities, the four or six wheel construction with *fixed wheel base* has had to be abandoned on account of the curves on plantation railways. The standard *two-truck* or bogie car, each truck having four wheels and a short fixed wheel base, has found an extensive application for plantation work.

An interesting construction is shown in *Fig.* 44. It is the so-called *Arch Bar Construction*, extensively used on early American railroads. For cane cars it is still widely used on account of the simple construction and ease of repair. The top arch is subject to compression and is therefore of heavier flat iron than the rest. The truck bolsters are made of two joists (I beams) well supported on springs, which rest on the inside of a heavy channel. Two double springs are used for heavy loads. The bolsters rest on a turntable with a secured pivot. To prevent swinging, small bearing surfaces on the outsides of the bolsters are provided and the curvature of the road should not exceed the limit, that by the gyrating of the truck on the curve the upper bearing surfaces should project beyond the lower ones on the truck bolster.

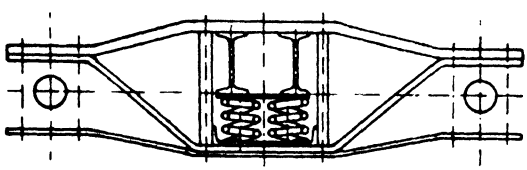


Fig. 44.—Four-Wheel Truck.

Although the design of *Fig.* 44 is widely used on cane cars, it is now in many instances replaced by the *cast steel integral box truck*, which has greater strength, less weight and fewer parts, resulting in less maintenance cost.

The *wheels* for light cars are made of pressed steel. For heavy car loads cast steel wheels are used. The use of wheel tyres of mild steel and shrunk on the cast iron wheels has been largely abandoned. Wheels of special cast iron and having chilled rims, which have an increased Brinell hardness and are less subject to wear, are now widely used.

On many narrow gauge rail cars there are no *brakes* and the total braking effort has to be achieved by the locomotive. But with greater speeds and especially on *gradients*, it is dangerous to run trains without braking devices other than that on the engine. In some countries the government railway supervision enforces the brakes as an obligatory part of the equipment. Hand-brakes are used for light haulage, but for heavy trains automatic air braking equipment should be insisted upon.

In *Fig.* 45 is shown the *Braking Gear* for a four-wheel truck. The brake shoes for each pair of wheels on the same axle are held by a transverse beam, operated in the centre by a double-armed lever. The lever lengths *a* are equal and this lever therefore is placed in an inclined position to keep clear of the track. The pulling force P on the brake rod will give a total reaction of $2\ P$ on each set of two wheels, or once P on each wheel.

The brake shoes are located below the wheel centre, so as to give an upward component force. The braking force has to be determined in such a way, that sliding of the wheels upon the rails shall not take place, as this would destroy the wheel periphery. Generally, it does not happen with loaded cars, but it will occur with empty ones, especially when a train is made up of both empty and loaded cars.

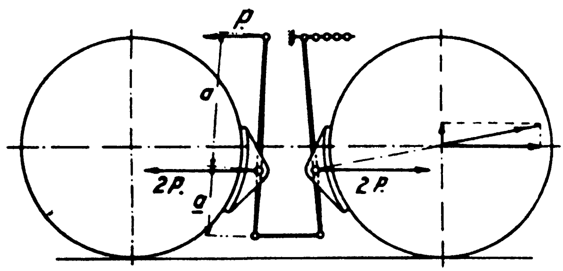


Fig. 45.—Scheme of Truck Brakes.

The brakes are provided with removable brake shoes for easy replacement. Wear on brake shoes will amount to about 5 lbs. per 1000 wheel-miles, although this cannot be considered as a fixed figure.

The friction between the wheel rim or tyre and the brake shoe depends upon the velocity on the periphery of the wheel and the condition of the rail, dry or wet. The coefficient μ, according to WICHERT, amounts to :

$$\mu = \beta \frac{1 + 0{\cdot}018\ V}{1 + 0{\cdot}096\ V} \qquad (18)$$

where V = car velocity in miles per hour, and :

$\beta = 0{\cdot}45$ for dry surfaces
$\beta = 0{\cdot}25$ for wet surfaces.

According to GALTON, the rail friction, i.e., the friction between the rail and the wheel tyre, is to be taken as :

$V =$	0	5	10	25	40	60 miles per hour
$\mu_r =$	0·33	0·273	0·242	0·166	0·127	0·074

According to *Fig.* 45, the brake shoe friction should be less than the rail friction, as otherwise sliding of the wheels on the rail will occur, thus :

$$P \times \mu < Q \times \mu_r$$

where Q = the wheel load.

Another important detail of a rail car is to be found in the car couplers. The inconvenience of the loose pin and link construction has been explained already. The cars should have a buffer arrangement, and those with spring attachment should be preferred.

With *chain or link couplers* it is important that the chain pull be centrally on the draft gear in a horizontal as well as in a vertical direction. The first condition, in the horizontal plane, is always adhered to, but this is not always the case in the vertical plane. Often the chain links are attached on top or below the buffer bars on a hook or cam. The car pull, therefore, will be out of the central axis and the drawbar gear will bend in the long run. The writer has seen instances where bending had taken place to such an extent that the buffers did not meet each other when coupling the cars in a train.

Automatic Couplers as shown in *Fig.* 46 are standard equipment in the U.S.A. and are also extensively used on industrial cars in the U.K. and continental Europe. For cane cars these buckeye couplers have given reliable service.

The couplers have swinging tongues *b*, which can be secured by locking bars *a*, operated from alongside the car. To disengage two coupled cars, one locking bar is lifted, which can be done when the couplers are not under tension. The tongue then opens the mouth of the coupler. To couple two cars, the tongues are opened by hand, before the cars bump together.

As the cars suffer when bumping together, and also when the train starts moving, a *friction drawbar gear* has been designed, of which the essentials are shown in *Fig.* 46. The bumping effect is indirectly taken up by the springs, as direct bumping will cause breakages of these latter. When the drawbar is pushed, the friction gear bottom plate is touching the lugs *c*, whereas the lugs on the opposite side are clear. The inclined surfaces of the blocks *e*, slide on similar surfaces of the wedges *d* and compress the springs *f*. The bumping effect is considerably reduced by the friction on these inclined surfaces.

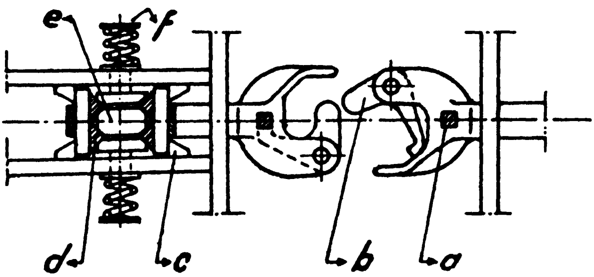


Fig. 46.—Automatic Couplers.

5.—Pneumatic Wheel Tyres.

The pneumatic wheel tyre has now invaded even the cane fields. These tyres are used to an enormous extent in road traffic and a very valuable amount of experience has been gathered regarding their manufacture and wearing qualities.

Originally devised for the bicycle, this invention soon was adopted also for the automobile, with a sectional diameter of about 4 inches. But for heavy road traffic these tyres had not sufficient resistance and *solid rubber tyres* came into use, which had a much higher bearing capacity but missed the cushion effect of the inflated tyre, causing vibrations which had a detrimental effect on the truck body and the moving parts. This led to the adoption of the so-called *balloon tyres*, i.e., inflated tyres of larger tubular diameter, up to about 9 inches, and of improved manufacture. These balloon tyres have given very good results on hard roads and will bear a load equal to any solid tyre. For field work on soft soil these tyres have also been tried out and have so far given satisfactory results.

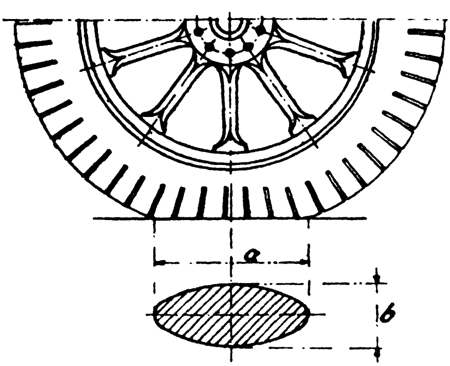


Fig. 47.—Pneumatic Wheel Tyres.

In *Fig.* 47 the principle of these tyres is shown. At the outset it should be recollected that, with rail traffic, the rail and wheel material are of sufficient squeezing or stretching strength to suffer only micrometric flattening within the elastic limits of the material. Steel tyres on soft roads will penetrate into the resisting soil, until sufficient bearing area is obtained to hold the load, and this explains why increased diameters of the wheels have come into vogue, as the bearing area is reached with less penetration. Here the soil is adjusted to the wheel rim. With pneumatic tyres, the tyre as well as the soil is doing its part to form a sufficient bearing area by flattening.

The bearing surface—it is assumed to be an ellipse although, through stiffness of the tyre material and soil penetration, the actual form might be somewhat different—has to comply with the equation :

$$L_w = a \times b \times \pi \times p_t \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (19)$$

where L_w = wheel load in lbs. ; p_t = air pressure of the tyre, in lbs. per sq. inch ; a and b are taken in inches (See *Fig.* 47).

On the other hand, resistance of the soil has to fulfil :

$$L_{ \text{r}} = A_s \times K_s \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (20)$$

where A_s = total soil bearing surface in sq. inches
K_s = the soil bearing coefficient in lbs./sq. inch.

It is obvious that the soil-bearing surface is made up by the ellipse mentioned in (19) and, moreover, by the additional area obtained by soil penetration. It follows that :

$$a \times b \times \pi \times p_t = A_s \times K_s ;$$

thus when $p_t = K_s$ there will be no penetration, and the ideal condition will be that *the soil bearing in pounds per square inch should be equal to the air pressure of the inflated tyre.* As unrolled soil will bear about 30 lbs. per sq. inch, a tyre pressure of 30 lbs. per sq. inch will not cause heavy penetration, and for field transport the inflated tyre has a future, as cost prices nowadays are within reasonable limits. In case a tyre should burst or puncture in the fields, a spare tyre with rim can easily be carried for each given group of carts. Wheel loads of 2800 lbs. have given good results in the cane fields, whereas manufacturers state axle loads of 7850 lbs. on hard roads with tyres 8 in. × 19 in. (inflated at 70 lbs./sq. in.) and a speed of 4 miles/hr.

The increase of air pressure by the flattening of the tyre when under load is negligible. The wheels are mounted on conical roller bearings and very little journal friction is present, although the rolling friction must be higher than on hard roads. The necessity for larger wheel diameters is not so great owing to the flattening or compression of the inflated tyres.

6.—Steam Traction on Rail.

The steam locomotive still holds the predominant position for haulage of cane trains on rail and it can be used with various cheap fuels, such as coal, wood or fuel oil. *Bagasse-fired* locomotives are not widely used, as the bagasse has to be briquetted for this purpose, but it should be considered for factories which have a surplus of this material. Oil-firing is very convenient for tropical countries, as it lightens the arduous task of the fireman, which on small equipment can be taken over by the engineer or engine driver.

In *Fig.* 48 is shown a *Narrow Gauge Plantation Locomotive* for 36 in. gauge, of the early American type, which still is built for many sugar estates. Of a total weight of 55 short tons, only 31 tons or 56 per cent. rest on the driving wheels, i.e., the adhesive weight. This adhesive weight is the basic factor for the *tractive force,* as it causes the rail friction, and as a matter of fact the maximum tractive or pulling force is equal to the maximum rail friction.

The engine has two cylinders, each 14 in. dia. by 20 in. stroke, on the outside of the engine frame. This construction is very practical as it facilitates inspection from the cab, when the locomotive is moving, and is convenient for lubrication and repairs. The writer has had experience with engines having the cylinders inside a high frame, built of plate and angle irons, which makes it very difficult to undertake repairs of the driving mechanism.

The above engine is of the "Consolidation" type, denominated "2-8-0," which indicates the number of front truck or bogie wheels, drivers, and rear trailing wheels in consecutive order. The eight driving wheels have a diameter of 36 in., whereas the front truck and the tender wheels are 26 in. The front and rear drivers have flanged rims while the centre ones have plain tyres, which will allow the engine to take sharp curves. The wheel centres are of cast iron and the tyres of open hearth steel.

The fixed wheel base is 10 ft. 9 in. and the front truck, as well as the two sets of four-wheel bogies under the tender, are mounted on pivots. The front truck has lateral play.

The *tender* is of the square top construction, having a water tank of U-shape, with the centre space left for fuel, which can be stacked on the top of the tender; when light fuel, like wood, is used, a rack is mounted for this purpose. The tender tank rests on a platform of rolled sections, having bolsters for the two bogies.

The wood-carrying capacity is sometimes given in *cords*, these being 128 cubic feet each.

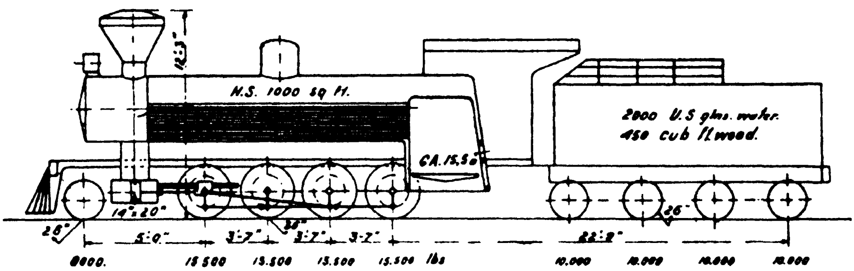


Fig. 48.—Locomotive for 36 in. Gauge. Engine 70,000 lbs. Tender 40,000 lbs.

The engine has slide valves of the balanced type, placed horizontally on top of the cylinders, which arrangement is practical for inspection and repairs.

The reversing gear is of the Walschaert one-eccentric construction, completely mounted on the outside of the driving rods.

The connecting rod has a closed head on the crosshead end with adjustable bronze bearings. The crank pin head is of the strap type.

The coupling rods have solid heads with bronze bushings and integral grease cups. Since the rods for the four drivers cannot be made in one piece (as the driver bearings must be made to move independently with the unevenness of the road in a vertical direction) they have to have joints on the crank pins. The centre crank pins should receive, therefore, the two heads of the two adjacent coupling rods. The turning joint of one of these rods for this reason is laid off the crank centre and located on an extension of the mid coupling rod.

It may be mentioned that, for heavy locomotives, the crank pin head of the connecting rod is made solid, having a floating bronze bushing. This bushing can rotate freely between the crank pin and a steel bushing pressed into the rod head.

To counterbalance the weight of the connecting and coupling rods and also part of the dynamic forces while moving, the drivers have counterbalance weights cast in.

The *boiler* is 44 in. in diameter and has a shell thickness of $\frac{7}{16}$ in. The tube plates of boiler and fire box are $\frac{1}{2}$ in. thick ; 119 tubes of 2 in. diameter having a thickness of No. 11 B.W.G., and a length of 181 in., form the greatest part of the heating surface. The firebox is reinforced by stay-bolts and bridge stays on account of its flat walls. The staybolts should have a small drilled core, so that steam will blow out when they break ; this can then be detected from the outside or inside of the fire box.

Too thin tube plates often result in the tubes not securing sufficient adhesion when expanded into the plates and troublesome leakages often occur. The writer has had to weld these tubes in some instances.

On account of the short path the flue gases have to traverse, any high efficiency of a locomotive boiler should not be expected. An *evaporation of 5 lbs. water per lb. of coal* of 12,000 B.T.U. per lb. is a fair mean for average conditions. When the boiler or the grate area is working under forced conditions, the evaporation may drop to as low as 3 lbs. of water per lb. of coal. In continental Europe, the h.p. of the locomotive engine is sometimes given, but general practice is to supply the tractive effort in lbs.

Superheating of the steam is common practice on many locomotives and the coal consumption may be reduced by 20 per cent., although the author knows an instance where no saving could be detected on plantation locomotives. All depends on the ability of the engine driver and fireman. In the case referred to, there was no record as to train load and mileage travelled or to stops, but only the coal consumption per tonnage of cane ground was calculated, as had been done previously.

The *boiler* has 1000 sq. feet heating surface, only 6·5 per cent. of this being in the fire box. The grates are of the plain bar type and the grate area is 15·5 sq. feet, giving a grate ratio of 1 : 60. A higher grate ratio, e.g. 1 : 40, will give easier firing and better evaporation, especially with low grade fuels.

The turbine-driven electric generator of 32 volts D.C. for the head and cabin lights should be mentioned. It is mounted on top of the boiler and the exhaust is discharged alongside the stack.

For giving signals a steam whistle is attached to the dome or on the top of the spring-loaded safety valves. When the locomotive has to pass villages or rural dwellings, a hand-operated bell is used instead of the whistle. On standard gauge locomotives these bells are operated sometimes by a small steam cylinder, having automatic steam release.

The brakes are of the steam-operated type and, therefore, the cars have to have hand-operated ones, when the train-load, velocity and gradients of the track demand it.

Where government supervision enforces brakes on all the cars, an air braking system should be employed, as also in case of heavy cane haulage.

The *tractive force* of this locomotive is given as 16,600 lbs. or 27 per cent. of the adhesion weight. The factor of adhesion is derived from the quotient :

adhesion weight $\div$ tractive force.

On the locomotive shown in *Fig.* 48 the factor of adhesion is 3·7, but generally amounts to 4 to 5, depending on engine horse-power and the condition of the rail. On wet rails the tractive force will diminish to about 60 per cent. compared with dry conditions. By sanding the rails in front of the drivers, the tractive force can be brought up to the full engine power. It is obvious that this can only be done for starting purposes, when the maximum tractive power may be required.

A *Tank Locomotive* has the total weight of the engine, water and fuel loaded on the drivers. The adhesive weight, therefore, is equal to the engine weight under working conditions. It will be obvious that when other conditions are equal, the wheel load of the tank locomotive will be higher than that of the separate tender type, and therefore heavier rails will be required.

Compound locomotives, having two cylinders, are not frequently used, as spares for the two different sizes of cylinder have to be carried, etc. The three and 4-cylinder compound engines are seldom to be found for plantation work.

A *Tank Locomotive of the Geared Type,* of special design, is shown in *Fig.* 49 for 36 in. gauge. The arrangement of the cylinders is different from that in the standard locomotive, as they are located in a V-position, the enclosing angle being 90°. The adhesive weight is 32 short tons under working conditions. The engine has eight driving wheels, mounted in two bogies, so the fixed wheel base is very short and the locomotive will take any curve where a cane car will go. The engine driving shaft is coupled by universal and sliding couplings to the centrally located main shaft, which has two bevel pinions of 12 teeth at each end. The bevel gears on the truck axles have 26 teeth, and are mounted on the inside axles, as shown in *Fig.* 49.

The engine is built for a minimum radius of 70 ft. of the track and a gradient of 1 in 10 can be taken. The tractive power is given as 12,800 lbs., figured on 20 per cent. of the adhesive weight, which gives a factor of adhesion equal to 5. The engine can develop sufficient power for 14,700 lbs. tractive force.

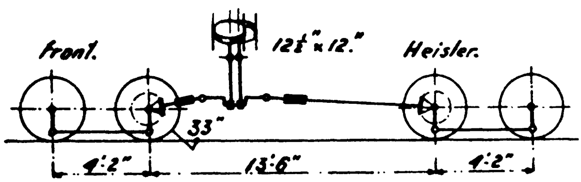


Fig. 49.—32-ton Geared Locomotive.

The two cylinders of the engine are of 12½ in. diameter and 12 in. stroke, equipped with Stephenson's reversing gear and suitable for 160 lbs. per sq. inch working steam pressure. The heating surface of the boiler amounts to 470 sq. ft. with 12·7 sq. ft. grate area ; 1,075 U.S. gallons of water are carried and 1½ cords (192 cub. ft.) of wood, as well as 700 lbs. sand in two sand boxes.

High speeds cannot be obtained with this engine, on account of the small diameter of the drivers, and the reducing gear ratio. The locomotive speed, therefore, is between 6 and 12 miles per hour, which means 61 to 122 revolutions per minute of the driving wheels, or 130 to 260 r.p.m. of the engine. The mean piston speed therefore amounts to $260 \times 1 \div 30 = 8{\cdot}66$ ft. per second, which is a low figure, as the reciprocating drive for express locomotives employs mean piston speeds up to 19 ft. per second. The designers have taken care to make the engine sturdy and well balanced, as the piston reactions otherwise would cause the engine to oscillate and vibrate.

As only two driving axles are operated by the bevel gears, coupling rods connect the drivers of each bogie with the adjacent pair of wheels.

Another 36 *in. Gauge Locomotive of the Geared Type,* but of different design, is shown in *Fig.* 50, this having also two sets of four-wheel bogies.

The engine is a vertical three-cylinder one, having 10 in. cylinder diameter and 10 in. stroke, working on a three-throw crank, with the cranks 120° apart. The engine is mounted on the right-hand side of the engine frame, when looking ahead and the boiler is eccentrically mounted to the left, so as to counterbalance the engine weight.

The main axle is on the outside of the driving wheels, having two pairs of sliding and four pairs of universal couplings.

All four driving axles have bevel gears. With an average weight of 72,000 lbs. in working order, a tractive power of 14,320 lbs. is obtained, the adhesive factor being 5·03. The bevel gear ratio is 20 : 41.

The boiler pressure amounts to 180 lbs. per sq. inch and the heating surface is 550 sq. ft. with 14·7 sq. ft. grate area.

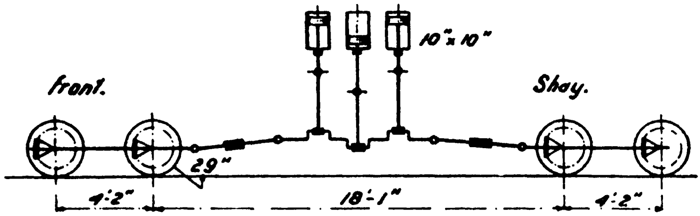


Fig. 50.—36-ton Geared Locomotive.

This engine will run on tracks having radii down to 75 ft. and on gradients of 12 per cent. (1 in 8⅓). The locomotive speed at its maximum is about 12 miles per hour. For shunting purposes in the yard, heavy cane haulage over short distances and for rolling country, the geared locomotive has advantages over the reciprocating direct-driven type and will accelerate more quickly on account of the favourable gear ratio.

For use within the factory buildings or any place where smoke is to be avoided, the *fireless locomotive* has scope for application. Moreover, the power is derived from excess steam of the factory boilers and no separate fuel consumption exists. The boiler of the locomotive acts as a steam accumulator and is filled with water for about 75 per cent. of its capacity. Live steam from the factory is injected by means of nozzles of special design, so as to cause a thorough circulation for the mixing and condensing of the steam. The *latent heat* of the steam increases the *sensible heat* of the water in the locomotive boiler. Superheated steam can be used as well as saturated. When the water has acquired a temperature close to the steam temperature in case of saturated steam, the steam supply is stopped, by disconnecting the flexible steam pipe from the factory main line ; and now low-pressure steam will be produced by converting the sensible heat of the water into latent heat necessary for the evaporation of the water. The steam pressure of the locomotive, therefore, will gradually drop until the lowest working pressure is reached and the locomotive boiler has to be re-charged.

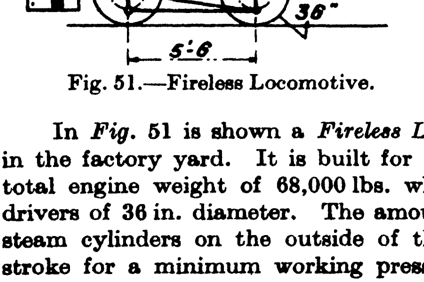


Fig. 51.—Fireless Locomotive.

In *Fig.* 51 is shown a *Fireless Locomotive* as used for shunting cane cars in the factory yard. It is built for 125 lbs./sq. inch charging pressure. The total engine weight of 68,000 lbs. when in working order rests on the four drivers of 36 in. diameter. The amount of water is about 18,000 lbs. and the steam cylinders on the outside of the engine frame are 20 in. dia. × 18 in. stroke for a minimum working pressure of 40 lbs./sq. inch. The low steam pressure explains why cylinders of large dimensions have to be used. The boiler is only a cylindrical tank, having neither firebox nor tubes and is protected by a heat-insulating cover of magnesia, and the customary outside shell of planished sheet steel. One man operates the engine and a distance of eight miles can be covered with each charge. To prevent the engine stalling, the engine driver must ascertain from the steam gauge when re-charging has to take place. The tractive power is about 12,000 lbs. at lowest steam pressure.

The fireless locomotive has proved a very useful piece of equipment.

7.—Steam Locomotive Details.

Some principal details of the locomotive, not mentioned already, may be dealt with now.

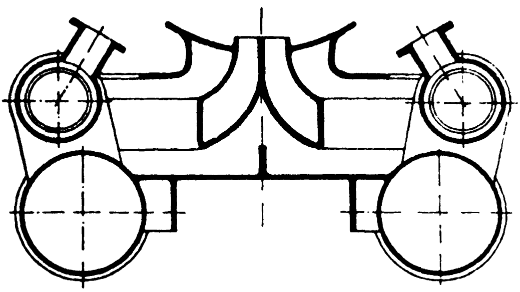


Fig. 52.—Two Cylinder Saddle Piece.

In *Fig.* 52 is shown a *Cast Iron Saddle Piece* for a two-cylinder locomotive with piston valve steam distribution. The saddle piece is an integral casting although it is generally split and bolted at the vertical centre line. The saddle rests on the engine frame and carries the smoke box, made of heavy plates, as it serves as a support for the front end of the boiler. The smoke box is provided with a valve or flap at the bottom to discharge accumulated soot and ashes.

The throttle of this engine is located within the smoke box and not in the boiler itself, which makes inspection and repairs much easier. A breaches steam pipe connects each cylinder. The blow-off pipe is cast integrally in the saddle piece and the exhaust steam is blown through a superimposed nozzle in the smoke stack, causing a forced draft.

The *Flat Valve* for steam distribution is still used to a large extent on small and medium size locomotives. It should be of the balanced type, as shown in *Fig.* 53, for which reason a spring-loaded cast iron ring is arranged in a corresponding groove on top of the valve. The ring is pressed against the machined inside face of the valve chest cover. The steam has no admittance in the enclosed circular space on top of the valve and by making this space of the same area as the valve surface, the latter will be counterbalanced. For superheated steam the flat slide valve has not given good operating results and the piston valve is thus to be preferred. The valve chest cover should have a tapped hole with a small pet cock at the centre of the circular equilibrium chamber, so that it can be ascertained if the balanced ring is really steam tight. The inside of the chest cover has to be lubricated and a groove of circular size should be provided on top of the ring and no grooves in the cover.

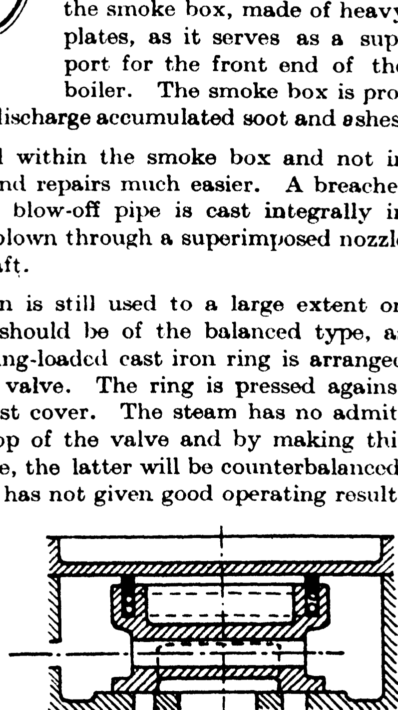


Fig. 53.—Balanced Slide Valve.

The question of the best type of *reversing gear* may next be considered. The first locomotive, that of STEPHENSON, had a gear which, although since somewhat improved, is still in use on many a locomotive. The Stephenson gear is composed of two eccentrics of equal eccentricity, one for forward and the other for backward motion, both keyed on the main driving shaft. The angle between the eccentric centre lines, therefore, will amount to : $360° - 2\delta$, where δ is the advancing angle between eccentric and crank centres.

The eccentric rods are connected to the ends of a curved slotted link and a link block, made of bronze, connected with the valve rod by means of a fork and pin, can move within the slot of the link. By raising the link and attached eccentric rods, the valve rod is connected with one eccentric, and by lowering with the other.

In mid-position there is still valve motion, and the engine will keep running in the same direction, when the resistance is not too great for the reduced power output.

A few changes to the Stephenson gear have been made by ALLAN and GOOCH, but the two-eccentric principle remains.

The eccentrics have to be located on the driving axles and therefore the Stephenson reversing gear is generally arranged within the engine frame.

For locomotives having the cylinders on the outside of the engine frame the *Walschaert reversing gear*, having *one* eccentric or counter crank is used on many plantation locomotives.

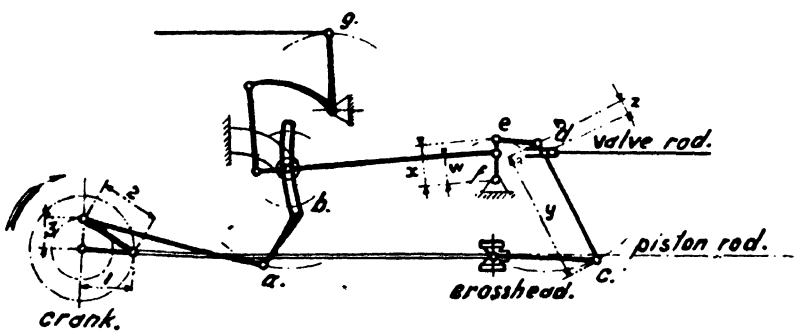


Fig. 54.—Walschaert Reversing Gear.

In *Fig.* 54 a *Walschaert Gear* of improved design is shown. On the engine crank 1 is placed a counter crank 2, forming the eccentricity 3. The eccentric rod is connected by a lever at *a* to the slotted link *b*, which can swing on pivots in its mid-centre. The link block is placed on the radius rod *b-e*, and this rod can be lowered or raised from the engine cabin by the rod *g* by means of a bent lever.

Formerly, the radius rod was connected directly to the combination lever *d-c*, which participates at *c* with the cross-head stroke. The eccentricity runs 90° behind the crank for locomotive motion ahead, i.e. backward crank motion.

It is obvious that the valve movement has a *vertical* component 3 and a *horizontal* component derived from the crosshead stroke. As the angularity of the rods will cause an unsymetrical valve motion, the design has to ensure that the rods are as close to the horizontal position as possible and large swinging angles must be avoided. The valve stroke, therefore, has to be small for well designed Walschaert gears and tends to cause wiredrawing.

To overcome this inconvenience, the lever *e-f* of *Fig.* 54 is placed in position swinging on the fixed point *f* and increases the valve travel to the proportion :

$$x \div w$$

without increasing the angularity of the main rod movements.

The motion derived from the crosshead is equal to the one derived from an imaginary eccentricity :

$$r = R \times \frac{z}{y}$$

when R is the crank radius (= half the piston stroke).

From *Fig.* 55 it is seen that this eccentricity has a horizontal position, whereas the crank eccentricity is located vertically and the resulting valve movement will have an imaginary eccentricity running $90° + \delta o$ ahead of the crank in case of outside charging valves, or $90° - \delta o$ behind the crank when inside charging valves are used.

The location of the link block in the slotted link will decrease the swinging movement caused by the crank eccentric from its maximum to 0, when the mid centre of the link is reached. On further upward movement, reversing takes place and the engine will run in the opposite direction.

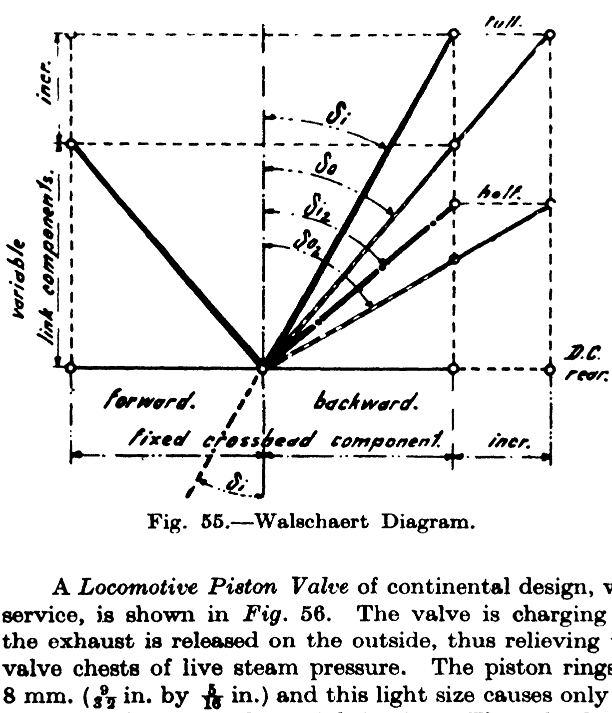


Fig. 55.—Walschaert Diagram.

The increase of the valve travel, caused through the interpolated lever *e-f*, is shown in *Fig.* 55, and the angular advance will *decrease* from δo until δi when the radius rod is in its lowest position. Raising the radius rod halfway between its maximum and zero will give an angular advance of δo_2 to δi_2. The improved design of the lever *e-f* therefore gives smaller angular advances, and early admission (lead) can be achieved over a longer range of cut-off, as well as a longer valve travel.

A *Locomotive Piston Valve* of continental design, which has given excellent service, is shown in *Fig.* 56. The valve is charging on the inside edges and the exhaust is released on the outside, thus relieving the stuffing boxes of the valve chests of live steam pressure. The piston rings have a section of 7 by 8 mm. ($\frac{9}{32}$ in. by $\frac{5}{16}$ in.) and this light size causes only a small expanding force, giving less friction and easy lubrication. The valve has a clearance of 0·75 mm. (0·03 in.) within the valve liner.

This piston valve is used on 0-6-0 tank locomotives, having six driving wheels of 1400 mm. (4 ft. 7 in.) diameter. The boiler has a heating surface of 89 sq. metres (957 sq. ft.) and the grate area is 1·47 sq. metres (15·8 sq. ft.) for coal firing. A steam pressure of 12 kg./cm^2 (170 lbs./sq. in.) is maintained and the engine is equipped with two steam cylinders, 450 × 600 mm. (about 18 in. × 24 in.). The total weight under working conditions, i.e., the adhesion weight, is 46,000 kg. (50 short tons) and a speed of 60 km., say 37 miles, per hour is obtained. The locomotive carries 4·5 cubic metres (1000 Imp. gals.) of water and 2 tons of coal.

Unlike car journal boxes, the *main driving box* of a locomotive has to transmit the tractive force from the cylinders, which are mounted on the engine frame, to the main axles and hence to the wheel tyres, where adhesion takes place. With two cylinder locomotives the full piston force acts on the main driving bearings, whereas in the four-cylinder loco the resultant of the combined piston forces is partially balanced.

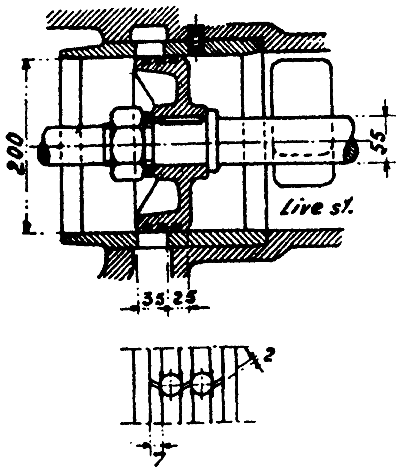


Fig. 56.—Piston Valve.

The *Main Driving Bearing* shown in *Fig.* 57 is for a two-cylinder locomotive, and the vertical bearing surface is extended to below the centre division line. The housing is made of cast steel, having bronze bearings lined with white metal. The engine weight does not rest on top of the housing, but is suspended by heavy eye bars. The space underneath the journal is equipped with a cover plate on which is laid a felt pad, drenched in heavy oil. Small spiral springs give a moderate thrust against this cover plate, so that the felt pad is pressed against the lower half of the journal.

The bearing side of the cast steel housing is lined with white metal to reduce friction on the sides of the openings of the engine frame. The main driving bearing shown belongs to a tank locomotive, type 4-6-4, having two cylinders, 508 × 660 mm. (20 × 26 in.).

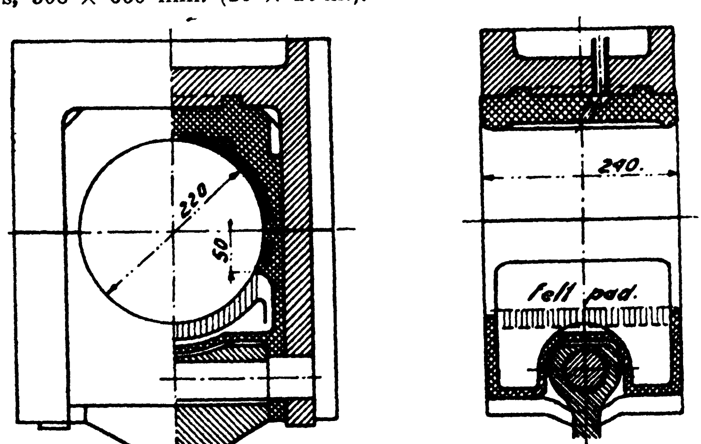


Fig. 57.—Main Driving Box.

An *Engine Frame* of cast steel and of American design is shown in *Fig.* 58 and belongs to a so-called "Prairie" locomotive, type 2-6-2, which is also used for standard gauge plantation work. The openings for the main driving boxes have an inclined side for wedge adjustment. These cast steel frames are thoroughly annealed and have done well in actual service. In case of accidents, like derailments or overturning of the engine, they might crack, but can be welded by acetylene or electric process.

The engine frame rests, through a set of springs, on the main driving boxes and truck bearings. Instead of each bearing having its separate spring an *Equalizing Arrangement* by means of levers which are pivoted to the

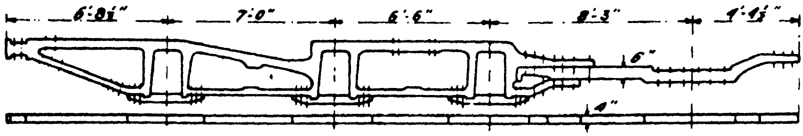


Fig. 58.—Cast Steel Engine Frame.

engine frame, as shown in *Fig.* 59, is used on some medium and heavier locomotives. Shocks are taken up by more springs, so that breakages will be fewer. The springs are of the flat steel type as used on horse-drawn road vehicles. The spring material is subject to crystallization, as are axles, wheel tyres, etc. The wheel tyres, axles and springs should be annealed at certain intervals after 50 to 75,000 miles have been covered and the springs re-hardened.

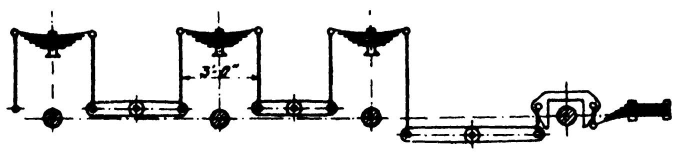


Fig. 59.—Equalizing Arrangement of Engine Springs.

The stuffing boxes now in use on locomotives have either woven asbestos or metallic packing. *A Woven Asbestos and Cotton Cloth Packing* of patented design is shown in *Fig.* 60, composed of three tongues *a*, laid between soft packing rings *b* and *c*. The edges of the tongue tend to press against the rod, and with the return stroke, when the crank end of the cylinder receives the full steam pressure, the rod movement will assist in preventing too heavy a pressure of the packing rings against the rod, thus reducing wear to a minimum. With the outgoing stroke when only the exhaust pressure is acting on the tongue-shaped rings, only a gentle pressure against the rod will be exerted. The tongue grooves are filled with oil and an efficient packing lubrication is thus provided.

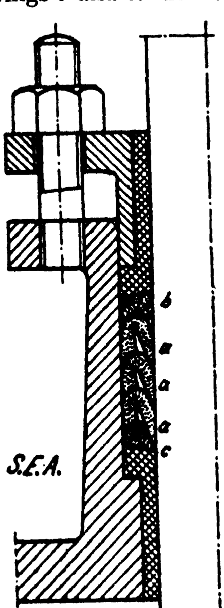


Fig. 60.
Woven Asbestos and Cotton Cloth Packing.

The heels of the tongues are reinforced by metallic wiring and the packing can be used for superheated as well as for saturated steam.

A *Metallic Rod Packing*, with which the author has obtained good operating results, is shown in Fig. 61. The white metal packing rings are split lengthwise on a spiral line and on account of the wedge action of the cover and bottom bushing, the spring pressure will close the rings gently on the rod and make a steam-tight joint. The melting point of the packing rings, of course, has to be considerably above the steam temperature, as excess heat is produced by friction, when lubrication, for instance, is lacking through clogged oil holes or the like.

The writer has seen operating performances covering four crops with this kind of packing before renewal was necessary. As the rods are liable to be covered with dust or fine sand from the tracks, whirled up by the train suction or the wind, an asbestos cord is laid in the groove on the outside of the cover. This cord has to be renewed at regular intervals when soiled.

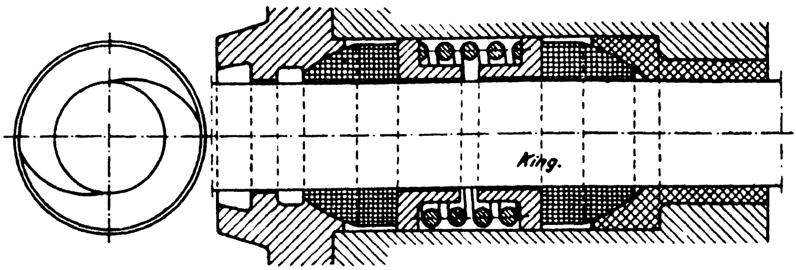


Fig. 61.—Stuffing Box.

The *Brake Gear* as used on many locomotives is shown in *Fig.* 62. The eight driving wheels have the brake shoes attached at the front, so that vertical reaction, when running ahead, will be downwards and the brake shoe suspenders therefore will be in tension. When the locomotive runs backwards, these suspenders receive compression stresses on applying the brakes, and instead of an upward reaction on the axle bearing, there will be a downward one. The brake cylinder piston force will act along the main brake rod, which is connected to the eight brake shoes. But it is of great importance for the braking effect that equalizing levers be present in the brake gear, so that there is equal braking effort on all the drivers. The tender is equipped with a brake gear similar to the one used on car trucks (*vide Fig.* 45).

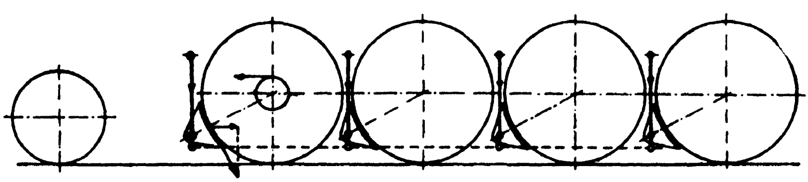
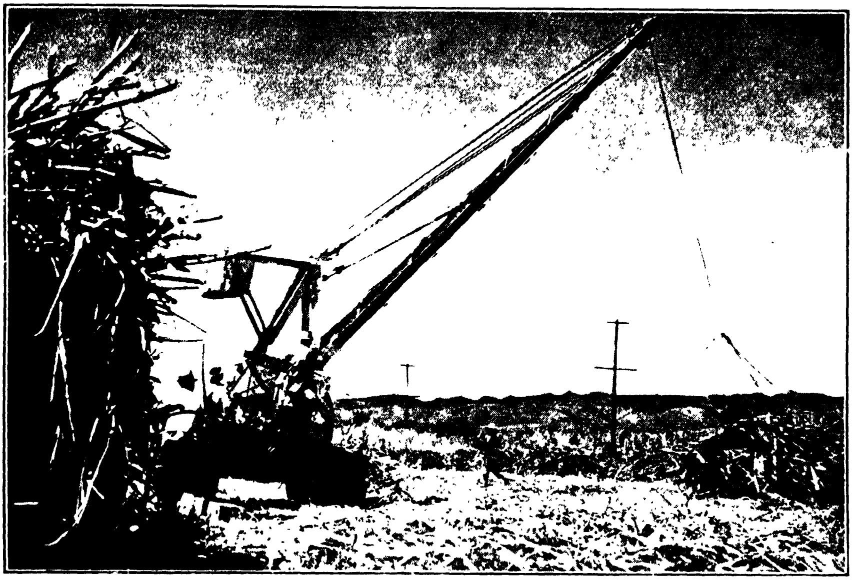


Fig. 62.—Brake Rigging.

The flat slide valve and the piston valve are universally used on plantation locomotives, as they form a very dependable mode of steam distribution and are well fitted for reversing the direction of rotation, as demanded for locomotive service.

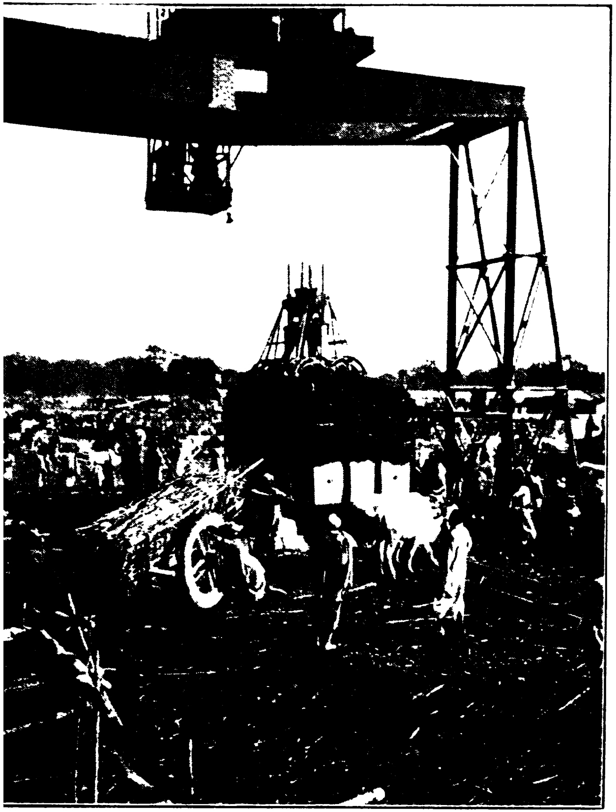
The poppet or drop valve, which came into use around 1865 on stationary engines on the European continent and is now used on modern engines in the U.K. and the U.S.A. as well, has been tried out on locomotives. The first application was made in Germany some 25 years ago ; and there being four balanced, frictionless distributing units, a better steam distribution should be obtained.

In *Fig.* 63 is shown a cross section of a *Poppet Valve Cylinder* for a narrow gauge locomotive of Continental design having two cylinders 260 × 450 mm. (10¼ × 18 in. approx.), the gauge being 2 ft. 6⅝ in. (780 mm.).



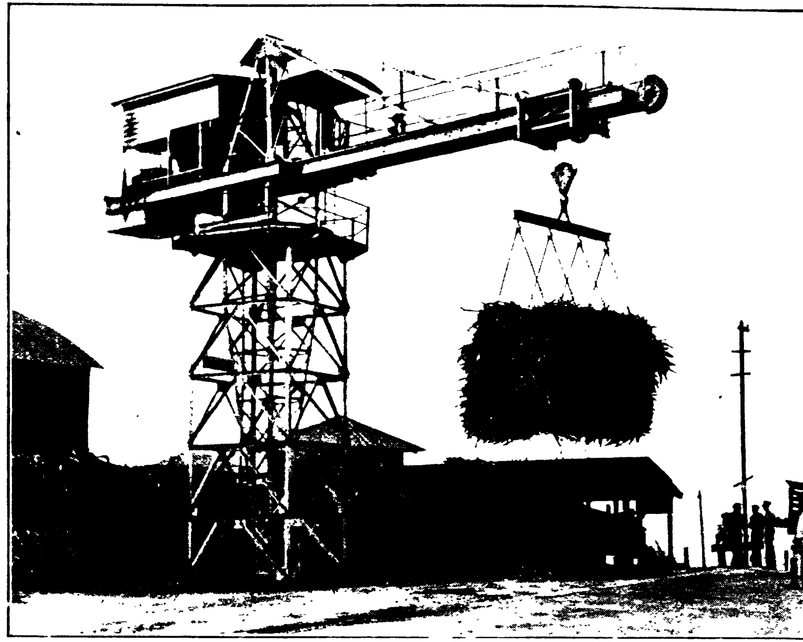
(*Photo. by Bucyrus-Erie Co.*)

LOADING CANE IN THE FIELDS WITH CATERPILLAR CANE HOIST.



Left—

HOIST FOR UNLOADING CANE IN BRITISH INDIA.
(*Gebr. Stork & Co.*)



5-TON TOWER CRANE, DIESEL-DRIVEN, 28 FT. MAXIMUM RADIUS, HOOK RISING TO 25 FT. ABOVE GROUND. (*J. M. Henderson & Co., Ltd.*)

The four poppet valves are in line parallel with the cylinder axis, the valve being double-seated and nearly balanced, while the operation is rendered practically frictionless by using unpacked valve spindles with labyrinth grooves.

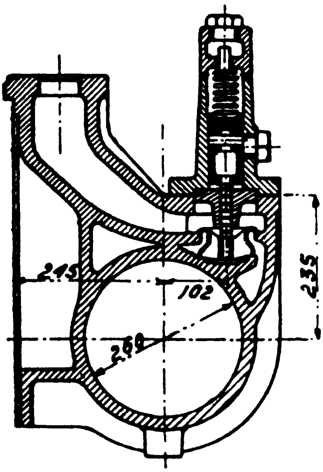


Fig. 63.—Poppet Valves.

The *Poppet Valve Gear*, as invented by LENTZ, is shown in *Fig.* 64, the distribution rod being moved to and fro under spring-loaded guide rollers, which will follow the curved slots and thus close or open the valves.

The Lentz poppet valve gear for locomotives has not found general introduction.

A more recent construction is the *Caprotti Poppet Valve Gear* for locomotives, as shown in *Fig.* 65. The invention is of Italian origin and has been tried out thoroughly on Italian railways, being applied now over the European continent, Great Britain and the U.S.A.

The four valves are placed in pairs on top of the cylinders, the admission valves on the inside and the exhaust ones on the outside. The valves are operated by knee levers, similarly to the operation of the valves of internal combustion motors. As contrasted with the Lenz valves, the Caprotti valves open, when pushed down and they are spring-loaded and double-seated.

On the main driving shaft of the engine is mounted a set of bevel gears, which drive a longitudinal countershaft and through another set of bevel gears transmit the drive to the cam shaft running parallel to the driving shaft on top of the cylinders, which is fitted with the cams which operate the aforementioned knee levers.

The cams are shifted by a very ingenious device for forward and backward motion of the engine, and each valve has its independent movement and is not affected by the operation of the other valves. A very good steam distribution can thus be obtained, and as the valves will stand the highest superheat, great economy in steam consumption is achieved and the indicator cards show considerably less wiredrawing than is possible with a well designed Walschaert gear. To obtain the same mean steam pressure a smaller admission is required with the Caprotti valve gear. From block tests in the U.S.A. in 1927 it was found, by comparing two locomotives of the same design and size, but one equipped with Caprotti poppet valve gear and the other with Walschaert gear and piston valves, that the Caprotti gear at 35 per cent. cut-off gave a mean pressure of 105·6 lbs./sq. inch, whereas the Walschaert gear gave at this same cut-off only 97·3 lbs./sq. inch. The speed attained was 14·5 miles per hour. Further tests showed the following :

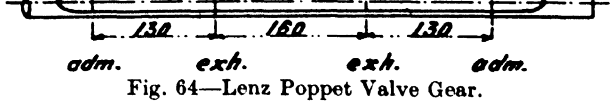


Fig. 64—Lenz Poppet Valve Gear.

at 29 m/h. mean pressures were respectively 79·8 and 67·5 lbs./sq. in.
at 43 m/h. ,, ,, ,, ,, 34·4 and 24·2 ,,

The cut-off at 29 m/h. was 35 per cent. and at 43 m/h. only 11 per cent., and these tests showed the great advantage of early cut-off with its reduced steam consumption, as the engine will run with earlier cut-off when speed has been attained. It has been shown by extensive tests in practical operation that a *reduction of* 20 *per cent. in steam,* i.e. coal or fuel consumption, can be achieved.

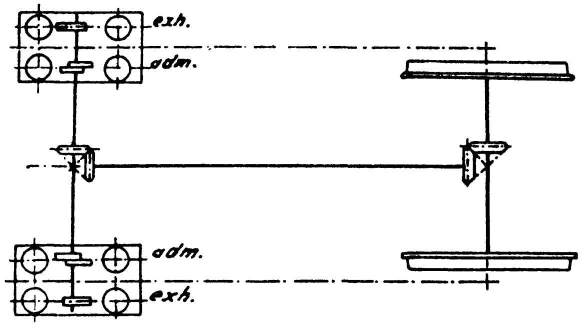


Fig. 65.—Caprotti Poppet Valve Gear.

The Caprotti locomotive can be used for plantation work, as it has fewer moving parts subject to wear than any link motion. But the first cost will of course be higher and as a plantation locomotive is only in use a part of the year—i.e. during crop time—a careful estimate should first be made, to prove that the higher capital expenditure is justified.

A further mechanical advantage of the Caprotti gear is that the gear box on top of each cylinder and containing the whole cam-shaft, lever and reversing mechanism can be readily dismounted by merely detaching a couple of bolts. After removal of these bolts or screws, the whole gear box will slide off the cylinder. The camshaft is provided with a set of sliding couplings for this purpose.

As mentioned already, the construction of the cylinders on the outside of the frame makes inspection and repairs easier than when they are located within the engine frame and the moving parts are less accessible. From *Fig.* 66 it is, nevertheless, seen that with high cut-off a *couple of forces* to the amount of

$$P \times a$$

perpendicular to the engine centre line prevails.

This couple of forces is in proportion to the distance a and therefore will be larger for cylinders mounted outside. Every engineer who has stood on the footplate knows the wobbling effect of these two-cylinder engines when pulling heavily, and this is especially noticeable on engines with a short wheel base. Lateral play in main driving boxes, or between the wheel flanges and the rail, should be kept within the smallest limits possible.

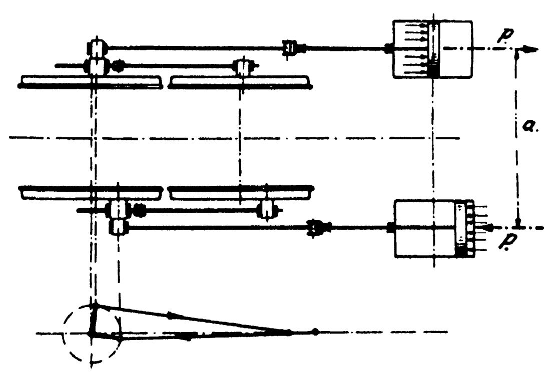
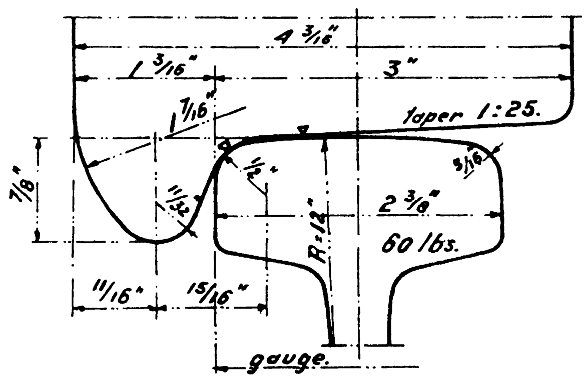


Fig. 66.—Horizontal Engine Couple.

As the whole load of the engine is carried by, and the tractive effort depends on, the *wheel tyre bearing upon the rail,* this should be considered more closely. In *Fig.* 67 is drawn the head of a rail section of 60 lbs./yd. and a standard flanged wheel rim of American design. To the average mechanical engineer the situation does not seem very favourable. The rail head is curved and is symetric to its vertical axle. The wheel rims are tapered from the flange to the outside edge to assist in keeping them centrally on the rails. For this same reason old horse-drawn vehicles had the wheels mounted not in a vertical plane but inclined slightly outwards, so that they remained more steadily on the road.

From *Fig.* 67 it is further seen that the wheel rim bears only at two spots on the rail, marked with a pointer. It is obvious that the biggest wear will be on those spots, and old rails and worn wheel rims will indicate this very clearly. The wheel tyres will wear more rapidly than the rails and they have to be turned on a lathe when worn excessively. Locomotives wheel tyres have a thickness of 2 to 3 inches, according to wheel diameter and they have to be replaced when worn to about half that thickness. The writer knows an instance where the running gear of a low-built engine encountered obstacles on the tracks, when the wheel tyres were worn; and in consequence bigger rims had to be shrunk on.



Figl 67.—Wheel Tyre Bearing upon Rail.

Old rails and worn wheel tyres reduce the tractive effort and increase the power consumption. When the rails are worn, they may be reversed, i.e., the inside edge is laid at the outside, as this outer edge generally does not wear. Sometimes "relaying" rails are bought for the sake of economy, but it should be carefully ascertained that these are still in good condition. It might sometimes prove to be a better proposition to buy *re-rolled rails*, which are rails of a heavier section rolled out to a smaller size.

After each crop the factory engineer should take a sheet iron template of the wheel tyres of the locomotives—one for each locomotive will do, as all the tyres have to be turned or replaced at the same time—and he will have a valuable record of the wear on these tyres and how this takes place. When wear is rapidly increasing, he knows that the tyres have to be turned on the lathe or replaced.

Tyres are shrunk on the wheels and when ordering a locomotive a spare set of tyres should also be obtained. Tyres are sometimes liable to burst through crystallization or through too heavy shrinkage. Locomotive manufacturers give the shrinkage measurements, when delivering the locomotive, or if they do not, these data should be asked for. As a general rule the shrinkage measurement should be of such a degree that the unit tension stresses will not be above the allowable limit of about 18,000 lbs./sq. inch.

As the fractional increase of the length e is derived from:

$$e = 1 \div E \times S = S \div E \qquad \ldots\ldots\ldots\ldots\ldots\ldots \quad (21)$$

where E is the modulus of elasticity and S the allowable unit stress mentioned, the resulting shrinkage factor will be at E = 30,000,000 lbs./sq. inch:

$$e = 18{,}000 \div 30{,}000{,}000 = 0{\cdot}0006 \qquad \ldots\ldots\ldots\ldots \quad (21'')$$

As there is a certain surface compression through the micrometric roughness of the wheel and rim material, which allows a certain penetration of the wheel material into the rim material and *vice versa*, the shrinkage allowance should be made larger as the material recedes, and this should amount to between 0·0010 and 0·0014 according to wheel diameters.

The wheel tyre has to be heated to about 150—200°C. to slip easily on the wheel and should be held in exact position by clamps until it has cooled off. For small wheel tyres this heating can be done in a clean open fire of charcoal or coke. For larger tyres a very handy outfit is shown in *Fig.* 68. It consists of a *Perforated Heating Ring* of 1 in. inside diameter, having on the inside of the periphery $\frac{3}{16}$ in. holes at 6 in. pitch or less for smaller diameters. The ends of the ring are held by round iron straps, attached to a piece of flat iron, so that a distance of 6 inches will be maintained for introducing the burner.

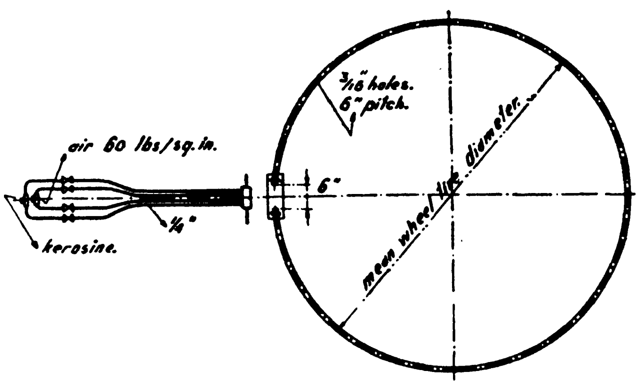


Fig. 68.—Wheel Tyre Heating Equipment.

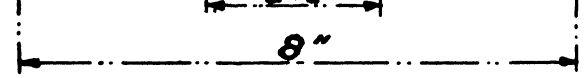


Fig. 69.—Burner Head.

The ring is laid around the rim and is held in place on the wheel tyre with a few flat iron clamps. The burner consists of two adjacently located $\frac{1}{4}$ in. tubes, one set for kerosine and the other for compressed air of about 60 lbs./sq. in. There are four valves in the charging pipes, so as to be able to regulate the amount of fuel and air for good combustion. The apparatus is ignited at the ring, after the burner has been introduced and the fuel and air allowed to pass.

The *Burner Head* is shown in *Fig.* 69 and it should be recollected that the fuel container must have the same pressure as the compressed air, as otherwise the fuel will be forced back. In case a closed receptacle for fuel is not at hand, the burner should be designed as a *suction ejector*, which will aspirate the fuel.

8.—Railroad Itinerary.

It is very useful to prepare a railway itinerary for plantation work, as it will greatly improve the railway performance and consequently the supply of cane at the mill and, as a matter of fact, in many instances reduce the amount of equipment needed. In *Fig* 70 is given a *Graphic Time Table* and as this shows more clearly the train movements than a tabular one, its application will be of great assistance for making up the itinerary. The writer knows a case where the number of locomotives in service has been reduced from seven to four and a better cane supply obtained by merely checking the railway operation by means of a time table. Locomotive engineers, even on a plantation, are rapidly resorting to fixed railway schedules.

As regards the speed of the trains, i.e., the inclination of the lines in the graphic table or diagram, there should be taken into consideration the condition and the grades of the track. In upgrade direction some allowance for reduced speed should be made and in the reverse direction the speed can be increased up to the safe limit the evenness of the track and the curves will permit.

Before making up the itinerary diagram a responsible member of the factory staff should travel several times with the cane trains so as to collect exact information about shunting times, crossings and running times. In the diagram the sidings, water stations, bridges, etc., can be easily entered.

In *Fig.* 70 two different tracks are assumed to exist, one of a length of 25 miles and the other 20 miles long. The left one has two loading stations and the right one five. The locomotives cannot make two trips on the longest branch a day, so they are used on botb. A normal speed of 20 miles an hour

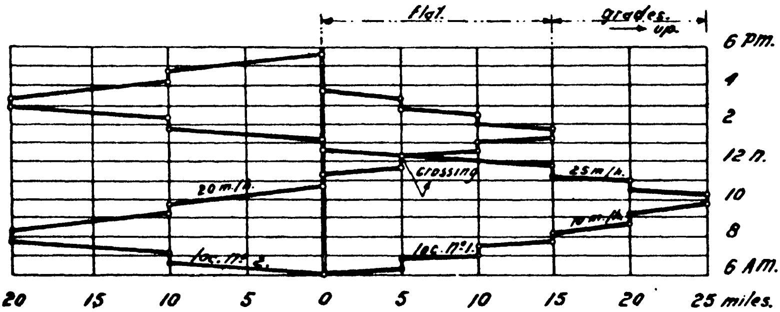


Fig. 70.—Graphic Time Table.

is attained and for upgrade track only 10 miles can be reached, whereas downgrade speed will rise to 25 miles. The regular shunting time at each cane loading station is from 20 minutes to half an hour.

There is only one crossing at 12·15 p.m. and these crossings should be arranged in such a way that they will not hold up trains. Sometimes it pays to have a siding laid down at a convenient spot for this crossing purpose.

9.—Electric Traction.

At some cane sugar factories, cane hauling and/or shunting of the cane cars in the yard is done by electric locomotives. The proposition should be carefully considered, when there is an excess of bagasse fuel and no other additional power consumption is needed, such as for factory or irrigation pumping stations, light and power supply to a nearby village or the public road, etc. But of course a preliminary estimate should be made to show if electric traction will be a paying proposition.

For plantation railways *direct current* will come almost exclusively into consideration, as it will be the simplest form of electric traction. The voltage generally will be low, from 250 to 1500 volts as a maximum. On U.S. and Continental railways direct current is used up to 3000 volts, but the motors on the electric locomotives are in pairs and connected in series, as proper insulation for such high voltages is not an easy task.

As a general rule it can be said that the greater the distance, the higher the voltage has to be, as the main feeder or trolley lines are subject to *Ohm's Law*

$$E = R \times I \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (22)$$

where E is the electric force indicated in volts, R the resistance of the wire or conducts, measured in ohms, and I the intensity of the traversing current, measured in amperes.

For long trolley lines a feeder, parallel with the trolley line, carries the current to the extreme range of operations, as the ohm resistance is proportionately reduced by the increase in current-carrying capacity. The resistance of a conductor is increased by increased temperature and is given generally at 75° Fahrenheit for tropical conditions.

A copper trolley wire 0·410 in. in diameter, or No. 000 according to the Brown & Sharpe wire gauge, has a resistance of 0·0625 ohms per 1000 feet length or 0·339 ohms per mile at 75°F. temperature.

The *current-carrying capacity* of wires in general depends on whether they are insulated or bare and are used for inside or outdoor use. The trolley wire above mentioned will carry 460 amperes; from 3 to 6 amp./mm.^{2} or from 2 to 4 amperes per 0·001 sq. in. being the average. Wires or trolley lines having a larger diameter will allow less current per unit section to be carried than thinner wires.

Sometimes aluminium trolley wires are used and the price per volume, i.e., section multiplied by the length, is lower than that of copper. The specific weight of aluminium is about 30 per cent. of that of copper, but the conductivity for electric current is only 0·625 that of copper, so aluminium section have to be about 1·6 times as large as copper wire sections of the same current-carrying capacity. For trolley wires, therefore, aluminium wires will be about 50 per cent. of the weight of copper wires for the same purpose. The tensile strength of aluminium is about 50 per cent. of that of hard rolled copper wire and therefore the same pole distances can be used.

The electrical current has to travel in a closed circuit if there is to be any flow at all, and this flow has to be from the generator, through the trolley wire to the railcar motor, and thence back to the generator to the negative pole. With electric railways the return current is conducted through the rails and these are not only jointed by the common fish-plates, but have a copper wire or strand connexion as well, so as to ensure positively that a continuous circuit is formed.

There will be a short circuit when the trolley wire and the rail or the earth (as the earth is also a conductor and, when moist, a good one too) are connected by a current conductor. The air is a very poor conductor and an air gap will give good insulation; a distance of 36 in. will give good protection for 10,000 volts. The human body is a good conductor, and when a live wire falls on a man standing uninsulated on the ground or the rail, the current will flow through his body, with more or less disastrous results.

The short circuit is explained through Ohm's Law. When, e.g., the trolley wire mentioned falls on the return rail of 60 lbs./yard size, which has a resistance of 0·049 ohm per mile and the event occurs at a mile distant from the generator, then there will flow a current, when the voltage is 250 volts, amounting to:

$$250 \div (0{\cdot}339 + 0{\cdot}049) = 645 \text{ amperes.}$$

This heavy current will form a lightning arc and melt any metal at the spot touched.

In *Fig.* 71 is shown a 50-*ton D.C. Electric Locomotive* of the 4-4-0 type, having four driving axles and on each a 125 h.p. motor, designed for 1500 volts trolley voltage. The current is taken by a single-wheel trolley from the overhead wire and passes the controller connected with a set of resistances or rheostats, which are placed in series with the motor windings, when starting or stopping. The necessity for these resistances is easily understood when considering the short circuit thesis explained above. They are a protection of the motor windings against very heavy currents.

The total weight of 100,000 lbs. rests on the drivers and a tractive effort of 12,600 lbs. is obtained by the four motors for one hour at a speed of 14·9 miles per hour. The maximum tractive effort, limited by the combined motor power, is 36,400 lbs. which is above the normal rail friction as explained before. The maximum speed is 47 miles per hour. The driving wheels have a diameter of 36 inches and the driving axles receive the rotary impulse from the electric motors by means of reduction gears, having a ratio of 17 to 60 teeth. The motors are connected in pairs, so the voltage for each motor is 750 volts, which will reduce insulating costs. The locomotives can also be arranged for lower voltage down to 600 volts.

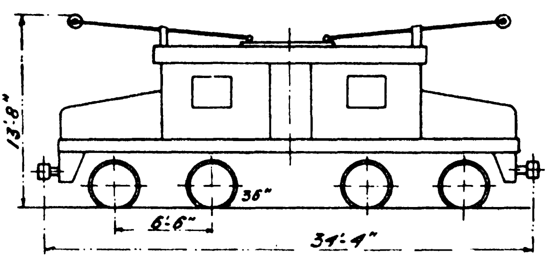


Fig. 71.—50 Tons D.C. Electric Locomotive.

The *Railway Motors*, shown in *Fig.* 72, are of a widely used and fully reliable construction. On the driving car axle is mounted a machine-cut gear wheel, having 60 teeth and two axle bearings, which support a part of the motor weight and torque reaction. The motor pinion drives this gear and the whole is mounted in dust-free casings, the gear wheel casing being partially filled with heavy oil. This oil must not enter the motor windings as oil is an insulator for electric current.

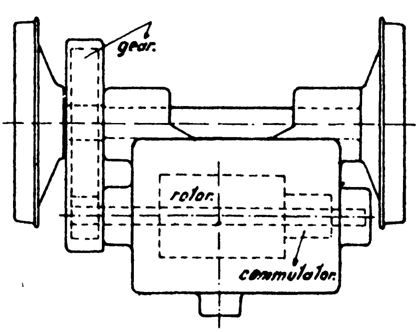


Fig. 72.—D.C. Electric Railway Motor (Plan).

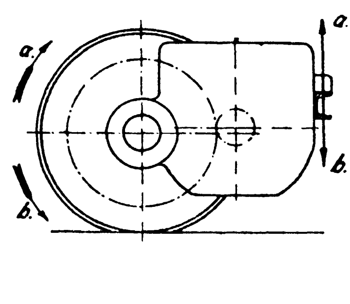


Fig. 73.
Side View of Railway Motor.

In *Fig.* 73 is shown the side view of the motor of *Fig.* 72, and it will be noted that the other weight and torque reaction is taken up by a beam or channel, suspended with springs on both sides of the truck armature. It is obvious that for rotating direction *a* the beam reaction will be upwards, whereas in the reverse direction *b*, the beam reaction will be downward.

Motors of this construction have been used on street-cars, shunting and traction locomotives for public and industrial railways, as well as for mines, under the most varied and severe conditions of operation.

A motor of 75 h.p. has a weight of about 4,700 lbs., including armature, gears and gear case, and therefore the adhesion weight of an electric locomotive has to be raised by cast iron weights attached to the engine frame, or by sealed water or sand tanks.

The motors have commutating poles to reduce sparking and are wound in series, so the same current will pass through the rotor as well as the field coils. These motors have the particular advantage that the starting torque is very high, as the motor resistance is very small when at rest. As already said, the current is braked off by the controller resistances in series with the motor when starting, as otherwise the initial current would be too high and would burn the motor. This will also happen if the motor should stop abruptly with open controller.

As soon as the motor acquires speed, a counter-electric current is produced and the controller resistances can be gradually cut out. When, through increased tractive resistance, the motor slows down, the counter-electric current will decrease and more current will pass through the motor and the torque be correspondingly increased. This capacity makes the D.C. series motor specially adapted for railway performance.

The *trolley wire* should be suspended at such a height that there will be no danger for railwaymen walking on top of the loaded cars or the locomotive. The trolleys are of the regular rod and wheel or bow type. Sometimes, especially for heavy equipment, pantagraphs are used, operated by a spring or pneumatic raising device and carrying one or two contact shoes for taking off the current. For long distances two trolley wires are sometimes used, to reduce the ohm resistance, just as is the case with a separate feeder.

The poles for the trolley wire should be placed at intervals of about 100 feet or more, according to the size of the trolley wire. The longer the distance the greater the *wire sag* and with big changes in temperature the sag will also change. Under tropical conditions the change in temperature will not be so high as for temperate zones with big differences between summer and winter temperature.

A special section of *Trolley Wire*, which can be easily clamped into the trolley suspenders, is shown in *Fig.* 74.

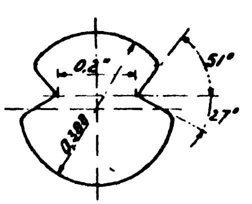


Fig. 74.—Special Trolley Wire.

In heavy electric locomotives a rear axle is sometimes driven by one or two motors and the rotary movement transmitted by *side or coupling rods*. In smaller locomotives the motor frequently acts on one driving axle only and here also side rods are in use.

As there is a great difference in the forces transmitted by coupling rods from a reciprocating prime mover and one of rotary impulse like the gear drive of an electric motor or that of an internal combustion motor or even of geared steam locomotives, this should be explained in a more detailed way.

With a reciprocating steam locomotive, the maximum force on the coupling rods is equal to the maximum piston force, thus :

$$P_{loc.max} = A \times p_{max} \cdots\cdots \quad (23)$$

when P is the rod force, A the piston area, and p the steam pressure. Rods and wheel trunnions can be dimensioned according to this force. At dead centres this maximum force will prevail and generally decrease towards mid-stroke.

With the rotary impulse this condition is completely otherwise. The electric motor is supplying a constant torque movement of the value :

$$M_{rot.} = P_{rod} \times R. \cdots\cdots \quad (24)$$

when R is the crank radius and P the rod force when crank and rod are at right angles. The distance between the rod and the horizontal centre line of the axles will decrease towards the dead centres to the amount of :

$$E = R \times sin\ \alpha \cdots\cdots \quad (25)$$

when α is the angle between crank and horizontal centre line. At dead centre the equation (25) results in :

$$E = R \times sin\ 0° = 0.$$

Computing E for R in (24) we have :

$$M_{rot.} = P_{rod} \times 0, \text{ or}$$
$$P_{rod} = \infty$$

so the rod force will have an infinitive value and in *Fig.* 75 this is shown graphically. If a rod at only one side of the engine were available, the momentum $M_{rot.}$ would break the trunnion or bend the side rod. Now there are generally two rods, the one advanced 90° on the other, so the transmission of forces can take place through the latter, when the former is at dead centre. It should nevertheless not be overlooked, that each rod and trunnion has to transmit the full rod force and not half of it, even if there are two side rods. Breakages of trunnions have occurred with these side rods, which were attributable to the play caused by wear of the main driving boxes and trunnion bearings.

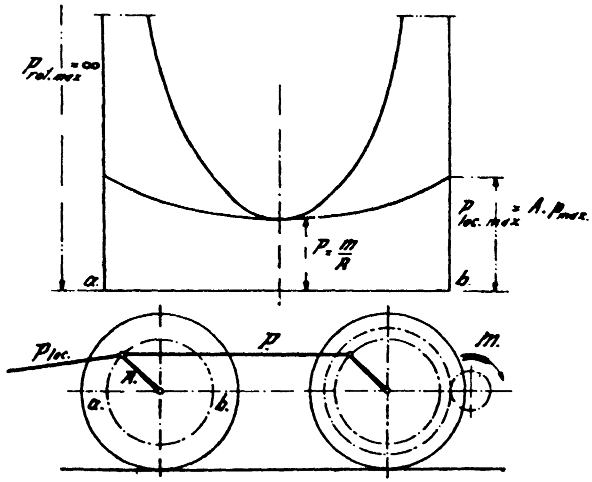


Fig. 75.—Rod Force Diagram.

Individual motor drive on each driving axle has advantages, but is more costly.

10.—Diesel-Electric Traction.

An electric railway system has a stationary power plant and the current is supplied by the overhead trolley wire or sometimes by a third rail to the motors of the moving electric locomotive. To do away with the trolley line and the dangers deriving therefrom, it was a logical step to the *Diesel-electric system of traction*, where the power plant is carried on the locomotive and high speed prime movers, which occupy little space, can be used to advantage and the reliable electric motor drive can be used. As the current has not to pass over a long stretch of trolley line, the voltage can be considerably lower, resulting in less danger, as the motors have a closed circuit on the locomotive itself and no rail or earth connexion is necessary, and wheels, axles, etc., do not carry any current and have not to be insulated.

In *Fig.* 76 is shown a *Diesel Motor Generator Set* of European design as used for standard gauge shunting locomotives. The motor is of the four-cycle solid injection type, having four cylinders of 150 mm. diameter and 185 mm. stroke and developing about 70 h.p. at 1000 r.p.m. The fuel oil is injected at a pressure of about 80 kg./cm.^{2} (1130 lbs./sq. in.) and is brought to ignition by the high temperature caused by an air compression of about 32 kg./cm.^{2} (450 lbs./sq. in.) in a special combustion ante-chamber, located in the cylinder heads.

The driving mechanism is completely enclosed, and forced lubrication and cooling are provided. The motor will start from cold without the use of heating coils or other pre-heating equipment.

The starting is effected by a small electric motor, receiving current from a battery; and the motor pinion is intermeshed with the toothed rim of the flywheel, and cut out as soon as the engine has acquired sufficient speed to keep running. A small dynamo supplies the D.C. current for charging the battery. These high speed engines may be subject to torsion at vibrations, and unbalanced inertia forces will cause unfavourable reactions on the engine frame. The motor and generator are therefore mounted in a frame made of rolled steel sections and suspended or carried by springs, so as to reduce vibrations in the engine armature.

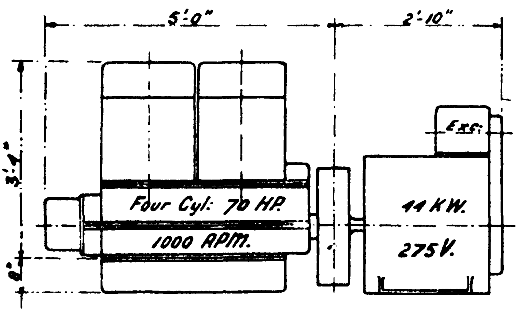


Fig. 76.—D.C. Diesel Railway Generator.

The consumption of Diesel oil on these four-cycle motors is low and will amount, inclusive of the customary margin of 10 per cent., to about 200 grms. per h.p./hour (0·442 lbs./h.p./hr.) and the consumption of lubricating oil is about 4 grms./h.p./hr. (0·009 lbs.). Manufacturers give specifications as to the fuel oil to be used, as well as to the lubricating oil.

The governor is arranged for full and half speed velocities.

The Diesel motor is directly connected to a direct-current generator of 44 kw. for 275 volts. The normal intensity of the current will be 160 amp. but for short intervals of time a considerably higher current is allowable. The generator as well as the electric railway motors are dustproof and carefully ventilated, so as to prevent them from overheating.

The generator is designed in this case so that the current for the field excitation is not taken direct from the rotor current, as is the case with series, parallel or compound wound generators, but there is a separate exciter for this purpose, mounted on top of the generator and driven by a chain drive or gear. The excitation of the field windings, therefore, is completely independent of the generator current, and field current regulation, therefore, is more efficiently obtained.

The *Current Circuit Diagram* of this Diesel-electric drive is shown in *Fig.* 77. The exciter current circuit is provided with a set of speed resistances, which control the excitation of the main field of the generator. The motors are of the series-wound type, as already explained, and each of the two has a normal output of 28 h.p. at 1160 r.p.m., corresponding to a current of 84 amp. but it can actually develop 38 h.p. and withstand a current of 112 amp. for one hour. The motors are geared to the driving axles in the usual manner, as in *Figs.* 72 and 73, and are short-circuited to the generator, thus without the interposition of resistances. All the generated current, therefore, has to be absorbed by the motors, but the generation of the current depends on the field excitation and this is efficiently controlled, as explained above. It is obvious that the resistances for speed are only for the small excitation currents and not for the much heavier main currents.

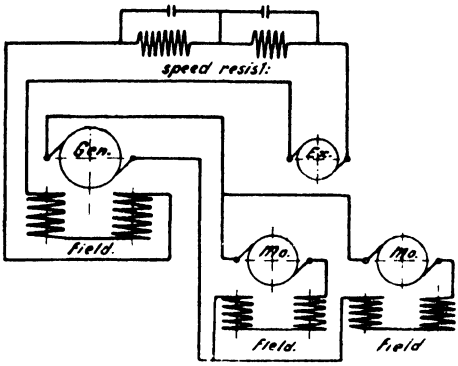


Fig. 77.—Circuit Diagram, Diesel-Electric Locomotive.

The controller, which is connected to the speed resistances, also carries the reversing switches. A maximum automatic switch protects generator, motors and the Diesel engine from overloading.

In *Fig.* 78 is shown a modern 12-*ton Diesel-Electric Locomotive* of British make, as supplied for sugar cane plantations. It has a six-cylinder solid injection Diesel motor, direct coupled to a D.C. generator. On account of the small size, only *one* electric motor is used for driving through a highly efficient worm gear one of the main driving axles. The four driving wheels are 33 in. in diameter and are coupled in pairs by side rods. Electric headlights receive current from a battery, which is also used for starting purposes. The control of the locomotive is very simple and does not require any special skill on the part of the operator. For inspection of the Diesel motor a good mechanic should be available at the mill, as these engines need better mechanical care than the average steam locomotive engine, and to ensure good working performance, unskilled labour should not be allowed to touch them.

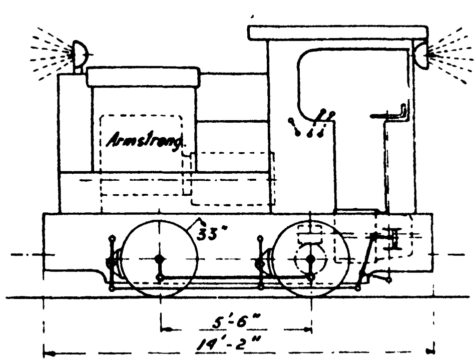


Fig. 78.—12-ton Diesel Electric Locomotive.

The wheel brakes can be operated by hand, as well as by compressed air. A small compressor driven from the Diesel motor shaft or by a small electric motor from the battery will supply the compressed air for the brakes and, when required, for a whistle. The loco is designed for 30 in. gauge and has a short fixed wheel base, so it will take any curve where a cane car can go. The overall width is only 6 ft. 6 in. and the coupling arrangement can be made to suit the car couplings. The fuel oil tank carries 80 Imp. gals., sufficient for a few days' operating performance, and a large radiator is fixed in front of the engine to treat the motor-cooling water. Sometimes a circulating water tank is carried to increase the amount of this water.

The Diesel-electric locomotive does not produce sparks, providing the exhaust pipe is cleaned at regular intervals, and no trouble is caused from smoke. The engine is ready for service at any moment, which is an advantage over the steam locomotive. The *starting torque* of the Diesel locomotive is about 30 per cent. higher as compared with the steam locomotive, which is explained by the high starting torque of the series direct-current motor.

The Diesel-electric locomotives shown in *Fig.* 78 are built in sizes from 12 to upwards of 80 tons adhesion weight.

11.—Geared Locomotives with Internal Combustion Engines.

Before the Diesel-electric drive came into vogue, geared locomotives with internal combustion engines found many an application on sugar cane plantations. There has been a divergence of view between operating engineers as to the reliability of the I.C. motor, but present-day design has advanced greatly and reliable equipment can now be had.

For small traction equipment, the *gasoline or petrol engine* as used on road vehicles with the standard gear box and plate clutch is a reliable piece of equipment, but has a much higher fuel consumption per h.p. than the Diesel engine and, moreover, the fuel price per gallon is also considerably higher.

For heavier locomotives with these I.C. engines, the geared Diesel locomotive has to be mentioned. The gear box is provided with gears, which are constantly in mesh and coupled to the driving or driven shaft by means of air-operated individual plate clutches. These gears have great resistance against wear, as they are all immersed and the clutches are operated from the cabin through compressed air lines. Reversing takes place with the usual bevel gear arrangement.

The newest type of power transmission for vehicles driven by I.C. engines is the *hydraulic coupling*, originally used in ship propulsion. It consists of a bladed impeller and a driven bladed rotor. Oil is pumped into the impeller-rotor circuit and transmits the torque very efficiently. By removing part of the oil from the circuit the number of revolutions of the driven shaft will decrease, but the same torque remains and the efficiency therefore amounts to :

$$e = \frac{n_1}{n_2} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (26)$$

where n_1 is the input r.p.m. and n_2 the output r.p.m.

For car or locomotive drives it is essential to increase the torque when starting, as is the case with the Diesel-electric drive, also with the gear box drive, where the reduction gear causes a torque increase. For high speed engines the torque is very small and the hydraulic coupling is not specially suitable for this purpose.

This has led designers more recently to construct a hydraulic coupling for variable torque. The principle of this *hydraulic torque converter* is the same as for the hydraulic coupling, with the difference that between the impeller and the rotor an interposed guide wheel or set of guides is placed, which changes the kinetic power transmission. The guide wheels can be set at different angles for different torque conversion and the output torque can be increased to *four times* the input torque, and general efficiencies of well over 80 per cent. are obtainable.

To avoid the complicated mechanism needed for changing the guide angle, a combination of both the hydraulic coupling and torque converter with *revolving fixed guide blades* is now made, and an equally good transmitting efficiency has been achieved.

Both types are now being tried out on rail cars and shunting engines driven by Diesel motors, and developments should be followed closely, to ascertain if their application to plantation locomotive engines will be of advantage.

Besides the Diesel engine, the *alcohol engine* should be considered for those plantations where this fuel is manufactured and, therefore, can be obtained at low cost. The alcohol is generally mixed with petrol or gasoline to give a good fuel mixture. For alcohol locomotives the motor-electric, as well as the gear drive, can be considered practicable.

12.—Tractive Effort and Engine Power.

A certain relation has to exist between the engine power and the tractive effort and it is no use having the engine power develop beyond the limit set by the adhesion weight and rail-friction, as this will merely tend to slip the wheels on the rails. On the other hand, the rail friction can be increased by sanding the rail in front of the drivers, but this can be done only for short periods when starting or on heavy grades, and moreover produces an increased wear on wheel rims and rail.

As the tractive effort of any locomotive is used to overcome the train resistance, the general rules for calculating it should be explained, although it may be said at the outset that a theoretical calculation is not possible in detail, and empirical formulae are much used in locomotive calculations.

The only true resistance known is the *grade resistance*. This resistance is equal to a force acting contrary to the train movement when running up grade or *vice versa*, and is composed of a component of the total train weight, i.e., deadweight of cars and locomotive and the load carried. It is obvious that :

$$R_{grade} = W_{total} \times \sin \alpha \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (27)$$

As for the very small angles α of the track inclination with the horizontal, $\sin \alpha$ is equal to $\tan \alpha$, and formula (27) may be read : $R_{grade} = W_{total} \times \tan \alpha$ and as the tangent is the proportion between the vertical elevation and the corresponding horizontal distance, the general rule is derived that *each per cent. of grade (1 in 100 length) will give 1 per cent. grade resistance.*

Public railways on the European continent will not allow grades above 27 per thousand (1 in 37), but on plantation railways they are not confined to such narrow limits and grades up to *5 or 6 per cent.* (say 1 in 20) are common. It will be readily seen that the tractive effort of a locomotive under such conditions is easily taken up by even a small train load, and small driving wheels, big adhesion weight and great speed reduction between engine and driving wheels will be required, and the electric-drive or the geared locomotive with an internal combustion or steam engine is to be preferred.

Secondly, but not so exactly determined, comes the *friction resistance*, and in formula (14) this is given for cane cars, but applies for locomotives as well.

In that formula the car weight is considered, but for more exact calculations the weight of the wheel sets and axles should be considered separately, as these have only rolling friction resistance and no journal friction, and the formula should read thus :

$$R_{friction} = \frac{W_{on\ journals}(x + \mu \times r) + W_{wheels} \times x}{R} \quad \ldots\ldots\ldots\ldots \quad (28)$$

"$W_{on\ journals}$" is the true load bearing on the wheel journals, and $\frac{W_{wheels} \times x}{R}$ is the rolling friction of the wheel sets. The weight of the wheel sets amounts to about 6 per cent. of the load on the journals. Where R and r are different for different cars and locomotives, each group should be calculated accordingly.

For overall calculations $R_{friction}$ can be taken as :

For locomotives .... 3·5 — 4 per thousand total weight
For cane cars .... 2·5 — 3 ,, ,, ,,

The rolling friction of the wheels on the rail is increased when the train is passing along a curve of the track. This increase in friction is equal to 0·4 per thousand total weight per degree curvature. The curvature angle in U.S.A. is expressed in degrees at the centre of the radius, having a periphery of 100 feet, measured on the rail. In Great Britain 66 feet is taken and the curve resistance has to be increased correspondingly.

For locomotives having a large fixed wheel base, the curve resistance might increase to double the amount above-mentioned.

Third comes the *air resistance* and this is a very variable factor, as the wind direction and wind force are equally variable. With a train velocity of 20 miles per hour and a wind velocity of 10 m.p.h. straight against the train movement, there will be a relative wind velocity of 30 miles per hour. The total front area of a standard gauge plantation locomotive, considered as 100 square feet, has an air resistance of

$$R_{wind} = A_{exposed} \times P_{wind} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (29)$$

or in this case $= 100 P_{wind}$.

The wind pressure P depends on the relative wind velocity and from practical tests in England and the U.S.A. it is to be taken as :

$$P_{wind} = 0{\cdot}0032\ V^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (30)$$

where V is the relative wind velocity in miles per hour, and P_{wind} in lbs. per sq. ft.

If the wind be blowing at an angle of 45° against the train movement, then not only is the front area of the locomotive exposed, but the side area of the whole train as well, and moreover the exposed parts of the car fronts through the gaps between the cars, without considering the whirling effect taking place there.

If it is assumed that the locomotive front has 100 sq. ft. exposed area, the 15 cane cars 3000 sq. ft., and also there are 750 sq. ft. of exposed car fronts, then the total air resistance will amount to :

$$R_{wind} = 3850 \times P_{wind} \times \sin 45° \simeq 2{,}700\ P_{wind}$$

which means that this resistance is 27 times as great as in the previous assumption that the locomotive front only has been exposed.

To make the calculation still more complicated, it should be recollected that the wind generally is not blowing parallel with the earth surface, but at an angle of about 10° towards it, and the top area of locomotive and cars also will be exposed.

When the track also has curves, it will be obvious that a theoretical calculation has no value whatsoever, but it will be readily seen that the wind might cause a considerable resistance, which at times will take up to 25 per cent. of the total tractive effort of the locomotive in stormy weather, and 5 per cent should be taken as the minimum under favourable weather conditions.

The fourth factor to be considered is the *acceleration resistance* as the train speed has to be increased from zero to the required rate and a force is necessary to overcome the inertia of the train weight. This force depends on the distance within which the required speed has to be reached and will be higher the shorter this distance or the equivalent time period. The formula for the acceleration resistance is written :

$$\frac{M \times v^2}{2} = R_{acceleration} \times D \quad \ldots\ldots\ldots\ldots \quad (31)$$

where $M = \frac{W_{total}}{32}$ is the mass of the total train weight, v the speed in feet per second to be attained and D the distance covered in feet, when the speed v has been reached.

As $v = \frac{5280\ V}{3600} = 1{\cdot}46\ V$, where V is the train speed in miles per hour, the acceleration resistance per 1000 lbs. total train weight will amount to :

$$R_{acceleration} = \frac{1000 \times (1{\cdot}46\ V)^2}{2 \times 32} \div D = 33{\cdot}3\ V^2 \div D \quad \ldots\ldots \quad (32)$$

For the metric system this formula is written :

$$R_{acceleration} = 3{\cdot}93\ V^2 \div D = 4\ V^2 \div D \quad \ldots\ldots\ldots\ldots \quad (32a)$$

derived in the same way per 1000 kg., V in km. per hour and D in metres.

It is readily seen that the longer the distance D, the smaller the acceleration resistance. Furthermore, it is obvious that when a train starts, the starting friction is high, as the oil has been pressed out between bearings and journals and a new oil film has yet to be formed, while the wheel material has penetrated more into the rail material than happens when running. Nevertheless, the air resistance is small, as well as the acceleration. As the train speeds up, the friction resistance decreases, whereas the air resistance and acceleration increase and there will come a moment when the tractive effort of the locomotive is in balance with the total resistance of the train and, therefore, the moment of inertness has arrived.

The total train resistance therefore is composed of :

$$R_{total} = R_{grade} + R_{friction} + R_{air} + R_{acceleration} \quad \ldots\ldots\ldots \quad (33)$$

On horizontal track the starting resistance will amount to 7 to 12 per thousand of the total train weight ; and at a running speed of 20 miles per hour, the resistance is about 6 per thousand, which may be considered as a fair mean for plantation railway operation.

As the product of tractive force and speed is equal to the engine power, the formula can be written thus :

$$HP = \frac{TE \times v}{550}$$

where TE is the tractive effort, v the speed of the train in feet per second, and 1 HP = 550 ft. lbs. per second.

As derived from formula (32), $v = 1{\cdot}46\ V$, where V is the train speed in miles per hour, and the formula now becomes :

$$HP = \frac{TE \times V}{375} \quad \ldots\ldots\ldots\ldots \quad (34)$$

where HP is the tractive horse power or power output at the wheels.

For metric conditions, where v is given in metres per second, 1 metric h.p. = 75 kg.m. per second and V in km. per hour, the formulae above mentioned are written thus :

$$HP = \frac{TE \times v}{75} \qquad v = V \div 3{\cdot}6$$

and the general formula will be :

$$Metric\ HP = \frac{TE \times V}{270} \quad \ldots\ldots\ldots\ldots \quad (35)$$

As the engine mechanism has a mechanical efficiency η from 0·8 to 0·9, this has to be added in formulae (34) and (35) :

$$\underset{\text{(engine)}}{BHP} = \frac{TE \times V}{375 \times \eta} \qquad \underset{\text{(engine)}}{Metric\ BHP} = \frac{TE \times V}{270 \times \eta}$$

TE is given in pounds for the British system and in kg. for the metric one.

The engine power is transferred to the periphery of the drivers on the rail and therefore the adhesion weight of the locomotive has to be of sufficient size to transmit the engine power, thus :

$$W_{adhesion} \times F_{rail} = TE = \frac{HP \times 375 \times \eta}{V} \quad \ldots\ldots\ldots\ldots \quad (36)$$

where F_{rail} is the rail friction, which can be taken as 0·36 for starting on dry rail, whereas for running under all weather conditions a tractive effort of 18 per cent. of the adhesion weight should be considered a normal figure. For the metric system the multiplier 375 has to be changed to 270. It will be obvious that the *engine tractive effort is equal to the peripheral force of the drivers.* This peripheral force is derived from the cylinder dimensions and steam pressure in the following way for a two cylinder locomotive :

d = cylinder diameter.
S = engine stroke.
r = crank radius (half the stroke).
P_m = mean steam pressure.
η = mechanical efficiency.
D = diameter of the drivers.
R = radius of the drivers.

$$TE = \frac{\frac{\pi d^2}{4} \times P_m \times r \times 2 \times \eta}{R \times \pi} \times 2 \quad \ldots\ldots\ldots \quad (37)$$

The mean steam pressure is found from the indicator diagram and depends on the steam admission or cut-off of the cylinders but the boiler pressure P is generally given, and when $P_m \times \eta = 0{\cdot}6\ P$ the formula (37) is simplified to :

$$TE = \frac{d^2 \times 0{\cdot}6\ P \times S}{D} \quad \ldots\ldots\ldots\ldots \quad (37a)$$

and applies to a locomotive with twin cylinders. It is obvious that $\frac{r}{S} = \frac{R}{D} = \frac{1}{2}.$

For American conditions the multiplier 0·6 is taken as 0·85 and, therefore, an American locomotive of the same dimensions as a European one will be specified with about 40 *per cent. more tractive effort.*

Before concluding the different sections dealing with rail traction, a few words should be said about derailments. When cars or engines are overturned, a heavy long beam crane, mounted on rails, the salvage or break-down crane, driven by a steam or oil engine, will replace the derailed equipment very quickly on its wheels. The capacity of this crane has accordingly to be equal to the heaviest engine weight, and the writer knows an instance where a crane of 80 tons hoisting capacity was available for standard gauge equipment. Where such a crane is not available, a wooden structure with overhead beam and two guyed supports of heavy timber should be erected at the spot and chain hoists be used for lifting.

Where the cars or engine are still on their wheels, but off the track, car replacers or ramps can be used to advantage, and these should be carried on every locomotive in actual service. A few powerful jacks should also be carried. Hydraulic jacks have the inconvenience that when the leather cups break, the load will tumble down again, as these jacks cannot be stopped at any position when the hydraulic packing leaks.

To close traffic on existing tracks, *derailers* are used, which can be obtained from the manufacturers.

13.—Traction Engines and Tractors.

For trackless transportation in the fields, the traction engine or tractor is used in many cases, as its operation is less troublesome than the laying of portable track, and loading of carts can be done anywhere in the cane fields.

The *steam traction engine* is the oldest type and of very reliable operating performance, but, due to their relatively heavy weight, these engines should be used on roads and are not very well fitted for operating within the canefields. Where sufficient parallel roads of a certain soil firmness exist, they can be used to advantage but they have to be provided with spark arresters as the danger of cane fires is not imaginary, especially when soft fuel, like wood, is used.

The standard steam traction engine is built with single or compound cylinders for working pressures from 140 to 160 lbs. per square inch. The power output is from 15 to 28 h.p. and the total weight is from 20,000 to 32,000 lbs. in working order. A speed from 2 to 4 miles per hour is attained, thus not far in excess of ox traction. On hard roads these traction engines will pull a load of from 45,000 to 100,000 lbs., including the carts, at low speed. The driving wheels are about 6 ft. diameter and 18 in. wide, having flat steel slats riveted or welded on the wheel rims. The steering is easy and sometimes an engine-operated drum with about 75 feet of winding rope is attached to the engine frame, so as to be able to pull carts from a distance.

Of more recent use is the *Caterpillar Tractor* driven by a petrol or gasoline engine as used on road trucks with the corresponding gearbox and rear axle drive. As the fuel consumption of these engines is about 300 grms. per b.h.p. (0·65 lbs.) and the price of fuel is high, the Diesel engine has made its entrance into this special field, having only about 200 grms. (0·44 lbs.) fuel consumption per b.h.p. As Diesel oil is considerably cheaper than petrol or gasoline, the fuel cost is reduced from about one-quarter to one-sixth of the other. The Diesel engine has to work on the cold starting system and be of the airless type, i.e., with direct or solid injection of the fuel into the cylinders. Four or six cylinders

5

are to be preferred, as the two-cylinder engine has a resultant engine couple of forces, which causes detrimental vibrations. The number of revolutions of the Diesel engine is generally less than that of the petrol engine, and a step-up gear is applied in front of a light gear box, in order to take up the heavier torque. The steering of the caterpillar tractor is accomplished by stopping one caterpillar belt and operating the other. There is, of course, heavy slipping effect and this operation should be done carefully when on soft soil or within the canefields.

The caterpillar belt is different from the one for cane carts shown in *Fig.* 24, as the endless belt has to transfer the tractive effort and therefore has to be of stronger construction. In *Fig.* 79 the general design of such a caterpillar belt is shown. The rear axle is driven through the gear box with a helicoidal gear and both axles are fixed to the tractor frame. The rear wheels have teeth which intermesh with the chain openings. The heavy chains are of the roller type and have overlapping steel slats provided with riveted slats of angle iron to cause a better grip on the soil. The bearing surface on the soil is large and little penetration is to be expected.

The normal Diesel tractors are powered from 35 to 60 h.p. and a normal speed from 4 to 6 miles per hour can be achieved on good country roads. The operator should have the necessary instructions how to handle the equipment and should have a fair mechanical knowledge. The exhaust is discharged on top of the tractor and the exhaust pipe should be cleaned from time to time as soot will cause sparks. Soot formation will only take place when improper combustion occurs, caused by fuel atomizer defects, such as after-dripping, and carbonized lubricating oil from the cylinders.

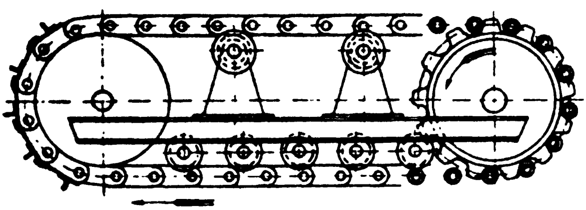


Fig. 79.—Tractor Caterpillar.

14.—Cane Loading Devices.

For handling cane on the fields and at the loading stations or factory yard, different appliances are used. In *Fig.* 80 a *Motor-driven Cane Loader* mounted on caterpillars is shown. This apparatus can be used in the cane fields for loading the cut cane direct on the cane carts or railway cars. The cane is generally tied in bundles for this type of cane loader, and this is done with chain slings which have a self-tightening device, so that the bundle can be kept intact from leaving the field until delivered at the factory cane carrier. The crane boom has a length of about 30 feet and will handle a load of about 10,000 lbs. cane. The bundle normally does not weigh over 7,500 lbs. The caterpillar will take any unevenness of the cane fields like furrows and small ditches.

The hoisting engine, with drums for hoisting and luffing, is mounted on a turntable, which swings around to a full circle with the boom.

In *Fig.* 81 is shown a rail car with a *Steam Cane Loader*, equipped with a 30 ft. boom made of 6 in. seamless tube, as well as the derrick supports. This loader is especially used where there are no fixed loading stations and the cane is transferred from the field carts into the railway cars. It is convenient to have the cane loader situated on a siding, parallel to the main track, so that the cane train can be moved along on this main track. The cane loaders on rail when, as is here the case, not equipped with a balanced turntable, have small hoisting capacity, when the boom is fully sideways and therefore the derrick of *Fig.* 81 is provided with a pair of detachable jacks, which support the front part of the car against overhanging. The jacks have foot plates, and wooden ties are laid alongside the track to increase the bearing surface. Sometimes the jacks can be placed in such a way that the railway ties will serve for the purpose. The slewing of the boom is obtained by a set of pulling ropes, operated by hand, but a tiller attached to the boom turntable and operated by the steam winch may be used to advantage.

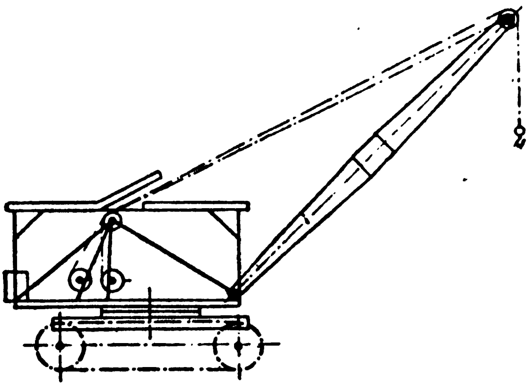


Fig. 80.—Motor Driven Cane Loader.

A rope speed of up to 150 ft. per minute on the drums is usual. The engine is generally a twin with two cylinders of 8 in. × 12 in. The boiler is about 4 ft. 6 in. in diameter and has a height of 10 ft. For the hoisting and luffing operation two drums are at hand, provided with the customary clutches and brakes, so the engine rotates in the one direction and need not be reversible. For the tiller operation a double set of drums is applied, one having right-hand wound rope and the other left-hand. The hoisting speed is not over 75 ft.

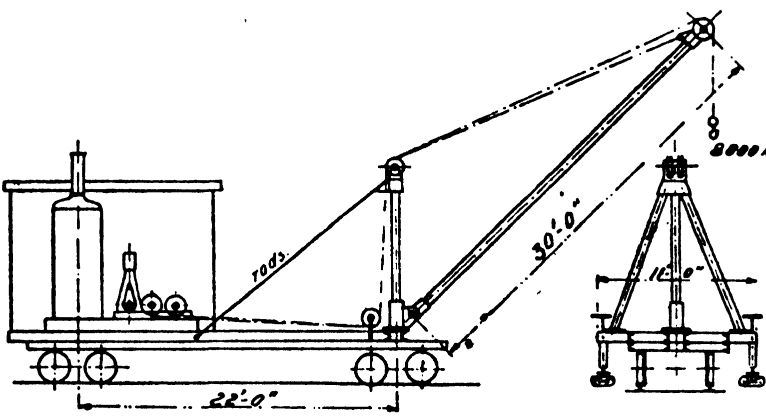


Fig. 81.—Steam Cane Loader.

per minute, as generally a loose pulley is mounted in the hoisting and luffing rigging.

Where loose cane has to be handled efficiently, a *cane grab* should be used, as it will speed up the loading time. The simplest form is the one-line grab, which can be attached to any hook from a derrick or cane loader. This one-line cane grab has a spring locking device operated by a rope-pulled lever. When the grab is loading the locking device is closed and by lifting will close the grab. When the load has arrived at the dumping spot, the lock rope is pulled and the grab opens by gravity of the cane and grab material.

For fully controlled operation from the operator's platform, the *Two-Line Grab* is used and a diagrammatic sketch of its working is shown in *Fig.* 82.

The cane is grabbed by the tines, which are placed in rows all revolving around the same centre axis. This axis is firmly attached to the pulley *c*, whereas the combined tines are connected to the grab bars, held by the rope *a*. When this hoisting line *a* is held firmly and the rope *b* is pulled, the pulley *c* will be raised and the grab closes. The cane within the grab is now lifted and the speed of rope *a* has to correspond with the speed of rope *b*. To avoid slackening of the rope *a*, an equalizing counterweight is mounted within its rigging. When the grab has arrived at the unloading site, rope *a* is held tight and rope *b* is

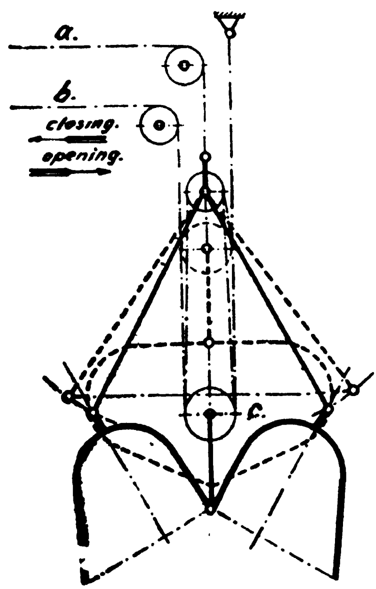


Fig. 82.—Two Line Grab.

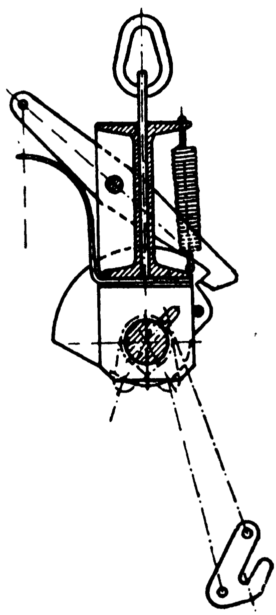


Fig. 83.—Spreader and Trip.

paid out, so that the pulley *c* will be lowered and, assisted by the cane weight, the grab will open. Friction shock-absorbers are provided for smooth operation.

The rope lines *a* and *b* are subject to the same hoisting load, when the same number of pulleys is used in each rigging. It is obvious that line *b* can be split up into two lines for wide grabs and the three-line grab is the result. When line *a* also is split up, a four-line grab is created.

To be able to sling chains around loose bundles of cane from carts or cars, the car floors should have a few transverse strips of wood, on which the cane can rest and a space underneath is thus left where the chains can be pulled through by means of hooked iron rods. For a certain length of cane bundle, four to six chains of about 18 ft. length and ½ in. link iron diameter, will be needed. These chains are attached to a *Spreader Beam*, as shown in *Fig.* 83, constructed of two upright channels riveted together and carrying underneath a well-supported shaft of about 3 in. diameter. This shaft has a spring locking device at one end outside the channels. On the shaft are hung the links of the fixed chain ends, whereas the tripping device has two chains, one attached to the shaft and the other to the eye pin, which is shown in an inclined position in the drawing. When the cane bundle is hoisted, the load is carried by the left hand fixed chain and by the right hand tripping chain, the shaft being locked. When the lock rope is pulled, the load will be transferred from the outside tripping chain to the inside one, but this will cause the tripping link to turn over and disengage the link of the sling around the cane bundle, and thus drop the load.

It should be mentioned that the disengaging chains suffer a violent jerk and operators should keep clear of them.

Fig. 83 also shows a spring arrangement for automatically closing the locking device when the cane has been dropped.

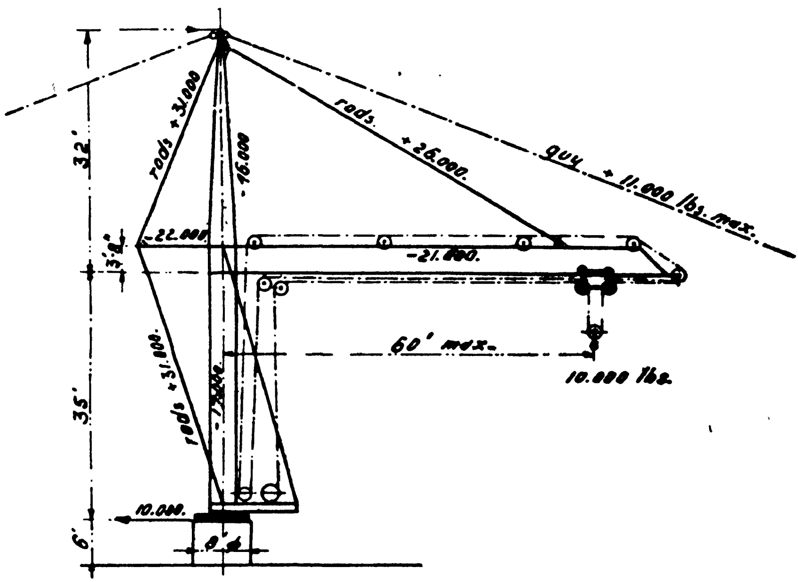


Fig. 84.—Cane Storage Crane.

Where cane has to be stored, a *Cane Storage Crane*, as shown in *Fig.* 84, is used in many cane sugar countries. The crane has a working radius of 60 ft. and the cane can be stored to about 25 ft. high. The boom radius is within reach of the cane carrier to the mills and a considerable amount of cane can be stored in the circular space. The hoisting capacity is 10,000 lbs. and the operating winches are located on a platform at the bottom of the vertical mast and will rotate with it. The vertical mast is well supported on the main bearing on the foundation, and at the top a bearing is arranged, attached to four cable guys. These guys have to keep clear of the swinging boom. The winches can be driven either by steam, combustion motor or electricity, as the case may require.

The stresses in the structure and guys, caused by the load alone, thus without considering the tare weight of the crane, are indicated in the drawing, the tension stresses being marked with the positive sign and the compression stresses with the negative sign.

By slewing the crane, the acceleration force has to be transmitted by the square mast to the boom, so this part of the structure is subject to a considerable torque, as is the braking of the slewing action, and diagonal bracings are applied by the manufacturers. A tubular construction of this lower mast section could be used to advantage, as it is suitable for withstanding heavy torque when properly reinforced by rings of angle or T-iron.

For *bundled cane*, which occupies less space than loose cane, the storage capacities for different sizes of cranes amount to :

60 feet arm	....	4,000 short tons
80 ,,	....	8,000 ,,
100 ,,	....	14,000 ,,

The storage crane will materially reduce the amount of transportation equipment, so far as this refers to carts and cars, and night storage of the cane within the cars is not needed. In countries where bundled cane is transported, this will prove advantageous.

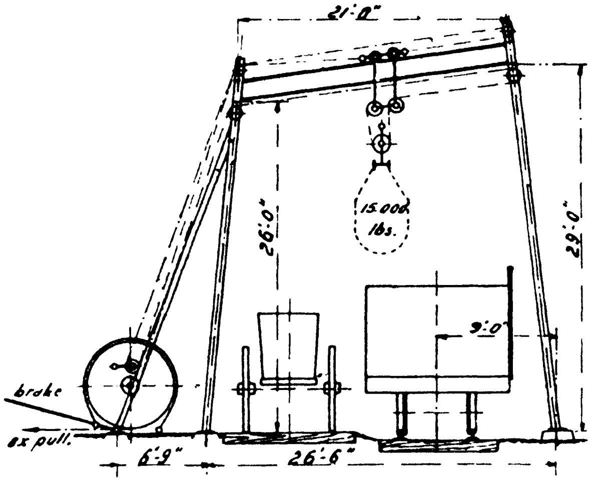


Fig. 85.—Cane Hoist.

A *Cane Hoist*, as used in Cuba, is shown in *Fig.* 85. It is used for transferring the cane from the ox carts to the railway cars. The uprights are of a braced construction and placed in an inclined position. The cross beam, 15 to 18 inches high, is placed with a slope towards the hoisting end, so that the empty trolley will return more easily to above the cane cart. The hoisting is done by a loose pulley arrangement, and the hoisting cable or rope is attached to a drum of 16 in. diameter by 9 in. wide. This drum is mounted on the same axis with the hoisting wheel of 6 ft. 9 in. dia.; the latter is made from inverted 8 in. channel, bent in the form of the periphery of a circle. The cable wound around this hoisting wheel is pulled by a team of oxen, and the hoisting load, L_{hoist}, is derived from :

$$L_{hoist} = \frac{P_{oxen} \times 6{\cdot}75 \times 3 \times \eta}{1{\cdot}33} = 11{\cdot}5 \, P_{oxen} \quad \ldots . \ (38)$$

when the mechanical efficiency η is taken as 0·75 through the rather small rope pulleys.

With three yoke of oxen and a maximum pulling force of 250 lbs. per animal, a load of 6 × 250 × 11·5 = 17,000 lbs. maximum can be hoisted. The maximum cart load will seldom reach 500 to 600 arrobas (25 Spanish lbs. or 11·5 kg. each).[1]

For traversing, a small hand-operated double drum is applied, having the ropes wound in reverse direction and arrested by a hand brake or ratchet wheel.

[1] 1 Spanish lb. = 1·015 lbs. avoird.

The big driving wheel has a brake attached to the side of the rim, by means of which the load can be held at any position.

These cane hoists are sometimes provided with a net weight weighing mechanism, but this has to be arranged in such a way that no rope reactions will act on it, as this would affect the true cane weight.

In several countries use is made of *aerial cable ways*, especially where ravines, rivers etc., have to be crossed, or broken country makes road transportation difficult. For storage of cane these can also be used to advantage and the straight or radial travelling cable ways can cover a considerable area.

The span of a cable way should not be less than 300 feet and the maximum in use is well over 2000 feet. The loads handled for sugar plantation work are not over 10,000 lbs., and considerable speeds can be attained in hoisting and transporting. On a 1660 ft. cable way of the two-line travelling type, 22 tons of cane has been handled per hour.

The one-line cable way has one rope for transporting and carrying. It is constructed with an endless rope on horizontal pulleys, but supports have to be placed at regular distances, on which the cane cradles can be hung. For broken country, the location of the intermediate supports may cause inconvenience.

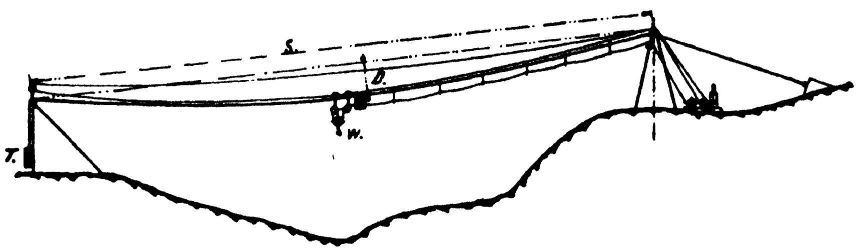


Fig. 86.—Cableway.

The two-line cable way is better adapted for this purpose and a heavy rope is used as a carrying rope alone, on which moves a trolley, whereas the transporting and hoisting are effected by separate ropes. The carrying rope should be of a closed slightly twisted construction, so that it will have a smooth surface on which the trolley wheels operate. These cables are made for ultimate stresses up to over 300 tons and the capacity of this kind of cable way is almost unlimited. There are no intermediate supports, but only the two terminals.

In *Fig.* 86 is shown such a *Cable Way*, in which the driving is effected by a double drum steam winch. Electric operation or the use of an internal combustion engine is also feasible.

The carrying cable has a stress considered equal to the counterweight T, although in practice it will be a little in excess owing to the friction of the carrying pulley. The deflection D at the centre has a close relation to the stress T, and the stress T will decrease as the deflection D increases.

The hauling cable is supported by rope carriers to avoid the slack and inertia forces arising therefrom. These rope carriers are spaced through a special guide rope carrying buttons of perfected construction at the desired intervals.

The cane can be hoisted in bundles, using the already-mentioned trip gear.

In *Fig.* 87 is shown the *Diagram of Forces* for a cable way. The carrying cable is supported at the extremes, with a span S. The load W is causing a stress in the carrying rope, as is, moreover, the actual weight of the rope. In the figure it is taken that the load pulls the cable in a straight line, although in practice through the actual weight of the cable the line will not be straight but partake of a parabolic curve.

The stress T can be calculated from the parallelogram as :

$$\frac{W}{2} = T_{load} \times \sin \alpha \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (39)$$

As the deflecting angle α is small, sin α may be assumed equal to tan α and in good practice the deflection D is taken as 5 per cent. of the span S, thus :

$$\tan \alpha = \frac{D}{\frac{S}{2}} = \frac{0{\cdot}05\,S}{0{\cdot}50\,S} = 0{\cdot}10$$

which gives the angle α as about 6°, and :

$$T_{load} = \frac{W}{2 \sin \alpha} = \frac{W}{0{\cdot}2} = 5\,W \quad \ldots\ldots\ldots\ldots\ldots \quad (40)$$

When the deflection is less, e.g., 3 per cent., the load stress will amount to 8·3 W, and when only light cables are at hand or the load has to be increased a larger deflection should be given.

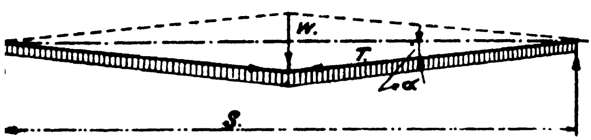


Fig. 87.—Diagram of Forces for Cableway.

The tare weight of the cable or rope has to be considered too, and is taken as : $W_{tare} = S \times w$, when S is the span in feet and w the weight of the rope per foot. This formula is not entirely correct, as the true length of the carrying rope is $\frac{S}{\cos \alpha}$, but with the small deflecting angles of 5 per cent. prevailing, this will only amount to about 1 per cent., and can be neglected. The proper weight of the rope is not concentrated at the mid-centre of the span, but is uniformly distributed over the whole length, which is equal to a concentrated load of $\frac{S \times w}{2}$ at the centre spot.

The stress caused by the actual rope weight therefore works out under the above-mentioned condition of 5 per cent. deflection as :

$$T_{tare} = \frac{S \times w \div 2}{0{\cdot}2} = 2{\cdot}5\,S \times w \quad \ldots\ldots\ldots\ldots\ldots \quad (41)$$

The total cable stress is the sum of the load stress and the tare weight stress, and for a 5 per cent. deflection the rule applies :

The carrying rope stress is equal to five times the concentrated load W plus 2·5 times the tare weight of the rope.

As $\tan \alpha = \dfrac{D}{S \div 2}$ and $\sin \alpha$ is assumed to be equal, formulae (40) and (41) may be written :

$$T_{load} = \frac{W}{2\dfrac{D}{S \div 2}} = \frac{W \times S}{4\ D} \text{ and } T_{tare} = \frac{S \times w \div 2}{\dfrac{2\ D}{S \div 2}} = \frac{w \times S^2}{8\ D}$$

from which follows :

$$T = \frac{W \times S}{4\ D} + \frac{w \times S^2}{8\ D} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (42)$$

and:

$$D = \frac{W \times S}{4\ T} + \frac{w \times S^2}{8\ T} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (43)$$

The pulleys should be made in such a way that the rope is carried on the lower half of its periphery without binding on the sides, and the grooves therefore should have a groove diameter $\frac{1}{32}$ in. in excess of the rope or cable diameter. When the grooves are too wide, the cable will flatten.

The fleet angle, i.e. the deviation from the centre line between mid-pulley and middle of drum, should not be over 2° or otherwise special pulleys or guide wheels should be used.

Cable splices have to be made carefully by skilled workmen, and possess a length about 300 times the diameter as a minimum. A good splice is as strong as the cable and does not alter the outside diameter of the rope.

For connecting the cables to the tension weight by carrying ropes or to the drums for transporting ropes, use can be made of a thimble, where the rope is slung around, forming an attaching eye. The loose end should have a length of about 40 times the diameter of the rope and instead of splicing, cable clips should be preferred, as they are easily attached and give about 80 per cent. of the cable strength, when applied in sufficient number and in such a way that all receive a part of the load. According to the cable diameter, four to eight clips have to be used.

A funnel-shaped socket can also be used. The cable wires are in that case untied and after dipping into a muriatic acid bath the funnel is filled with molten zinc, so-called *spelter*. As this zinc will splash, workmen should protect eyes and hands when pouring the metal.

Clips and sockets should not be used too close to the pulleys as high bending stresses will occur. Corrosion should be prevented by applying a protective coat. The socket has been less favourably accepted, as some of the muriatic acid will penetrate to the hemp core or between consecutive wires, this having a detrimental effect on the life of this part of the cable.

Long hoisting ropes may be mounted in layers on top of each other on the hoisting drum. The cable-carrying capacity of a drum can be ascertained as follows :

Let D be the drum diameter and F its flange diameter, while L is the drum length between these flanges, then the volume of the hollow cylinder, where the rope is wound, will obviously be :

$$C = 0{\cdot}785\ (F^2 - D^2) \times L$$

in cubic inches.

The volume occupied by the cable has to be calculated as though it were of *square* and not of round section ; thus each linear foot of cable has a volume of $d^2 \times 12$ in cubic inches, when d is the cable diameter. The quotient of both will give the length of the cable that can be wound on the drum, thus :

$$C_{feet} = \frac{0{\cdot}785\,(F^2 - D^2) \times L}{12\,d^2} = \frac{0{\cdot}065\,(F^2 - D^2) \times L}{d^2} \quad .. \quad (44)$$

In Hawaii a unique method of cane transportation is used. It consists in floating the cane to the mill in V-shaped canals, called *flumes*. The primary condition for this kind of transport is a considerable amount of water available well above factory level, as about 150 to 200 times the weight of cane is needed in water.

The cane has, through the difference in level, sufficient gravity energy available to be transported on an inclined cableway, e.g. without any other driving power input and the water is only used as a floating medium and a very low mechanical efficiency is the result. Although the unlimited supplies of high level water available in the Hawaiian Islands and in some other countries where a more profitable use is not feasible may justify flume transport, this method has not been adopted elsewhere.

15.—Cane Weighing Equipment.

Most of the cane transported to the mills is weighed on *Platform Weighing Scales*, where the rail car or ox cart is weighed together with the cane. This system has the inherent inconvenience that the tare weight, previously ascertained, does not remain constant, particularly so with ox carts where a varying amount of mud is carried between the wheel spokes and rims.

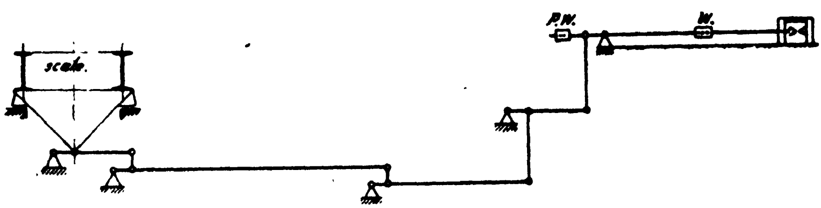


Fig. 88.—Scale Diagram.

Rail cars can be weighed on the train, i.e., when coupled together. When they have to be weighed separately, the weighing platform has to be of such dimensions that it can hold a complete car.

On the other hand, the total train weight is composed of the total wheel loads, so if these are located successively on the weighing platform, the resultant total weight has to be the train weight.

From *Fig.* 88 it can be seen that a very large reduction operates through the weighing levers, to as much as one pound of the sliding weight W on the weighing arm for every ten thousand pounds on the track platform. It is obvious that a vertical movement of one inch on the end of the weighing arm corresponds with only one ten-thousandth of an inch rise of the platform. This explains why coupled cars can be weighed, as the vertical movement is of micrometric order and too small to cause any vertical sliding between the couplers.

There are scales with through-going track, so that the rails are not interrupted at the intersection of the weighing platform, but this needs very careful adjustment of the lever supports and such equipment is practically not used on sugar plantations in tropical countries where heat expansion will play its part.

The weighing platform rests directly on the weigh levers and its knives, as the rise of the platform is too small to apply a bearing support when not weighing. The levers and the knife supports, therefore, are of heavy construction to resist the vibration when a car is placed upon the platform.

Sideward swinging of the scale is prevented by chains, which are fastened to the platform and the foundation. Due to the very small lift, there is no interference with the true weight. On the other hand, any additional weight on the last levers is particularly sensitive and will cause an error in weight. Dirt, therefore, should not be allowed to settle on the mechanism, and living creatures like frogs or lizards should not have their home in the scale pit, where frequent cleaning underneath is essential for good scale performance.

Foundations have to be laid very carefully, and when possible this should be carried out to the instructions of the manufacturer of the weighing equipment. Regular inspection by the factory expert or some competent person should take place.

The *no load balance* should be frequently ascertained, and correction is possible by the equalizing weight *PW* of *Fig.* 88, which is mounted on a threaded spindle for adjustment.

Automatic cane weighing has not yet found application, owing to the rough handling the delicate parts of these scales receive and the errors arising therefrom.

As the cane weight is not only essential for commercial reasons, as with the *colono* system where the planter is paid accordingly, but also for the manufacturing end, many a "stunt" has been achieved by the laboratory staff, where the real cause lay in unchecked cane weights.

The maintenance of the scales includes the proper care of knives and bearings, which have to be free of dirt and well protected. Cleaning the knife surfaces with compressed air is convenient, and they should be sparingly lubricated, if only for protection from rust.

But too much grease might accumulate dirt and is to be avoided. A further protection is to have the scale platform covered with dove-tailed boards to prevent dust falling through.

In the fields the cane is weighed when transferred from the ox carts to the rail cars, to provide a check on the mill scales. Where the cane is hoisted in bundles or on a spreader, these bundles can conveniently be weighed on a net weight scale attached to the cane hoist. As mentioned before, these net weight scales should not receive any stresses from the rope or cable pull as it will interfere with the true weight.

CHAPTER III.

FACTORY BUILDINGS.

Building Arrangements—Hazards from the Elements—Details of Steel Buildings—Details of Wooden Buildings—Travelling Cranes—Platforms—Foundations.

The machinery and apparatus of a sugar factory should be located within a building or a set of buildings, as otherwise atmospheric influences will have a detrimental effect on them, especially so in tropical countries. The mill proper is sometimes, in small factories, placed in the open air, but this is not to be considered good practice. The operators of the equipment should equally be protected from outdoor conditions.

For tropical cane sugar factories, a steel or wooden building frame is generally used, the roofs being covered with corrugated sheets of galvanized iron or asbestos cement. The walls are of brick or wooden boards, or else of corrugated material like the roofs.

Steel buildings will suffer from corrosion when not properly and regularly painted, and it should be recollected that corrosion in tropical countries takes place at a much higher rate than in temperate zones. This applies to a considerable extent to column bases and to the roofs, where the effect of moisture and heat is greatest.

Wooden buildings are attacked by insect pests, such as termites or white ants, but fortunately there are many tropical hardwoods which are not affected by this plague. The steel building has nevertheless a better fire protection, but the fire risk is determined more by what is kept in the building than by the building material itself. Where such inflammable material as sugar is carried, additional care has to be taken in this respect. Bagasse also is inflammable and when piled in big heaps with a large moisture content, spontaneous combustion can take place, caused by the fermenting chemical action of the decomposing fibrous material.

Brick walls are usually expensive, but they offer greater resistance to deterioration and are to be considered as fire-proof. They should be built of well shaped bricks, laid in a good lime-cement mortar.

The ground floors are preferably made of concrete, the top layer being of a hard cement-sand mixture. Brick floors can be applied to those parts of the factory where there is no spilling of water or juice. The platform floors are generally made of wood, and heavy boards 1½ to 2 inches thick should be used, especially in those departments like the filter-press station, which have to be cleaned constantly. In modern factories use is made of concrete slabs for this purpose and these give impermeable floors and improve the cleanliness of the factory.

The selection of the building material is exclusively an economical one, as it will be obvious that brick or hard wood, which might be available close to the factory at low prices and without transportation charges, is sometimes to be preferred. Again, the maintenance cost of brick walls and wooden buildings is low, as they do not need painting at regular intervals, but only white-washing such as can be done by unskilled labour ; this is not the case with painting, as an unskilled painter will use two or three times the amount of paint a skilled man would.

Small sugar factories need only a few buildings, but the larger ones may have a building complex grouped as in the following list :—

(1) Cane house for unloading cane.
(2) Cane carrier house to protect the carrier.
(3) Mill house.
(4) Boiler house and sometimes bagasse store house.
(5) Sulphur and liming house for sulphitation and carbonatation factories.
(6) Clarification house.
(7) Filter-press house.
(8) Evaporator house.
(9) Vacuum pan or boiling house (which is distinct from the boiler house).
(10) Centrifugal and crystallizer house.
(11) Sugar dryer and store house.
(12) Power house.

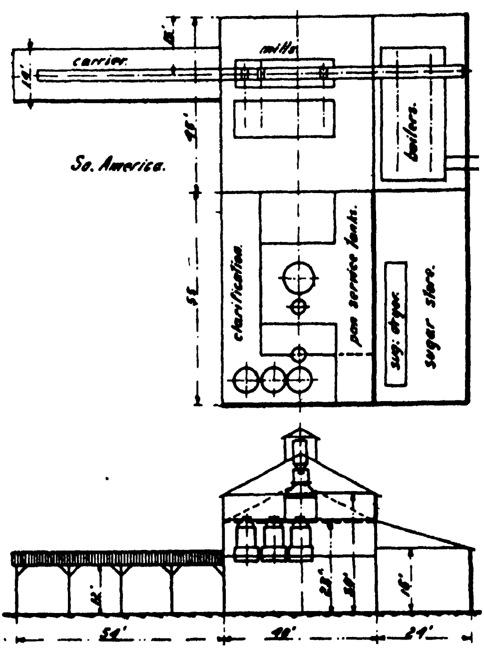


Fig. 89.—120-ton Sulphitation Sugar House.

There are a few other departments requiring a relatively small space, like the laboratory, repair shop, factory store, etc., which generally are located in one of the afore-mentioned buildings.

The construction of the different buildings follows the same fundamental rules and only the arrangement of the machinery or apparatus will demand our consideration.

1.—Building Arrangements.

To give a general idea how cane sugar factories are built, a few examples of typical factories of different capacities, which have been erected in different cane growing countries, are described below.

In *Fig.* 89 is shown a small *Sulphitation Cane Sugar Factory* of about 120 tons' daily cane grinding capacity. The capacity is generally stated per 24 hours, but in Java custom deems 22 hours a grinding day, so as to allow to a certain extent for hours of stoppage. As a rule, it may be accepted that capacity is a very flexible figure, which has been amply proved in Cuba and also in other cane growing countries, where the capacity has been increased without adding extra equipment. This, nevertheless, will require an efficient operating staff and the factory capacity is a function of the latter's ability. Of course there are limits.

The sulphitation equipment of the factory in *Fig.* 89 does not require a special building and is located within the main structure. For larger factories it is convenient to have the sulphur ovens and sulphiting tanks in a separate building, as otherwise the sulphur gases might contaminate the whole atmosphere within the factory.

The cane is unloaded by simple means direct from the cane carts on to the cane carriers. The mill house contains a crusher and two mills and the bagasse is carried by a conveyor to the adjacently located boilers. Between the mill and boiler houses a division wall is sometimes built, to keep out dust and smoke from the former.

The rest of the factory is under one roof, save for the sugar dryer and packing house, which is located in a shed in continuation of the boiler house, but separated by a division wall.

The factory is built according to the *gravity system* as customary with American and some British designs. To save floor space, the vacuum pan is mounted on a high platform, the crystallizers underneath, and then below that come the centrifugals. The massecuite therefore has not to be pumped, but will flow by gravity. These gravity sugar houses should have sufficient windows to allow the daylight to penetrate to the intermediate floors and also to the ground floor; and platforms should be arranged in such a way that from the pan floor nearly the whole factory can be overlooked.

The condensers for the vacuum pan and the triple effect are located in a tower in the main roof.

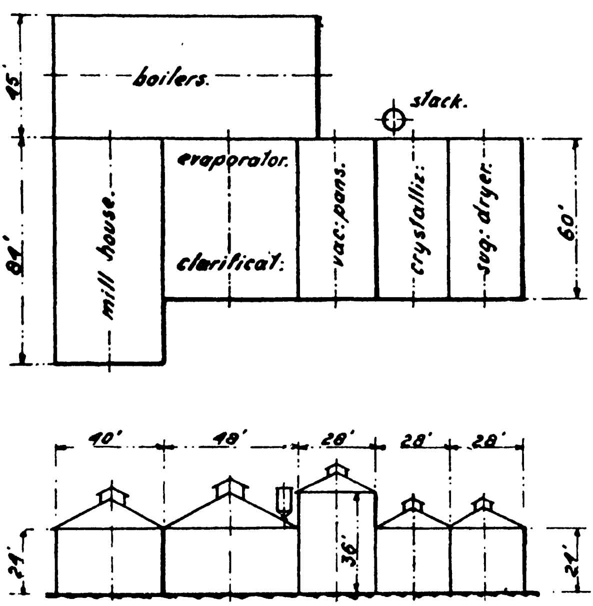


Fig. 90.—400-ton Sulphitation Sugar House.

The covered area is about 60 square feet per ton daily cane grinding capacity and the volume or cubic capacity below the roof trusses about 1,300 cubic feet per ton cane. The buildings form a compact assemblage.

Fig. 90 shows a *400-Ton Sulphitation Sugar Factory* built according to the same system. The boiler house is located at right angles to the mill house axis and the small carrier shed is not shown in the drawing.

This mill is built for *future enlargement,* which is a very convenient consideration when the factory is designed, as it will be sometimes extremely difficult for the designing engineer to arrange additional equipment in existing buildings, so as not to crowd them or prevent easy and efficient operation.

The evaporator and clarification house is next to the mill and boiler house, which will reduce the length and cost of the live and exhaust steam lines. The vacuum pan house is the highest and the condensers are located on the roof of the clarification house at the pan house side. The crystallizer and sugar houses have separate roofs, but they might be constructed as a single span building.

The covered area is about 36 sq. ft. per ton of cane ground per day and the volume 910 cub. ft. per ton cane. It is obvious that floor area and cubic capacity per ton of cane ground per 24 hours will decrease with larger capacities.[1]

Fig. 91 shows the building complex of an up-to-date *1500-Ton Sulphitation Sugar Factory* in Java. The boiler house here is again at right angles to the mill house centre axis and as the factory is completely electrified except for the mill engines, a large power house is provided, where also the vacuum pumps are located. This power house is arranged midway to one side of the boiler house, so that the live steam lines for the power plant and the mill engines will be short. The exhaust steam lines to the boiling house are also of a reduced length.

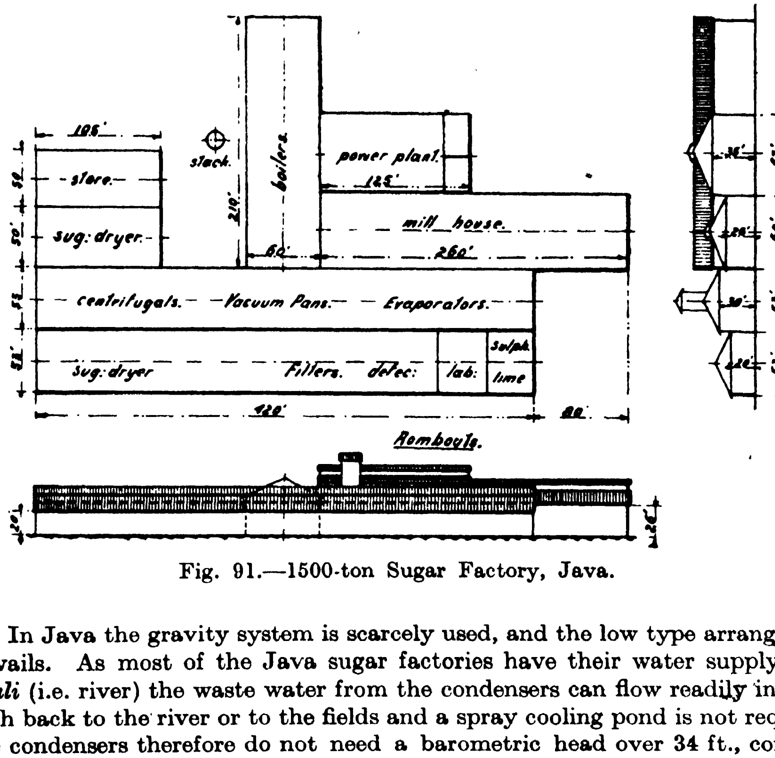


Fig. 91.—1500-ton Sugar Factory, Java.

In Java the gravity system is scarcely used, and the low type arrangement prevails. As most of the Java sugar factories have their water supply from a *kali* (i.e. river) the waste water from the condensers can flow readily into the ditch back to the river or to the fields and a spray cooling pond is not required. The condensers therefore do not need a barometric head over 34 ft., contrary to sugar factories depending on a more limited water supply and where spray cooling is essential. This is also one of the reasons why gravity houses are used in tropical sugar factories, as the waste water can be discharged at the cooling tower head, and waste water pumping thus be avoided.

The factory buildings are not crowded with equipment and an easy supervision is thus possible. The buildings have a steel framing with brick walls erected between the columns and not corrugated wall sheeting, as is common practice on the Western hemisphere.

The evaporators, vacuum pans and centrifugals are located in a bay, 420 ft. long by 53 ft. wide. The filter-press station and the clarification department

1 The designers of the factories shown in *Figs*. 89 and 90 are not known to the author, so credit cannot be given here to the origins.

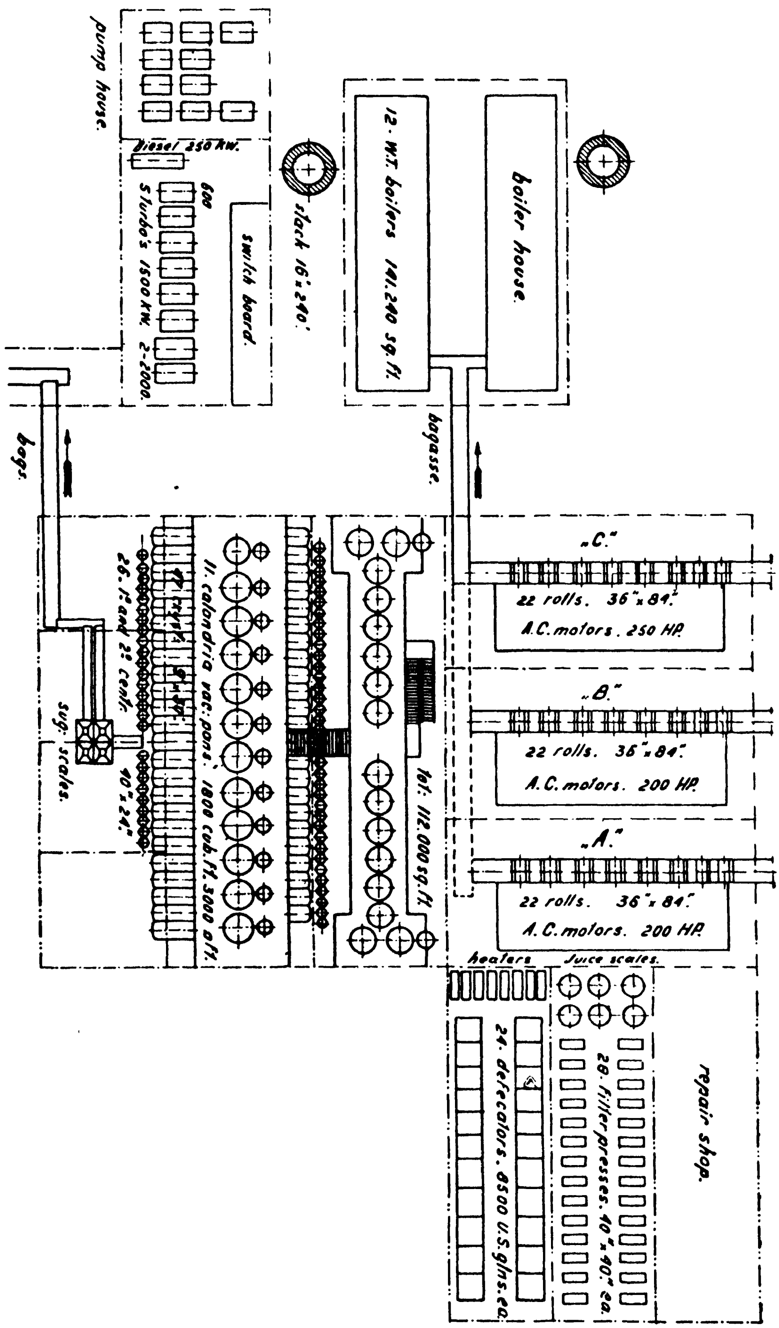
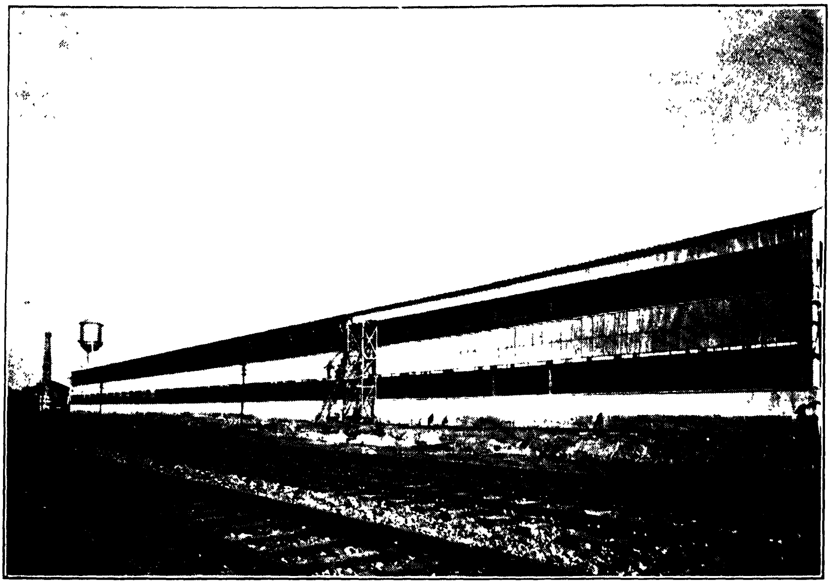


Fig. 92.—11,500-ton Raw Sugar House, Central Jaronu, Cuba.

Front View of a Cuban Mill destroyed by a Cyclone (1933).

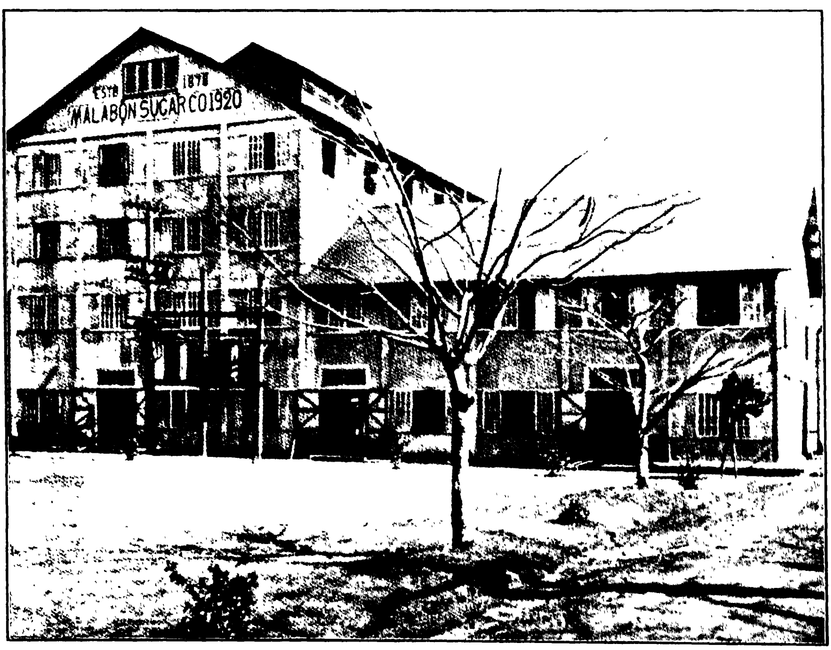
View of the Rear Side of a Cuban Mill destroyed by a Cyclone.

PLATES 11 & 12.



CORRUGATED TRANSIT SIDING AND ROOFING AT A LARGE BRAZILIAN FACTORY.

(Johns-Manville Int. Corp.)



BUILDINGS OF TROPICAL NORIT REFINERY IN THE PHILIPPINES.

are on the outside of the building, so the filter-press cake and factory refuse can be easily removed from the buildings. The sugar drying and sugar storage buildings are located adjacent to the centrifugal house.

The central condenser is placed in a tower on the boiling house roof.

The total covered floor space amounts to about 53 square feet per ton of cane ground per 22 hours and the cubic capacity under the trusses is about 1590 cubic feet per ton of cane. It is seen how spaciously this factory is built, and daylight can penetrate to nearly every corner.

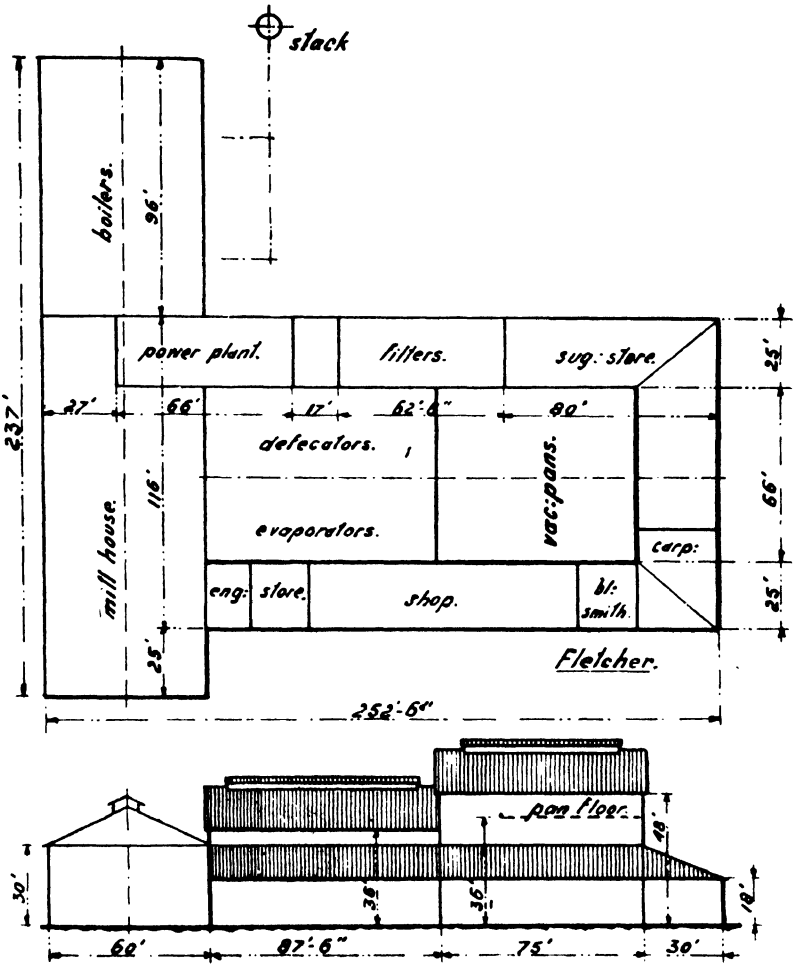


Fig. 93.—1000-ton Raw Sugar Factory, British Dominions.

A very compact design of a *1000-Ton Raw Sugar Factory* of British origin is shown in *Fig.* **93**. The mill house and the boiler house are arranged in line which is very convenient for future extension, and for bagasse storage and cane supply. The steam consumers in the mill house and the power plant are located at short distances from the boilers and the exhaust steam lines are very short, as the exhaust steam consumers in the boiling house are very close to the prime movers. The factory is completely electrified with the exception of the mills.

6

The boiling house is surrounded on the outside by low sheds in which are located the filter-presses, sugar store and repair shops. The factory works on the gravity system and has sufficient height for the condensers, so there are no special roof requirements for this kind of equipment. It should be mentioned that gravity houses are very well ventilated and the pan floor is one of the coolest places in the house, which is an advantage for tropical conditions.

The occupied area is only about 36 square feet per ton of cane ground per 24 hours or about two-thirds of the factory shown in *Fig.* 91. The cubic capacity under the trusses is about 1016 cubic feet per ton of cane. It is obvious that gravity factories involve reduced cost in respect to buildings.

In *Fig.* 92 is shown the *General Arrangement of Central Jaronu* in Cuba, the world's largest cane sugar factory. This layout was sketched by the author during one of his visits to the factory and *Fig.* 92 is not drawn exactly to scale. The factory has a grinding capacity of 11,500 long tons of cane per 24 hours and possesses three tandems, each having two crushers and six mills of the largest size. It is interesting to note the huge amount of equipment necessary for this large capacity, as there are twelve water-tube boilers, having an aggregate heating surface of 141,240 sq. ft., 28 filter-presses, 24 settling tanks of 8500 U.S. gals. each, four quadruple effects of 28,000 sq. ft. each, eleven calandria vacuum pans of 1800 cub. ft. each, 47 crystallizers and 62 centrifugals, 40 in. × 24 in. The power plant has a generating capacity of 12,100 kw., as the complete factory is electrified, including the mills.

Although the factory is very spaciously built, the occupied area is only about 30 square feet per ton of cane ground per 24 hours. The volumetric capacity is high, as the buildings run up to 86 feet under the trusses for the vacuum pan house.

The power plant is located close to the boiler house and only the exhaust steam line is carried into the boiling house, with a small live steam line for the vacuum pans.

2.—Hazards from the Elements.

Sugar factory buildings are more exposed to the elements than the equipment therein, and therefore the buildings have to be designed for the atmospheric conditions of the country where the factory is located. The elements to be dealt with are four :—

(1)—Watershed and rain.
(2)—Cyclones, typhoons and wind.
(3)—Earthquakes.
(4)—Fire hazard.

Rain is a first condition to be considered and it should be recollected that tropical rains are more copious than those in temperate climes. Downpours of 2·5 inches of water per hour are the maximum which have come to the author's knowledge and the roofs of the factory buildings should have sufficient slope to guarantee an easy discharge of the rain water. It is customary to have this slope 4 to 6 inches per horizontal foot. Although it may be considered superfluous to state that roofs should be water-tight, this point nevertheless is greatly neglected at many sugar mills. As the rainy season is generally during the dead season and most of the machinery is dismantled at the time, drippings will cause heavy corrosion on the equipment. Every factory should therefore have a good roof patcher who knows his work, as it is very dangerous; all roof sheets are not of sufficient strength to bear a man's weight and many casualties have resulted from taking risks when walking on old roofs.

Rain gutters have to be provided between sloping roofs. These are also sometimes fitted on the outside of the buildings, but generally of too small a size to be of any use, so they might as well be omitted. A roof overhanging 2 to 4 feet should be provided, so that drippings will not fall right on or close to the building foundations. It is good practice to have a brick-lined ditch or gutter next to the building bases for carrying off the rain water falling from the roofs.

Where rain gutters are provided, they should have a free section (unobstructed) of 0·0002 of the roof area to be drained, or approximately 3 square inches for each 100 square feet of roof area. The gutters should have free discharge at both ends and if drain pipes are fitted at those ends, they should be provided with a receiving tank, so that a small hydrostatic head may accumulate before entering the drain pipe.

As the gutter section for long roofs will become too large to be practicable, intermediate drain pipes have to be located at regular intervals, preferably adjoining the building columns. It is nevertheless found that most designers make these drain pipes too small and the nuisance of overflowing rain gutters results. Drain pipes, therefore, should not have a smaller diameter than 6 inches, as some obstructing matter is sure to be carried along. Leaves or plant growth may not be expected on the roofs of tropical sugar factories, but sometimes a large amount of unburned carbonized bagasse is deposited on the roofs and will pass to the gutters and finally reach the drain pipes. A drain pipe section of 0·00015 of the covered roof area (2 sq. inch per 100 sq. ft. roof) should be considered good practice and the above-mentioned small intake receiving tank has a definite purpose.

As the rain does not fall vertically in most instances but is affected by the wind velocity, so that under cyclonic conditions the rain will be driven in nearly a horizontal line, the designer has to take care that louvres and ventilation gaps are covered in the horizontal plane.

Of much more importance to the strength of the building are the *wind forces*, and maximum wind velocities should be ascertained at the site where the factory has to be erected. From formula (30) the wind pressure is derived, when the wind velocity in miles per hour is given. The factory has to have sufficient strength to withstand the impact of the wind force, and a well designed building will stand the heaviest wind pressures. This applies especially for those countries affected by cyclones, and Plates 9 and 10 give graphic views of a Cuban mill destroyed by a disastrous hurricane in 1933, which has since been reconstructed under the author's supervision. The buildings in question were in fairly good condition and had been properly maintained. As they were constructed about 25 years ago, they had withstood many a storm, but the last cyclone was of such terrific force, that the buildings collapsed, the wind pressure being around 42 lbs. per square inch, as could be calculated from an empty molasses tank of 16 tons weight, which was blown from its foundations.

As a curiosity it may be mentioned that the smoke stack (of the self-supporting type) did not come down, and the construction of this American-designed chimney is to be found elsewhere (page 303).

Even the two condensers went down, one completely to the ground, but it is rather doubtful if these were actually tumbled by the wind force, as it is more likely to expect that the falling roofs would drag along the heavy vapour lines and damage the lower parts of the condenser towers.

The *Course of a Cyclone* is shown graphically in *Fig.* 94. The wind force is directed towards a place of low barometric pressure, the vortex, and readings down to 27 inches mercury are known to the author. Around the vortex at a considerable distance are two diametrically opposed loci, where a high barometric pressure prevails, and which are called the anti-cyclones. Through this phenomenon there are two wind arrows with recorded wind velocities of from 80 to 150 miles per hour, and the whirl moves along its course with a travelling velocity of 8 to 15 miles per hour. From *Fig.* 94 it will be seen that the first impact from the North, say, will have a moderate effect on the factory buildings shown, whereas the second impact from the South will exercise all its force on the factory structures. At the locus of the vortex, there is a complete lull, as the wind is whirling high overhead towards upper strata.

There are official Government bodies, like the U.S. and British Weather Bureaux, which broadcast on the radio and file atmospheric conditions within their jurisdictions, and although this is meant principally for ships on the high seas, information from these announcements can be assimilated for land purposes as well.

To calculate the resistance building structures must have, so that they will not collapse, it is assumed that the wind forces on a vertical wall act horizontally. On the inclined roof area, the *normal wind pressure* ***N***, as shown in the upper left part of *Fig.* 95, is calculated in a different way, as any deviation from the horizontal will increase the normal wind pressure on the roof and maximum values therefore have to be considered.

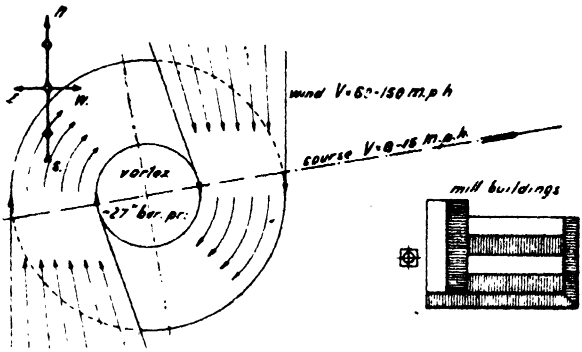


Fig. 94.—Course of Cyclone.

Let P be taken as the horizontal wind pressure in lbs. per square foot and β be the angle of inclination of the roof, then evidently the normal wind pressure N_1 will amount to :—

$$N_1 = P \times \sin \beta$$

From experiments of DUCHEMIN it has been found that the normal wind pressure is larger, and this author has developed the empirical formula :—

$$N = P \times \frac{2 \sin \beta}{1 + \sin^2 \beta} \qquad \ldots\ldots\ldots\ldots\ldots\ldots \quad (45)^1$$

where P is the horizontal wind pressure on a vertical plane and β the roof sloping angle. From the upper left part of *Fig.* 95 it is seen that, according to DUCHEMIN'S formula, the wind pressure P is assumed to be acting at the angle α with the horizontal. Several authors take this angle as about 10° with the horizontal.

The building structure is furthermore exposed to a lateral wind force P_1 on the side of the building. The normal wind pressure on the roof N is acting

[1] From MERRIMAN, "Civil Engineers' Handbook."

at right angles to it and the combined weight of the truss and roofing W will act vertically. The resultant force R given in the centre diagram of *Fig.* 95 has a collapsing effect on the building and it is assumed that both columns receive 50 per cent. each of the force R for that particular section. The columns are held in upright position by the foundation bolts or their anchorage, whereas knee braces Z give a firm connexion to the trusses. It is obvious that heavy lateral wind forces will cause a bending action on the columns as shown in the right part of *Fig.* 95, assumed to be caused by two couples of forces $Q \times e$. If the base connexion of the columns is to be considered as absolutely fixed, then the *point of counter-flexure*, i.e., the point where there are no bending stresses in the column material but only shear, will be located midway between the base and the connexion of the knee brace to the column, thus at $h/2$. In practice an absolutely fixed base connexion can hardly be obtained and the point of counter-flexure O is considered as at a height of $h/3$ from the column base.

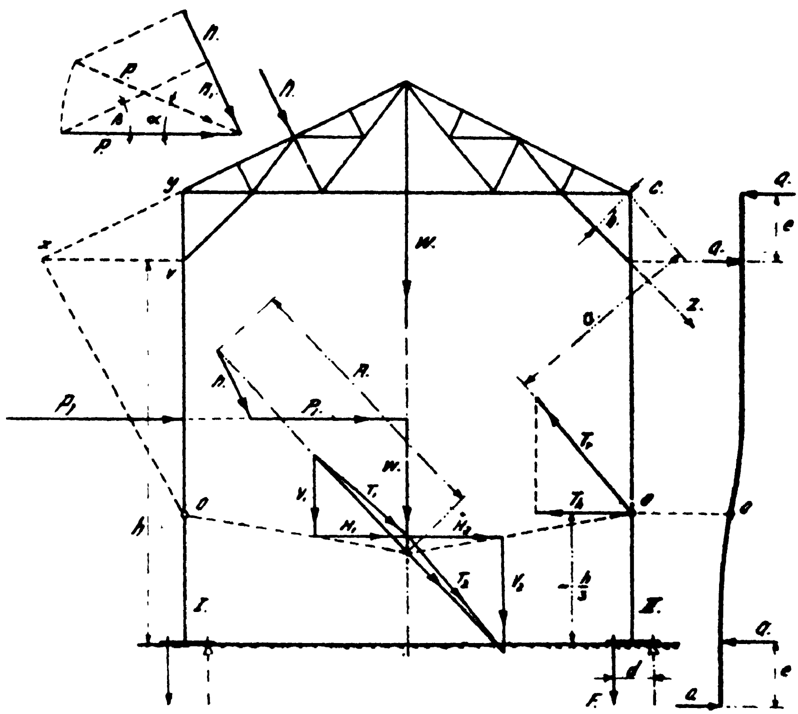


Fig. 95.—Column Bending and Wind Bracing.

The resultant force R is divided into two equal parts, each half acting on the opposite column and is split into two equal horizontal forces H_1 and H_2 and two vertical forces V_1 and V_2. The latter are not equal, as evidently the left hand column I receives a lifting impact, so the downward reaction is reduced, whereas the downward reaction of column II is increased. The resultant forces T_1 and T_2 are shown for the corresponding column.

. The counter-force T_r of column II is acting at the point of counter-flexure O, causing a momentum on point C and the knee brace stress has to balance this momentum, thus :—

$$T_r \times a = Z \times b \quad \dots\dots\dots\dots\dots\dots\dots\dots \quad (46)$$

where a is the distance between T_r and C, whereas b is the distance between the knee brace and this same point C. Z is the *total stress in the knee brace* and it should be recollected that this stress is a tensile one for the left hand brace and a compression stress for the right hand one. The knee braces therefore have to be dimensioned accordingly.

As the roof truss stresses are generally determined by a Cremona diagram of forces, the auxiliary members yx, xv and xO are assumed to exist, as their presence will not alter the stresses in the rest of the structure.

The foundation bolts are dimensioned by the momentum $T_h \times h/3$. T_h is equal to H_1 and H_2 and the equation reads :—

$$F \times d = T_h \times h/3 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (47)$$

where F is the total tension stress in the left foundation bolt or bolts and d

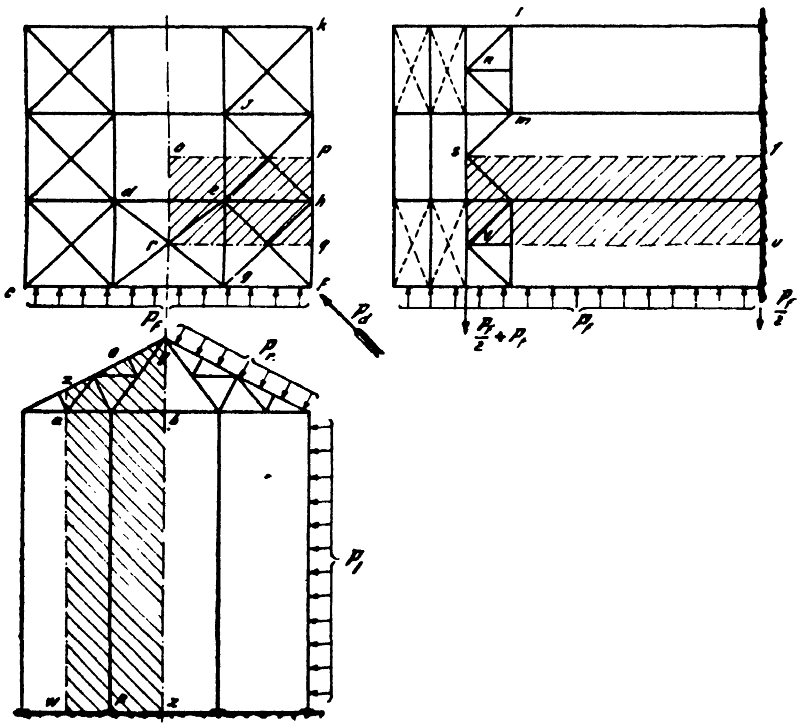


Fig. 96.—Front and Lateral Wind Forces.

the distance between the bolts. It is assumed that the bearing line of the base plate coincides with the centre line of the right hand foundation bolts. Furthermore, it will be obvious that the column base also has to resist this bending momentum and it is no use having heavy foundation bolts, when the connexion between the base plate and the column is not equally solid. The anchorage of columns I and II is subject to equal forces.

The *front of the building* is also subject to the impact of the wind force and the front columns po have to resist the wind pressure on the area $wxyz$, as shown in *Fig.* 96. The column base has to resist a shearing action $P_f/2$, when P_f is the total wind force on the area $wxba$, whereas the lateral truss reaction is equal to $P_f/2 + P_t$, where P_t is the wind force on $abyz$.

At the bottom chords of the two end trusses a bridge construction *cdef* is provided to resist this part of the wind force on the building front.

The lateral wind force P_1 already mentioned acts on the rectangle *stuw*, whereas the normal wind pressure on the roof is determined by the inclined area *opqr*.

When the wind force acts diagonally in the direction P_d, the upper building structure is subject to a smashing action and reinforcements *ghjk* at the bottom chords of the trusses and knee braces *lmn* between the columns have to be provided.

The dotted lines in the side view indicate the usual reinforcing bracing with tie rods between the top members of the trusses.

With the reinforcements applied good protection is obtained against cyclone hazard, and the author has carried out such construction in practice with good results.

In certain countries *earthquakes* will occur and it is very difficult to ensure proper protection, as the intensity and direction of the acting forces are not known. The structure shown in *Fig.* 6 has withstood several earthquakes of average intensity, and good bracing of the building structures is the essential factor.

The fourth element to be considered is the *fire hazard* and, as already mentioned, the building proper is less subject to fire than the contents stored therein. Heavy columns of hard wood will not easily catch fire, but thin boards are naturally more susceptible, and should be used as little as absolutely necessary. Of several factory fires known to the author, the cause had to be sought for in the sugar stored in the building, and even steel buildings will not offer an adequate protection, as the heat developed is of such a temperature that iron structures have not sufficient resistance and will crumble down. The sugar store, therefore, should be in a building separate from the factory proper. Spontaneous combustion of sugar is liable to take place when moisture content and microbic action cause deterioration and sufficient heat is evolved to cause ignition.

3.—Details of Steel Buildings.

In *Fig.* 97 is shown the construction of a mill building with travelling crane. The columns are of constructed plate and angle sections and the roof trusses all have knee bracing for lateral wind forces. On the apex of the roof there are louvres at the sides for ventilation.

The purlins are made of channels, dimensioned according to the wind force prevailing and these are covered with galvanized corrugated sheets, having corrugations 2½ in. and ¾ in. high. These sheets are subject to heavy corrosion, especially when close to the sea, and have to be painted regularly.

Corrugated sheets are available in lengths from 4 to 10 feet and a width of 27½ inches. The overlapping has to be one corrugation in crosswise direction and 6 inches in lengthwise direction. To allow for overlapping, the total area of the sheets has to be 15 to 20 per cent. in excess of the roof surface depending on the sheet length.

The thickness of the corrugated sheets should not be under 22 B.W.G. for the roofing, whereas No. 24 B.W.G. can be used for side walls. The weight of these sheets is respectively 1·54 and 1·26 lbs. per square foot. For more enduring capacity, sheets of No. 20 B.W.G. should be selected, having a weight of 1·84 lbs. per square foot.

At small additional cost, copper bearing galvanized steel sheeting can be bought, having a better resistance against corrosion.

Many tropical sugar factories are covered by corrugated sheets of asbestos cement, which are nearly everlasting and of good insulating capacity. The thickness of good asbestos sheeting is $\frac{3}{8}$ in. to give sufficient strength against heavy wind forces. The corrugations have 4·2 in. pitch and the sheets are made in lengths of 3 to 11 feet by 42 inches wide. The weight is 4·1 lbs. per

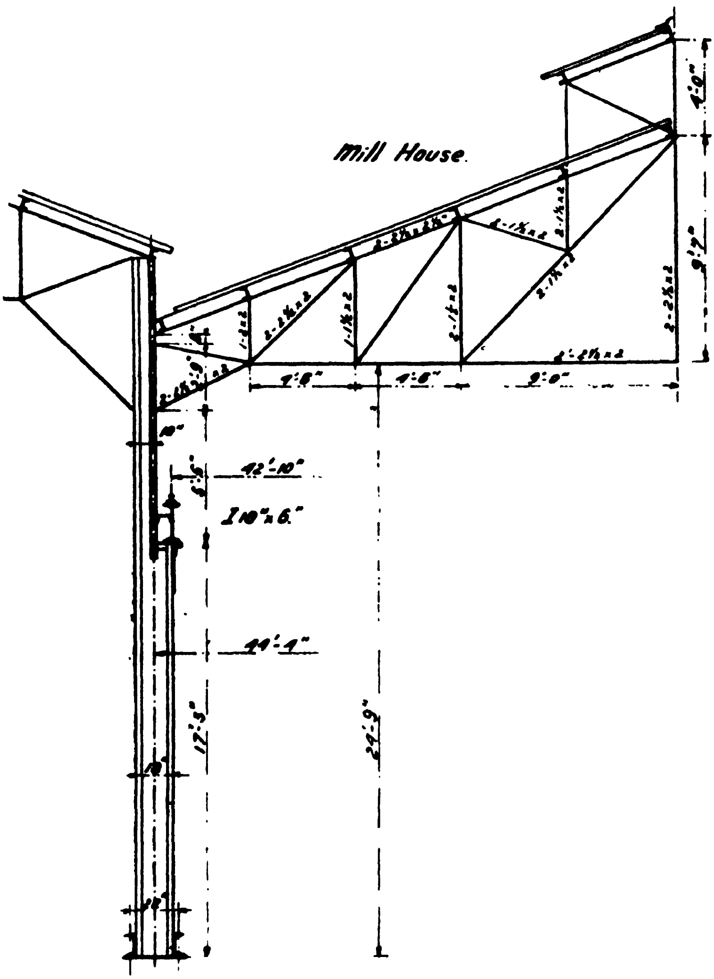


Fig. 97.—Mill House Framework.

square foot and the cost price is considerably higher than for galvanized sheets, but they are to be preferred as the interior of the building remains cooler and painting is not required.

The purlins are of channel section, 6 to 7 inches high according to span and sheet length. The open side of the channel should be located sloping downwards, as otherwise leaking water might accumulate and start corrosion. The purlins are interconnected between the trusses by $\frac{5}{8}$ in. tie rods, so but little sag will take place, as the opposite top ones are also held together.

The span of the building of *Fig.* 97 is 44 ft. 4 in. and the crane runway is of sufficient strength for a 20-ton crane, the columns being 15 ft. apart.

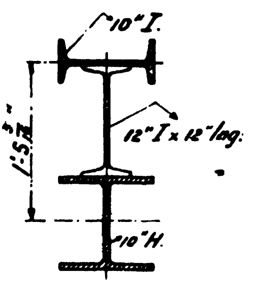


Fig. 98.—Mill House Column.

A heavy *Column Construction* for a large mill house, having 57 ft. 10 in. span and 40 ft. 0 in. under the trusses, is shown in *Fig.* 98. The main building columns are of 10 in. H-section, 46 lbs. per foot, whereas the crane runway rests on columns of 10 in. I-section. The crane and building columns are joined together by pieces of 12 in. joists at regular intervals, to give the lower part of the column an increased moment of inertia. The crane-carrying capacity is 20 tons and the crane span 55 ft. 0 in. centre to centre of rails. The building columns are placed 24 ft. 0 in. apart.

A *Boiling-house Roof Construction* is shown in *Fig.* 99, having a span of 89 ft. 4 in. and a height of 52 ft. under trusses. The roof is of an interrupted FINK construction, having a slope of 6 in. per foot. The purlins are 7 in. channels, 5 ft. 6 in. apart for 6-ft. corrugated sheets. Heavy knee bracing is provided for and the front and rear have standard window frames as shown.

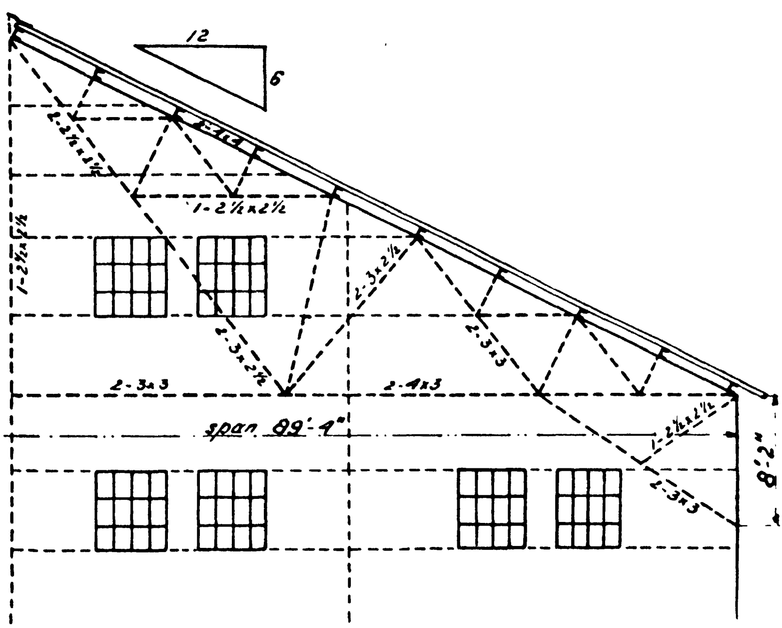


Fig. 99.—Large Span Boiling House.

For overall calculations the *truss weight* has to be estimated in advance and KETCHUM[1] has developed the formula :—

$$W = \frac{P}{45}\left(1 + \frac{L}{5\sqrt{A}}\right) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (48)$$

where : W = truss weight per sq. ft. horizontal projection.
P = bearing capacity of truss in lbs. per sq. ft. horizontal projection.
L = span in feet of truss.
A = distance between trusses in feet.

1 BURR & FALK—"Influence Lines for Roof and Bridge Construction."

This formula will give an average weight of :—

2 lbs.	per sq. ft.	covered	area	for	20 ft. span.
3 lbs.	„	„	„		40 ft. span.
4 lbs.	„	„	„		60 ft. span.
5 lbs.	„	„	„		80 ft. span.
6 lbs.	„	„	„		100 ft. span.

Window frames are nowadays a standard manufactured product, and in U.S.A. standard panes of 12 in. × 18 in. and 14 in. × 20 in. are used. The frames are manufactured from 3 to 6 panes wide and from 2 to 7 panes high. For ventilation revolving frames are built, having pivots 2 inches above the horizontal centre, so that the windows will close by gravity.

The window area for tropical sugar factories need not be too large, as the light is very penetrating and direct sunlight is not desired. From practical experience the author inclines to 6-10 sq. ft. window area per 100 sq. ft. floor space, and intermediate floors have to be counted as well. Sugar warehouses will need much less. Corrugated glass panes for transparent roof covering are not used to any big extent in sugar factory buildings, to avoid direct sun rays.

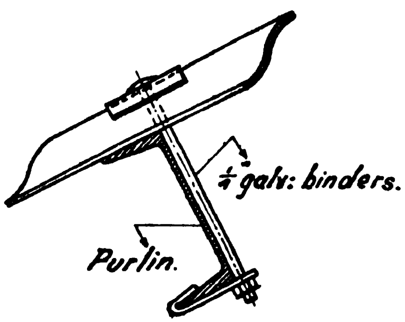


Fig. 100.—Sheet Binders.

All buildings should have proper *ventilation* and the air entrance has to be close to the ground level, whereas the air exit is effected by louvres in a monitor on the roof or on the sides of the building. For sugar warehouses ventilation is only allowable under special atmospheric conditions and in the corresponding chapter this will be dealt with.

Revolving roof ventilators are used in many instances as they take advantage of the direction of the wind and its kinetic energy, resulting in a forced draft. Manufacturers state that with an average wind velocity of only 5 miles per hour a ventilating capacity of 350 cub. ft. per minute will be achieved per square foot free ventilator duct area.

A good and approved construction of *Roof Sheet Binders* is shown in *Fig.* 100. The corrugated sheets are attached to the purlins by galvanized ¼ in binders with nuts and a bent galvanized strip around the bottom flange of the purlin. A lead washer is generally placed under the flat head of the binder, but in cases where a lasting attachment is required a galvanized plate, about three inches long and corrugated according to the roof, will prove a better construction where cyclones prevail, as otherwise the roof sheets are easily torn off. A lead washer can be used on the top and the cover plate should be well painted underneath with thick red lead.

4.—Details of Wooden Buildings.

The author has been commissioned at times to design wooden factory buildings and this is of special interest to those factories which have an abundant supply of hard wood within easy reach, as it may prove an economical proposition.

Fig. 101 shows a *wooden roof truss* for an existing sugar warehouse, having a span of 81 ft. 0 in. The whole truss is built from boards 9 in. × 2 in., well bolted together with ¼ in. steel gusset plates. Each truss is supported on two intermediate columns to increase the strength of the truss, as the sugar bags are hoisted by tackles attached to the truss cords. The purlins are 5 in. × 3 in. and the distance between trusses is 15 ft.

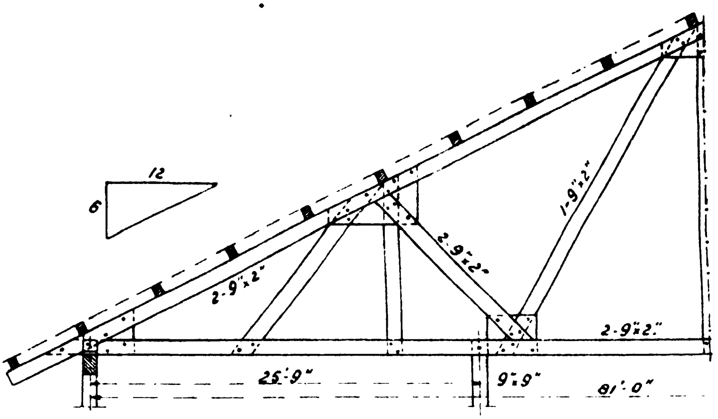


Fig. 101.—Wooden Roof for Sugar Warehouse.

A *Heavy Wooden Column Splice* for a 12 in. column for a sugar factory is shown in *Fig.* 102. The dovetail is held together by two wedges *a* of hard wood, which are driven in firmly and the whole splice is reinforced by two 12 in. channels, of which the flanges are flush with the sides of the wooden column. Five 1-inch bolts keep the splice and channels together.

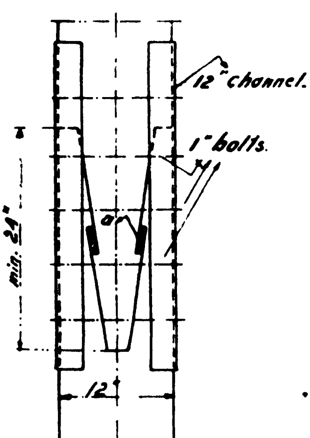


Fig. 102.
Wooden Column Splice.

Wooden buildings have been built to the same dimensions as steel buildings and the author knows instances of wooden buildings having 80 ft. span and a height of 68 ft. under trusses.

5.—Travelling Cranes.

For dismantling and repairing heavy machinery, hoisting equipment is necessary and in nearly all mill houses of cane sugar factories, where dismantling, exchange of mill rolls, etc., is frequently taking place, a travelling overhead crane is provided. The hoisting capacity of such a crane is determined by the heaviest piece of machinery it has to handle with a fair excess for eventual future requirements and additional safety. For new factories, a travelling crane will speed up erection and generally has paid its cost before the factory starts working. Moreover, the pieces of machinery and especially the mill rolls are of such a weight that they cannot be hoisted on the existing roof trusses.

A mill house having 7 ft. mills should be equipped with a travelling crane of not less than 20 tons hoisting capacity and provided with two trolleys, each designed to take the full load.

In the boiling house a travelling crane is convenient for erection and repairs, but dismantling of apparatus in this department is not frequent, and therefore in many sugar factories the crane is omitted here.

Large electrified mills will have electrically-driven cranes. It is, nevertheless, not necessary to have the three operations, viz., hoisting, traversing and travelling, power-driven. The hoisting is of most importance, whereas traversing and travelling can be done by hand gear. Electric hoisting will save a considerable amount of time, as the pulling chains of the hand gear are not always in an accessible place, which is another inconvenience of hand-operated travelling cranes.

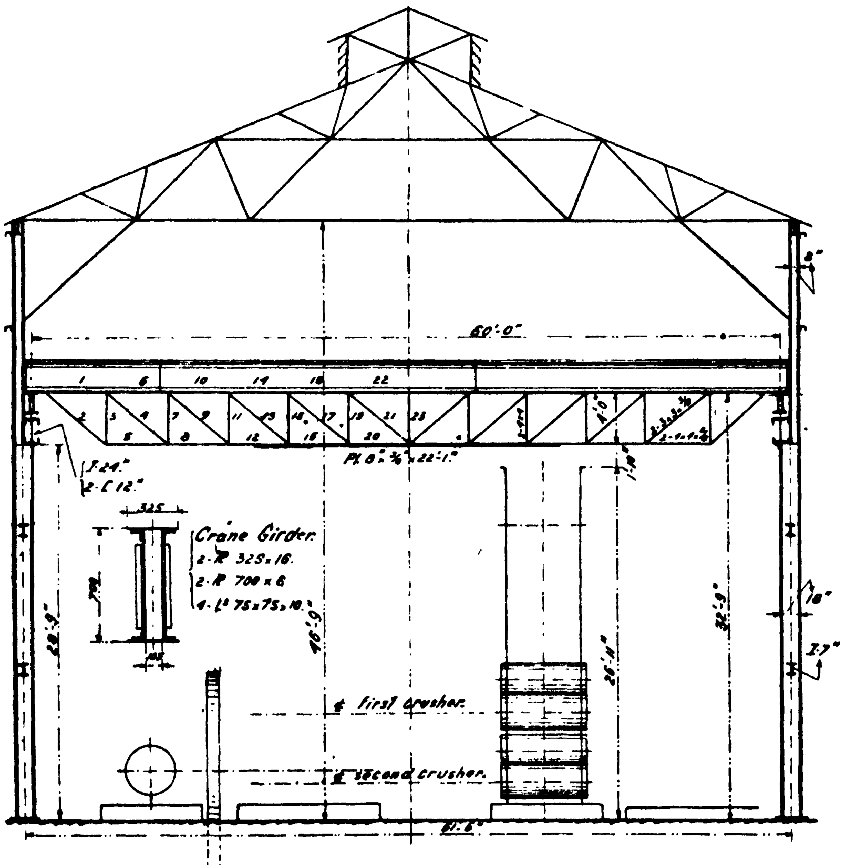


Fig. 103.—Mill House with Travelling Crane.

A typical *Mill House with Travelling Crane* which the author had to design, is shown in *Fig.* 103. The crane has a span of 60 ft. and has to pass over the top of the cane chute, 26 ft. 11 in. above the factory floor.

The building is made of steel, having 18 in. columns of I-beam section. As the radius of gyration is small, when taken on an axis parallel to the web, struts of 7 in. I-beams are applied between the columns, to reduce the compression length of the column in relation to the stated axis. The distance between columns is 18 ft.

The crane girders proper had to be constructed from existing box girders at hand, of a built-up section 325 × 700 mm., as shown in detail in *Fig.* 103. On account of the long span, these girders proved to be weak and, moreover, the pieces at hand were short, so very costly splicing had to be done; it was accordingly decided to make a bridge truss underneath, 4 ft. high.

In *Fig.* 104 is shown the *Cremona Diagram* for the crane truss of *Fig.* 103. The crane has two trolleys of 20 tons hoisting capacity each, although the total load will not exceed 20 tons. The proper weight of the crane and hoisting gear is about 19 tons and for reasons of safety each reaction P for each crane girder has been taken as 10 tons.

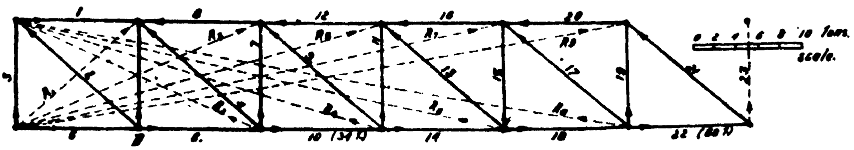


Fig. 104.—Cremona Diagram for Crane.

From *Fig.* 104 it is seen that the following computation has been made :—

		P resultant of forces in			1 and 2.
		2	,,	,,	3 and 5.
1 and 3	joined to	R_1	,,	,,	4 and 6.
4 and 5	,,	R_2	,,	,,	7 and 8.
		etc.			
16 and 17	,,	R_8	,,	,,	19 and 20.
18 and 19	,,	R_9	,,	,,	21 and 22.

The vertical members 3, 7, 11, 15, 19, 23 are all compression members and are made of two angles 4 × 4 × $\frac{3}{8}$ in. These members are subject to a maximum compression of 10 tons. Members 5, 8, 12, 16, 20 are tension members with increasing tensile stresses up to 57 tons made of two angles 4 × 4 × $\frac{3}{8}$ in. and reinforced in the centre by a flat bar 8 × $\frac{3}{8}$ in. and 22 ft. 1 in. long, whereas members 2, 4, 9, 13, 17, 21 are tension members of 15 tons max. each, and made of two angles 3 × 3 × $\frac{3}{8}$ in. The members for heavy compression, 1, 6, 10, 14, 18, 22, are formed of heavy box girders.

For long cranes lateral reinforcements are sometimes provided, as it will prevent lateral deflection. This is especially important for power travelling cranes, where lateral inertia forces prevail. In this hand-operated crane, such reinforcement is superfluous, as the box girders are of sufficiently stiff construction.

Instead of using the graphical method of CREMONA, the analytical method of RITTER or the method of moments can be applied as well. This method is based on the fact that the resultant or algebraic sum of all moments of forces in a certain section of the truss, and revolving around the same point, has to be zero, when the system is balanced, thus :—

$$\Sigma M_0 = 0 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (49)$$

For new buildings a standard crane from manufacturers of this kind of equipment and designed for hand or power operation, or both, will be the most economical proposition.

The *Trolleys* of 20 tons carrying capacity to be used for the crane of *Fig.* 103 are shown in *Fig.* 105. Each trolley is designed to carry a chain hoisting block of 20 short tons capacity and the hoist is attached to the trolley by means of an eyebolt, or else a U-bolt as shown by dotted lines, this latter being a better construction.

The chain hoist is of the spur geared type, having a reasonable degree of efficiency and a good brake, so that the load will stop at any position when the movement ceases. These 20-ton hoists have two hand chains, each needing a pull of from 140 to 180 lbs. to lift the full load; and the chain ratio varies from 210 to 160 respectively, so the hand chain has to be pulled from 210 to 160 feet, to lift the load one foot. The pulling force of a man is around 80 lbs. so four men are needed to operate this hoist. The minimum distance between hooks is from 70 to 60 inches. Although the cost price is low, the hoisting speed is only 0·6 feet per minute as a maximum, whereas electrically-driven hoists can easily attain ten times this speed, according to the size of the electric motor. As will be seen, the construction of the trolleys is made of a few channels in a very simple manner.

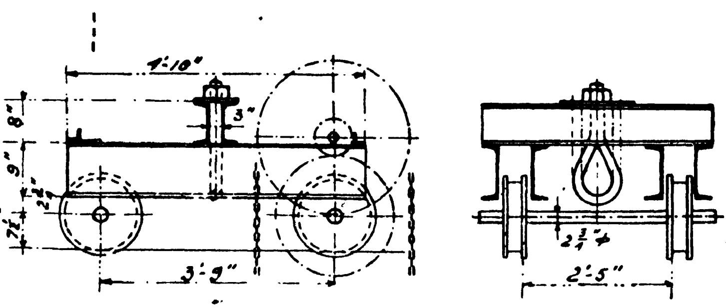


Fig. 105.—Trolley for 20-Ton Chain Hoist.

Electrically driven cranes use direct current, three-phase alternating current or single-phase A.C. Direct current has the advantage that the motor speed increases with small loads, as series wound motors are used for this purpose. Alternating current motors, on the other hand, have a constant speed according to the frequency of the current. For sugar factories, A.C. is the one chiefly applied for electrification and therefore will be the most used for crane driving.

The main travelling crane is a clumsy medium, when small loads have to be lifted which are yet beyond man-lifting power. The chief engineer of a large Cuban sugar factory therefore devised a system of trolleys, operating on I-beams, attached to bottom chords of the trusses. The arrangement of such an *I-Beam Trolley* is shown in *Fig.* 106; it is located over the mill housings, so the lighter pieces, like king bolts, carrier slats, sides, etc., can be more quickly removed than would be the case with the big travelling crane.

The truss has to be of substantial design, otherwise reinforcements have to be applied or additional members marked *a* are built into the existing truss, as the case may require. The trolleys are moved by pulling or pushing the load. On account of the considerable height, the chain block is suspended on a rod and the rod attached to the trolley. The size of I-beam to be used depends on the lifting capacity of the chain hoist and the distance between trusses. In *Fig.* 106 a 2-ton trolley is drawn.

Two or three of these trolley lines will be a big improvement in hoisting efficiency and several trolleys in front and behind the big travelling crane should

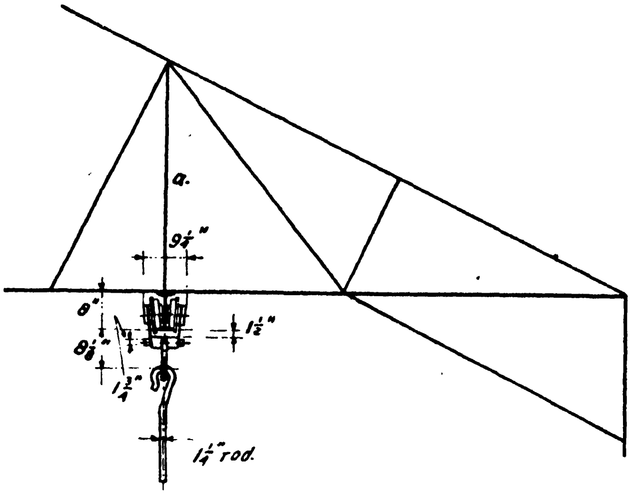


Fig. 106.—I-beam Trolley for Mill House.

be so fitted that they will not be interfered with by the movements of the main crane.

6.—Platforms.

The most economical method is to have the platforms so arranged that they form an integral part of the building structure. Any vibrations from evaporators or vacuum pans are generally due to faulty circulation and do not occur regularly, and crystallizers and tanks have only a dead load, so there is no inconvenience in having integral platforms.

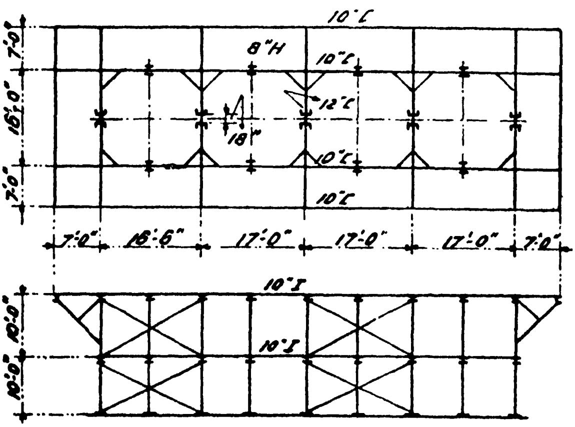


Fig. 107.—Platform for 28,000 sq. ft. Quadruple Effect.

In wooden buildings or existing steel buildings with light column sections, separate platforms or stagings have to be designed. *Fig.* 107 shows a *Platform* designed by the author for a quadruple effect of 28,000 sq. ft. heating surface. This platform has been arranged in an economical manner as the footings of the bodies are just on top of the columns, so the top frame can be made of light section, as it serves only as a connecting member, for attachment of the wooden floor.

All platforms should be well braced between columns in both directions, and the top platform, also, to prevent eventual twisting action. Wooden floors are nailed on wooden rafters, which are bolted on top of the steel floor beams. The top of the steel beams should be painted with three coats of red lead before applying the rafters.

Stairways leading to the platform should be of easy ascent and sloped 45° when possible. The steps should be 8 inches high and 8 inches wide. A smaller height for the steps is inconvenient, but they may be wider, though this will decrease the sloping angle and thus increase the space occupied. Overlapping steps are inconvenient when descending the stairs. When stairs are too steep, they are dangerous and for considerable heights should have intermediate platforms. Steel tube or wooden railing, supported on stanchions, should be provided on all stairs on both sides and around all platform edges.

7.—Foundations.

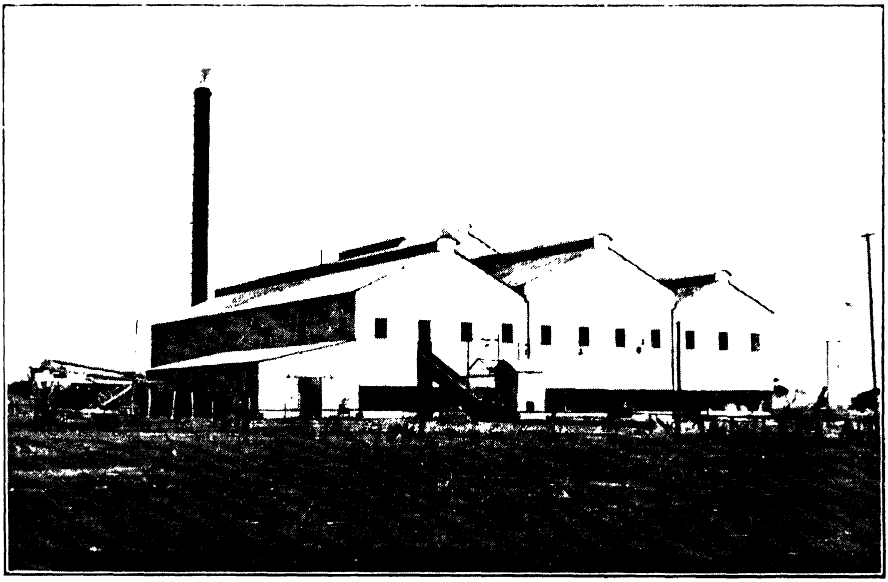
Foundations for the building columns and equipment bases should be laid on firm soil of sufficient bearing capacity. The soil bearing generally increases with the depth, and foundations for machinery should never be laid on unsuitable soil, like quicksand, etc.; the factory site should be selected with a view to avoid this. Pile driving (see Chapter I) may be necessary for a railway bridge head but would prove very costly for factory foundations. A rock bottom gives an excellent bearing, but it should be kept in mind that excavations in rock strata for cane car dumpers or underground pipelines are very costly.

Good dry and compact soil will bear safely from 8,000 to 12,000 lbs. per sq. ft., but for average conditions only 4,000 lbs. per sq. ft. (2 kg. per cm^2) should be allowed. The author knows instances where foundations, having a large bearing surface, vibrated through unbalanced inertia forces of the machinery placed upon them, and this can only be remedied by increasing the dead weight of the foundation block, which is costly. Engines, therefore, should be of the balanced type.

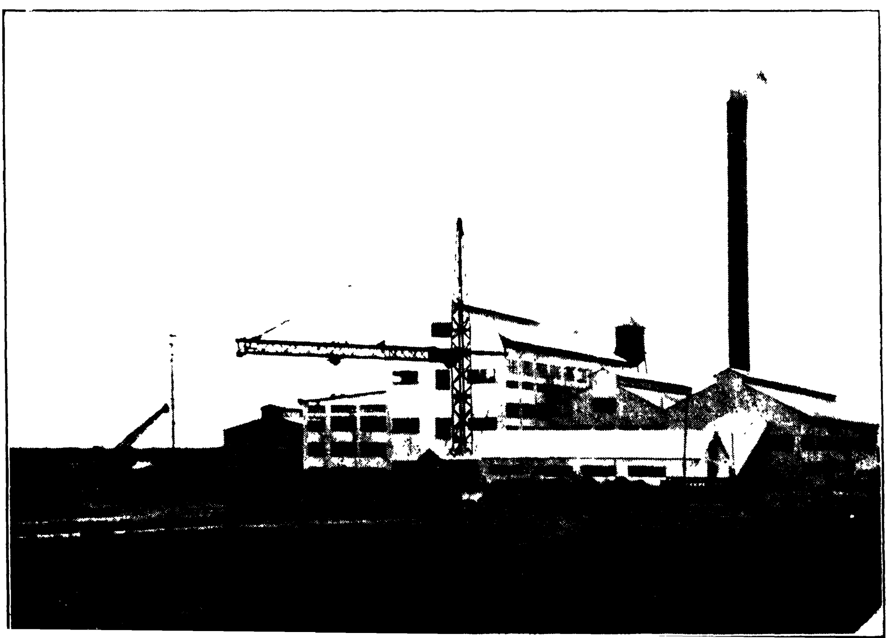
The base plates of columns, etc., on good concrete foundations will bear up to 30,000 lbs. per sq. ft. (15 kg./cm^2) but a certain allowance should be made for errors such as faulty grouting, and 20,000 lbs. per sq. ft. should be considered as normal. Brick masonry has a safe bearing of about two-thirds of this figure, but this depends not on the crushing strength of the brick, but on the mortar used.

In *Fig.* 108 is shown a *Column Foundation* of a type often laid out by the author. The soil bearing surface is large and the concrete is spread on top of a layer of broken or cobble stones. A timber mould is used to save concrete and the foundation bolts are hung in a wooden frame whereas a piece of old tube, preferably a boiler tube from 3 to 4 inches diameter as may be at hand, is placed between the wooden frame and the foundation plate, so that lateral movement is possible; this might be convenient when there is a slight difference in the foundation bolt holes of the base plates. The space around the bolt should be covered by a small plate or similar arrangement, so that it will not get filled with the grouting cement.

The allowable tensile strength of the foundation bolt should not exceed 10,000 lbs. per sq. in. measured on the sectional area at the bottom of the thread. The foundation plate bearing should be 50 sq. in. for each sq. in. bolt section and the plate be of $\frac{5}{8}$ in. thickness as a minimum.

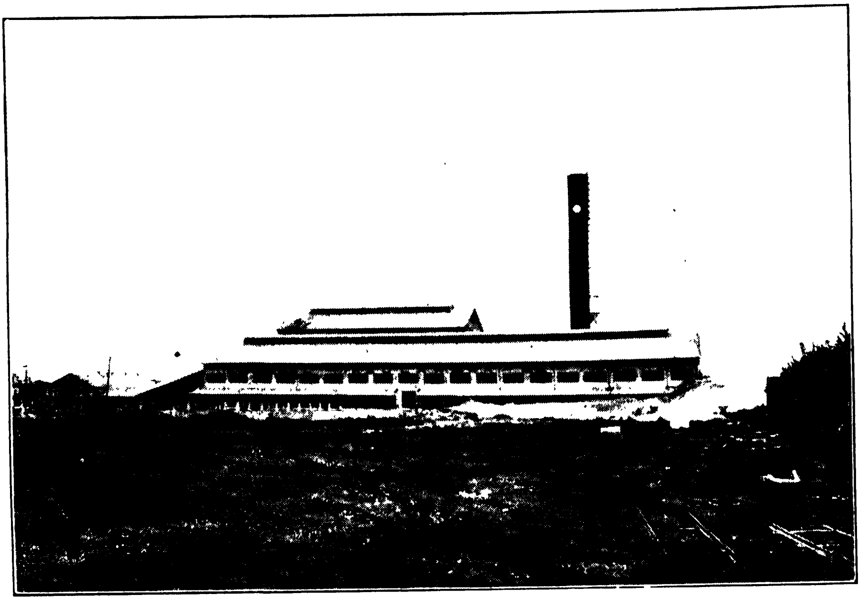


600-Ton Cane Sugar Factory in India.
(*A. & W. Smith & Co., Ltd.*)



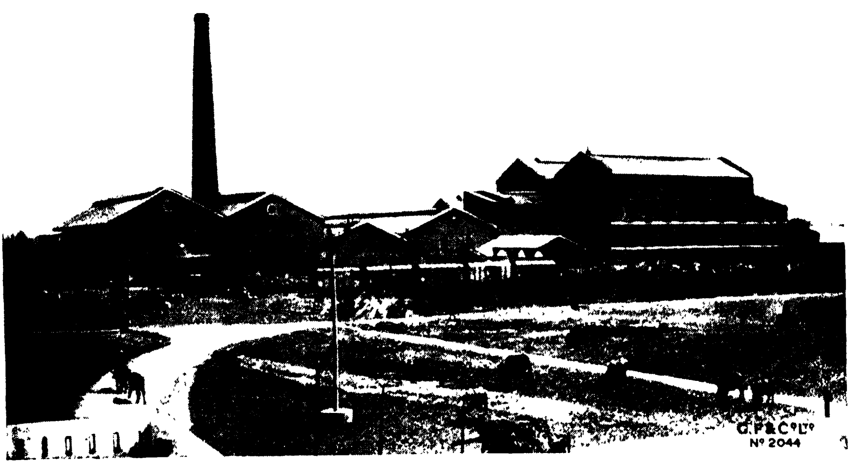
Front View of Del Mante Factory in Mexico.
(*Fulton Iron Works Co., Inc.*)

PLATES 15 & 16.



Central Azucarera de Tarlac, Philippines.

(Fulton Iron Works Co., Inc.)



1,300-Ton White Sugar Factory in India.

(Geo. Fletcher & Co., Ltd.)

For concrete foundations the bolts are sometimes completely embedded in the concrete, having a right angle bend at the bottom. For heavy foundation work this construction is less reliable than the one shown in *Fig.* 108.

The concrete mixture is of the utmost importance and although concessions are sometimes made for economical reasons the proportion cement : sand : broken stone by volume should be 1 : 2 : 4. Sharp sand is to be preferred and the size of the stone depends on the kind of work, as for small moulds or where reinforcement is used a smaller size of stone is required than for large foundations. In many sugar mills the stone is crushed at the factory and a definite grading according to size is not effected. This is no inconvenience as the spaces between coarse stones can be filled by smaller ones, resulting in a reduction in cement or mortar consumption (by mortar is understood the cement-sand mixture). For large foundations the author has used big boulders or cobble stones to advantage, but care has to be taken that the spaces between are well filled. Sharp stones are better than river gravel, which is rounded off by abrasion. Moreover, the stones have to be clean and not covered with moss or organic earth. Sand from the sea shore has to be thoroughly washed to remove all traces of salt. Sugar is still more detrimental, and cars used for the transportation of cement or sand should be well cleaned, when they have been used previously for sugar transport.

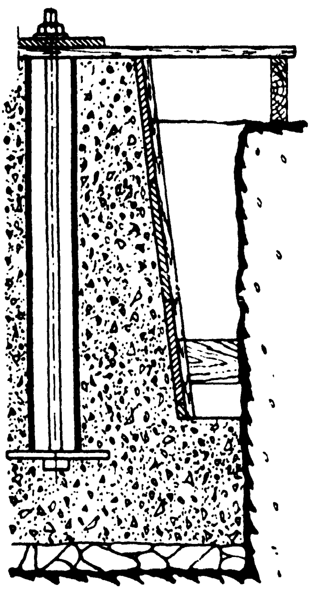


Fig. 108.—Foundation Bolts.

Mechanical mixing is generally better than hand mixing, and a small concrete mixer should be at hand at every sugar factory.

Bricks are not made in the same standard sizes all the world over, and local brick makers frequently have standards of their own or no standard at all. The amount of bricks needed has to be calculated by simple arithmetical procedure for a given volume, figuring on joints $\frac{1}{4}$ in. to $\frac{3}{8}$ in. thick. The thinner the joint, the better the masonry.

In the U.S.A. standard brick dimensions are $8\frac{1}{4}$ in. × 4 in. × $2\frac{1}{4}$ in. for building purposes; in Cuba the size is 11 in. × $5\frac{1}{2}$ in. × $2\frac{5}{8}$ in., whereas in Great Britain $8\frac{7}{8}$ in. × $4\frac{5}{16}$ in. × $2\frac{5}{8}$ in. is standard, and 9 in. × $4\frac{1}{2}$ in. × 3 in. is taken as the occupied volume of a brick with mortar. For Germany 25 cm. × 12 cm. × 6·5 cm. is common. Fire bricks are of different sizes and will be mentioned under "Boilers."

For foundation work hard-burnt bricks should be used, and for proper binding with the mortar bricks have to be laid wet.

A cubic yard of brick masonry will require about 0·3 cub. yd. of mortar and about 1400 U.S. bricks. Mortar can be made from a lime-sand or cement-sand mixture or a mixture of the three mentioned ingredients. In some tropical areas a very good nearly pure lime is found and should be employed when conditions allow one to do so. Portland cement mortar has a higher crushing strength than lime mortar, but for average purposes the following mixture has given good results :

1 volume cement, 1 volume lime, 3 volumes sand, 1 volume pure water

A working pressure of 175 lbs. per sq. inch on this mortar is allowable.

In *Fig.* 109 is shown a *Mill Foundation* the author had to lay for a second-hand set of two mills, bought by a Cuban mill owner. The soil had a rather low bearing capacity (blue clay of soft consistency), so a reinforced concrete slab of 12 in. thickness was laid, after the bearing capacity had been ascertained by a test block of several square feet surface and loaded with a heavy piece of machinery at hand to produce a specific bearing well over 50 per cent. above the normal one, and where the penetration had been measured at regular intervals for 48 hours.

On top of this reinforced concrete slab, which had been laid exactly horizontal, the wooden moulding was erected. A large tunnel was provided underneath and all the foundation bolts are within easy reach. The foundation plates each wear against two pieces of rail to increase the bearing surface.

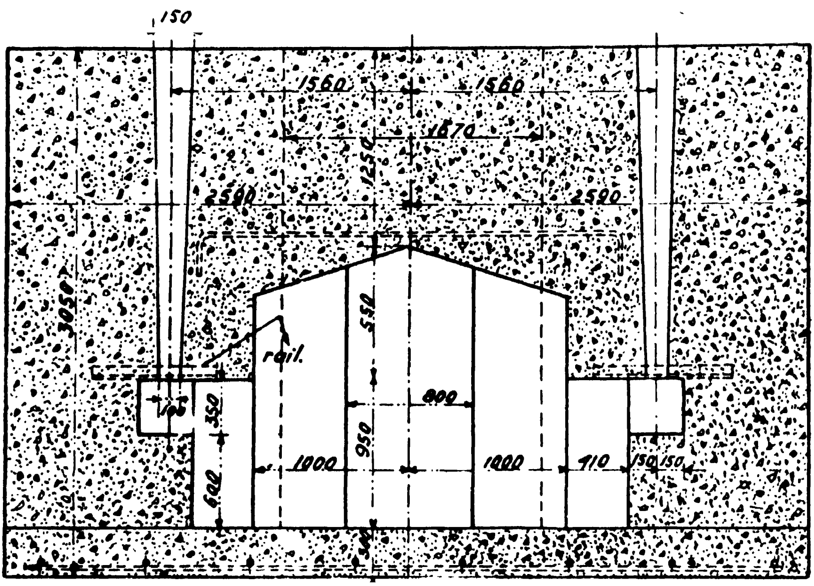


Fig. 109.—Mill Foundation on Soft Soil.

The thickness of the top layer, which has been reinforced, is of abnormal size on account of excavations on the top of the foundation.

The concrete foundation is laid to about 1¼ in. below the bases of the equipment to be placed on top. The machinery is erected on plates and wedge pieces, and grouting of a mixture of one volume sand to one volume cement, mixed with sufficient water to get the necessary consistency, is poured into this space. Care should be taken that no sugar enters the mixture, as it will prevent the cement hardening. As there is a shrinkage of the cement while hardening, the wedges on which the machinery has been laid should be removed after setting and the open spaces filled with cement.

The foundation of *Fig.* 109 proved to be dry and no sinking or cracking has since occurred.

For large foundations it is convenient to have all foundation plates on the same level, as it facilitates easy moulding. For the main bolts this advice should be adhered to by the designers of the machinery.

CHAPTER IV.

CANE UNLOADING AND CONVEYING.

Cane Rakes—Car Dumps—Cane Carriers.

When the cane arrives at the mill, it has to be transferred to the cane carrier at a speed regulated by the cane grinding capacity, and varying equipment has to be designed, as other factors, such as the length of the cut cane, have also to be considered.

For small mills and where native labour is cheap, the unloading of the cane cars or carts is mostly effected by hand. For capacities above 150 tons per 24 hours, a mechanical device should be preferred.

The simplest device is a trolley which will hoist the cane bundle from the cart, convey it until above the cane table or the carrier as may be the case and then drop it. The details for this kind of equipment are similar to those mentioned in Chapter II.

For larger capacities above 400 tons daily grinding a travelling or a revolving crane is very useful. These cranes have to be mechanically driven and the electrical drive is the most appropriate design for the purpose, as all the operations have to be done in a short interval of time and hand operation, therefore, will be too slow. The cane loads handled will be about 6 tons at most. Full length cane as well as short cut cane from 2 to 3 feet long can be equally well heaped up in bundles and hoisted by means of chain slings.

Where the cane is handled in full lengths the bundles have to be brought lengthwise alongside the carrier, and as the cane carts might come in at right angles to the carrier axis, a revolving device is sometimes provided on the hoisting hook of the crane, so that it will turn the load 90° by mechanical means.

Standard three-motor cranes using direct or alternating current are appropriate for this service and are employed in many sugar mills in different cane growing countries.

1.—Cane Rakes.

In *Fig.* 110 is shown a cane house with a *Reciprocating Cane Rake*. This apparatus is in use in a sugar factory in Central America and is of very simple design. The cane is cut in short lengths and the hinged doors of the cane cars are lowered by means of a couple of tackles, on arriving alongside the cane carrier, so that they will not suffer as would be the case were they simply dropped.

The rake proper is composed of two 4 in. tubes, hinged on the fixed end against a pair of wooden columns. The tubes are attached at the other end to a wooden block, about 5 ft. wide and provided with two iron tines *a*. Furthermore, a sliding block with tines *b* can move over the whole length of the tubes. The tines *a* are longer than those at *b*, so the fixed block and the tubes remain at a certain height above the car floor.

The operation of the rake is as follows : The right hand drum of the hoist, driven by a non-reversing steam engine, having twin cylinders, 6 in. dia. × 8 in. stroke, is engaged, when block *b* will slide to the end of its course, touching block *a* of the rake. The whole rake now is lifted, a cane car is run underneath and the hinged side lowered towards the cane carrier, which has its apron about 3 ft. below floor level. The right hand drum is veered at this moment and the tines of both blocks penetrate into the cane bundle. By engaging the left hand drum, block *b* is pulled, dragging the cane along, and this operation is repeated until the last sticks remain to be removed by hand. The carrier is not always loaded equally over the whole width and the left side carries more cane than the right side, which will cause uneven wear on the crusher and mill rollers. An equalizing device or a separate feed carrier is a convenient remedy in this respect.

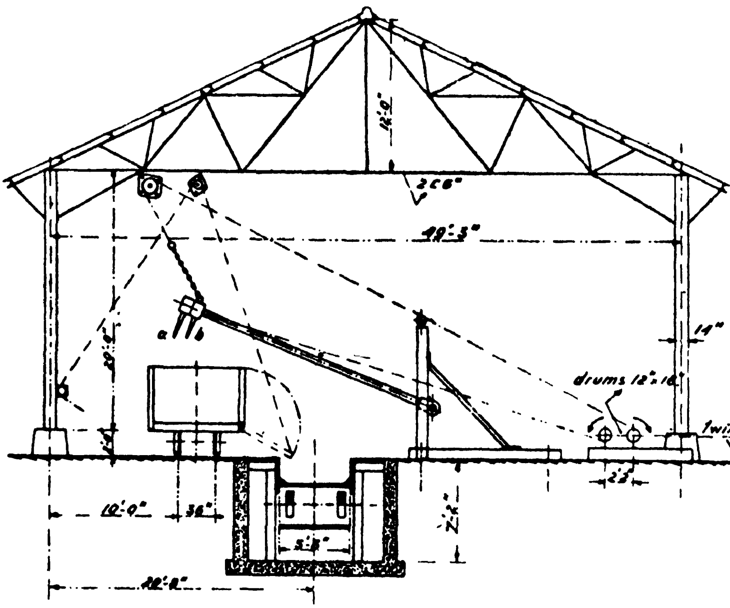


Fig. 110.—Cane House with Cane Rake.

It may not be superfluous to mention that, given improper handling by native labour, the car floors are apt to suffer from ramming by the tines *a*, as the wooden blocks have to possess a certain weight to achieve sufficient penetration into the cane bundle.

The outfit shown in *Fig.* 110 has easily handled 1000 tons of cane per 24 hours.

A variation from this reciprocating cane rake is the *Drag Carrier Rake*, composed of two sets of chain wheels, one revolving on a shaft at the fixed end of the rake and the other on a shaft on the outward end, whereon a chain is moving which has cross ties carrying tines—the so-called Hark type—which drag the cane from the carts. The free end of the rake is lowered and held in position by hand gear and the chain is power-driven by means of a steam engine or electric motor. The efficiency of this rotating rake is bound to be higher than that of the reciprocating one, as there is no dead movement of a return stroke, and a more gradual unloading is achieved.

In those countries where the cane is cut at full length and nearly straight, as is the case in Java, the unloading of the cane from the carts direct on to the cane carrier is not feasible ; so the cane bundles are hoisted on a cane table, which slopes towards the cane carrier. These bundles, moreover, are tied together by twine and this has to be cut on the cane table. From this latter the cane is transferred to the carrier by means of a rake.

A very ingenious *Self-contained Cane Rake* is shown diagrammatically in *Fig.* 111. The rake beam is made of two trough sections, welded together and carried in an eight-wheel guiding piece, which can be lowered or lifted by a screw arrangement *b*, which is attached to the rake guide at *c*.

The reciprocating movement is obtained by a calibrated steel roller chain, attached to the rake beam at *d* and *e* and operated by the sprocket *f*.

The operation of this rake is performed by two levers, which control two sets of reversing bevel gear clutches on the same shaft at *a* and *b*. The rake is driven by belt or by an electric motor. The power consumption is about 10 h.p. and the rake will handle 2000 tons of cane per 24 hours and in some cases more.

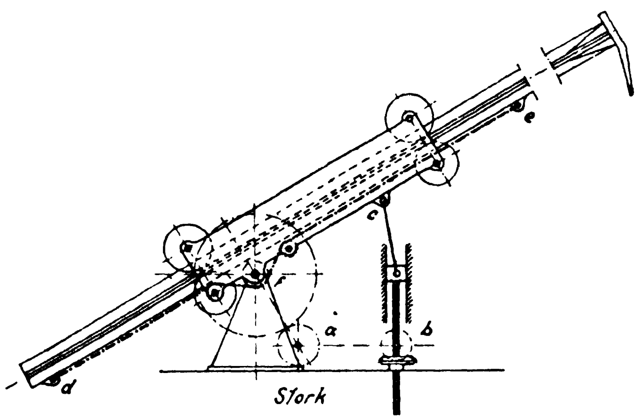


Fig. 111.—Self-Sustaining Cane Rake.

The carrier is not always equally loaded over the whole width and there is a heavy wear on the tines, but this rake is extensively used in Java.

A cross carrier about 20 ft. wide is also used in Java, attached to five or more strands of carrier chain. This rotating or travelling feed table moves at a speed of approximately 12 ft. per minute and requires about 10 h.p. The cane rake is unnecessary when the travelling feed table is applied, but unequal loading of the carrier is also inherent in this equipment.

2.—Cane Car Dumps.

To avoid the hoisting of the cane from the cars and the operation of a cane rake. a cane car dump which speeds up the whole cane unloading has been designed for average and heavy capacities. It can be applied where the cane is cut in short lengths of say 2 to 3 feet, as is the case in Cuba, and where cheap native labour is not available.

The first installations of mechanical unloaders of cane cars were built according to the principle sketched in *Fig.* 112, which shows a *Balanced Car Dump for End Discharge*. The designers haa in view a reduced power consumption, so the platform was pivoted at or near to the centre of gravity. In the most unfavourable case, only a tipping momentum $W \times c$ can occur, but as soon as the cane is dropped, the point of gravity is lowered and the equilibrium restored again.

Unfortunately very big excavations had to be made for both cane hopper and cane elevator, which is very costly when rock subsoil is encountered. Moreover, the cane cars cannot be divided into compartments and the lateral rigidity is only obtained by the end attachments of the cage, so they have to be built very substantially.

To overcome the first mentioned of these inconveniences successive designs had the fulcrum close to one of the extremes of the dump platform, so that the excavations would be less costly, but the power consumption was greatly increased, and expensive overhead hoisting equipment had to be built.

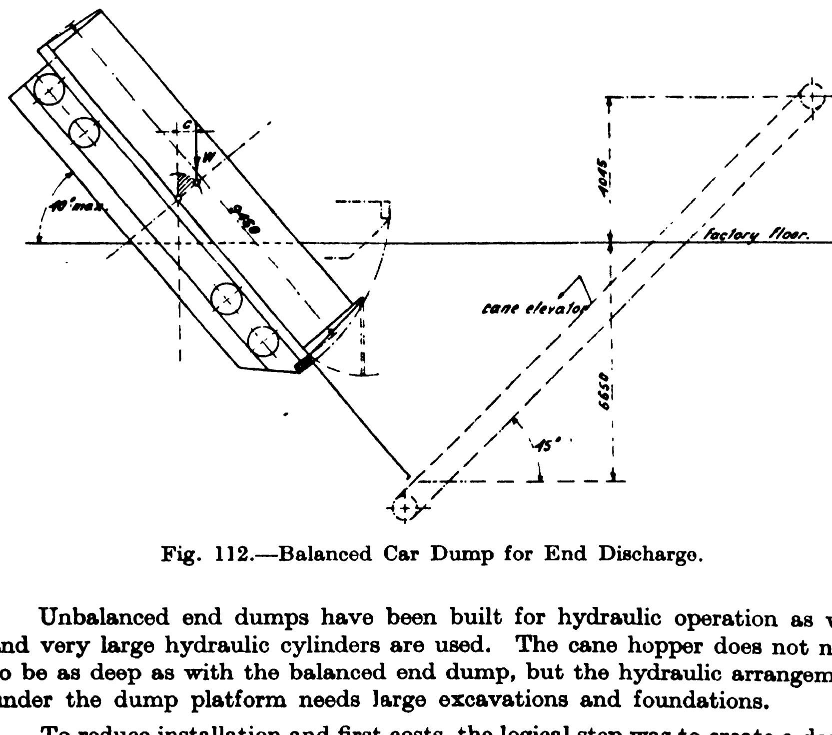


Fig. 112.—Balanced Car Dump for End Discharge.

Unbalanced end dumps have been built for hydraulic operation as well and very large hydraulic cylinders are used. The cane hopper does not need to be as deep as with the balanced end dump, but the hydraulic arrangement under the dump platform needs large excavations and foundations.

To reduce installation and first costs, the logical step was to create a design for lateral unloading and this has been successfully achieved, so it has come into general practice in Cuba, where very large quantities of cane up to 5000 tons per 24 hours are handled by one lateral car dump.

The design of such a *Lateral Car Dump*, which the author had to make, is shown in *Fig.* 113. This dump is operated by two single-acting hydraulic cylinders, having leather cup pistons 20 in. in diameter.

The equation of balance will be at any moment :—

$$W \times c = P \times 45 \quad \dots\dots\dots\dots \quad (50)$$

where W is the total weight on the dump, c the distance between the centre of gravity of the load W and the fulcrum of the dump, thus varying between 3 and 45 in. and P the force produced by the hydraulic pressure on the pistons.

When the dump starts tilting over, the force P will amount to $1/15\ W$, but this force will gradually increase, as the centre of gravity of the loaded car is moving along the curve rq, whereas that of the empty car is moving along the curve ts. It is obvious that the centre of gravity will remain within the area $qrst$ and because the moment of discharge is not exactly known and is dependent on the method of loading, maximum values have to be considered. There are, moreover, inertia forces which will increase when the operator is dumping the load abruptly and for which additional strength has to be applied in the construction of the details. With a total load of about 85,000 lbs., the maximum hydraulic pressure theoretically amounts to 135 lbs. per sq. in. The author has watched the performance in practice and under normal working conditions this pressure reached 175 lbs./sq. in., so a margin from 30 to 50 per cent. above the theoretical assumed value should be added for safety reasons.

The hydraulic piping has to be arranged for working pressures up to 250 lbs./sq. in. and is of very simple design. Both cylinders are connected by a $2\frac{1}{2}$ in. pipeline and midway between both a double branch is fitted, one from a small duplex pump $3\frac{1}{2}$ in. $\times$ $2\frac{1}{4}$ in. $\times$ 4 in. and the other to the pump suction tank of about 100 Imp. gals. capacity. In the pump connexion there is only a check or non-return valve, whereas the discharge pipeline is provided with a good size gland cock, so that the discharge flow can be regulated at will.

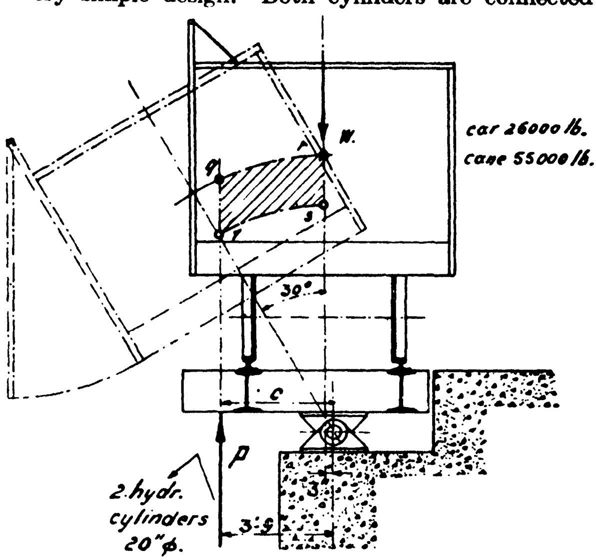


Fig. 113.—Lateral Car Dump.

The liquid should be preferably oil, but water or a water-oil emulsion can be used, when the hydraulic rams and the pump cylinders are brass equipped.

The cane is dumped on a separate feed carrier and the operator, standing on a platform over the carrier, has only three handles to manage. When the cane is dumped, the discharge cock is gradually opened, but not fully so, to cause wiredrawing and ensure smooth working. When the dump platform has to be lifted, the discharge cock is closed and the throttle of the steam pump opened. The pump has to be of sufficient working pressure to be able to lift the full car load on the dump. The third handle connects with the throttle of the steam engine, driving the feed carrier. These operations can be done at leisure and no special skill is needed.

One of the drawbacks to this kind of equipment was experienced at a factory where the hydraulic pipelines were emptied during the dead season. A locomotive wanted to take a short cut over the car dump, but was itself unfortunately dumped on—or rather through—the cane carrier. So during the dead season means should be provided to support the platform, or else it must be placed in a tilted position, making traffic over it impossible.

The power consumption is low and the number of labourers required for unloading the cane is reduced. It should, nevertheless, be recollected that when the cane cars are loaded too high, or the cane is stacked in such a way that a top load can be applied, the discharge does not take place spontaneously, but the twisted cane has to be pulled by hooks and a small operating platform is sometimes arranged on the opposite side of the carrier for this purpose.

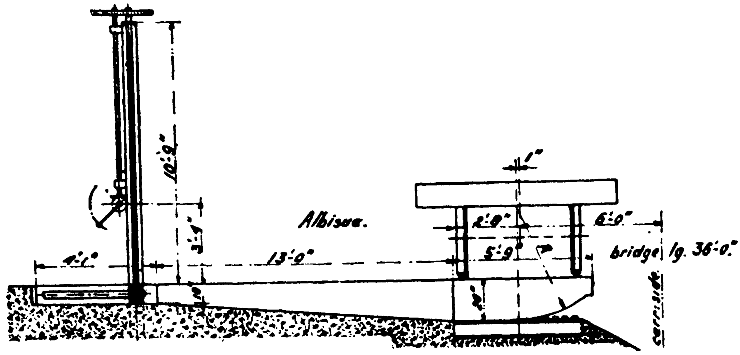


Fig. 114.—Variable Fulcrum Lateral Car Dump.

The maximum inclination of the platform is 30° and the whole operation of bringing the car on the platform, attaching the guy chains between the dump and the car platform, dumping, lifting and replacing the car, can be done within five minutes, so a very large capacity can be reached when necessary.

A lateral *Cane Car Dump with Variable Fulcrum* is shown in *Fig.* 114, where the fulcrum moves along nearly with the point of gravity towards the cane carrier, when the dump is operated.

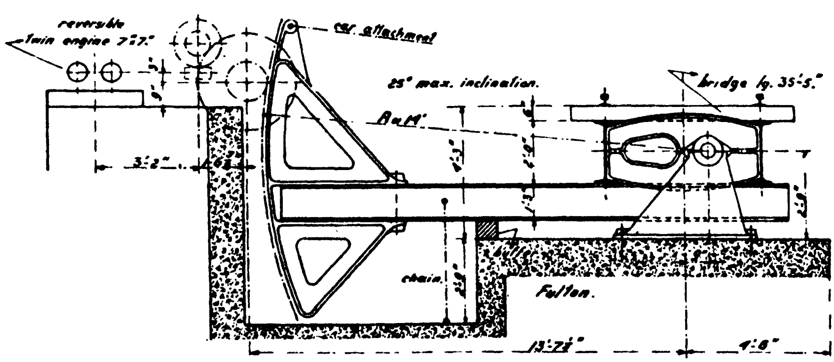


Fig. 115.—Steam Driven Lateral Car Dump.

An A-shaped lever attached to the side of the dump platform is operated by a vertically guided nut placed in a long horizontal slot and attached to a vertical threaded spindle, which is operated by hand gear. Two men can easily handle the heaviest cane car and the centre of the radius *R* will lie close to the centre of gravity of the empty car. For dumping a loaded car, the cane weight will be a positive dumping force and when the cane car is emptied, the dump is balanced again, and easily brought back to the horizontal position.

The vertical guides for the spindle nut have to be well anchored in the concrete foundation and a bracing is sometimes provided. The spindle thread has to be dimensioned for long wear as well as the nut, which is made of brass for out-door service.

The dump can also be arranged for power drive by an electric motor or by a belt from a steam engine or transmission close by. It is a very useful apparatus and for hand drive can manage about 2000 tons of cane per 24 hours.

A *Cane Car Dump with Fixed Fulcrum* and driven by steam is shown in *Fig.* 115; the platform fulcrum is 9 in. out of the gravity centre line at rest, so the loaded car will yield a momentum :—

$$M_0 = W \times 9 \text{ in.}$$

As soon as the centre of gravity is above the fulcrum the momentum is equal to zero and then becomes positive and only braking friction has to be overcome. When the cane load is dumped, the weight of the empty car only causes a small momentum until the centre of gravity is over the fulcrum again.

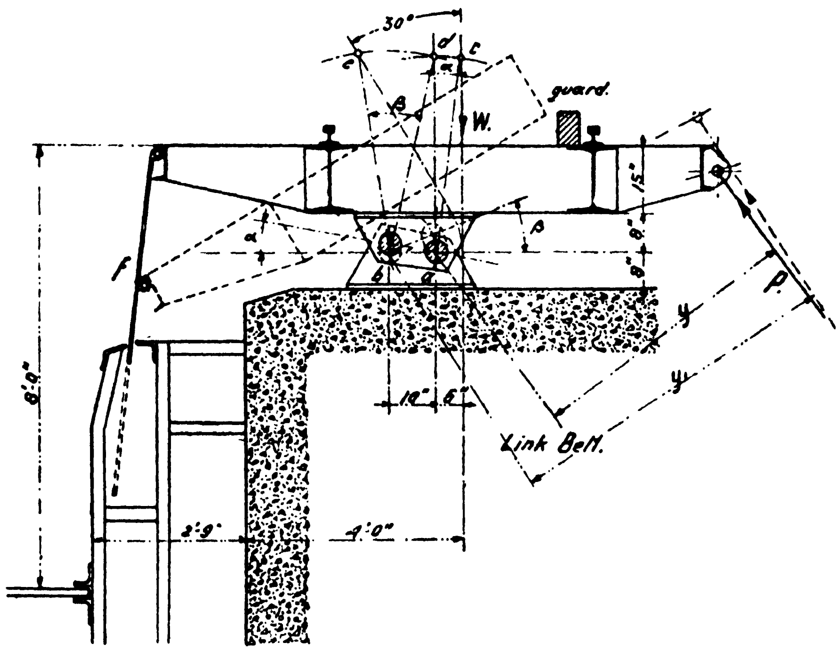


Fig. 116.—Double Fulcrum Dump.

The design, therefore, is one for low power consumption; the reversible twin steam engine with cylinders 7 in. dia. × 7 in. stroke operates the dump by way of a double reduction gear. The worm gear has machine-cut teeth, whereas the big segment has machine-moulded teeth.

A chain attached to the platform lever and a wooden buffer beam on top of the foundation provide the stop limits. The engine reversing gear has to be placed in mid position when the lifting extremes are reached.

Electric drive can be used instead of the steam engine, the motor being of the reversible type. The maximum output handled with a steam driven car dump of this type is as much as 5000 tons cane per 24 hours.

A construction with a double-acting hydraulic cylinder and having a *double fulcrum* is shown in *Fig.* 116. It is an intermediate design between *Figs.* 113 and 114.

In the horizontal position the platform rests on the pivots a, and a momentum prevails of the magnitude $W \times 6$ in. which has to be counterbalanced by the hydraulic couple $P \times y$.

When the centre of gravity has advanced from c to d, thus when the platform has acquired the sloping angle α of about 10°, the pivots a are released and pivots b will continue to carry the load. The momentum thus becomes $W \times 10$ in. and the hydraulic couple is increased to $P \times y_1$. By further advancing till the load starts to drop, the load momentum will decrease and 30° is the maximum sloping angle of the platform.

It should be mentioned that the cane cars empty completely on the lateral cane car dumps and only a few sticks dangling between the cage bars have to be removed by hand. As stated, the discharge at the start sometimes causes delay through improper loading.

The operation of the double-acting hydraulic cylinder has to be gradual as the top part has to be discharged, when pressure is applied under the piston or *vice versa*. The bottom and the top of the cylinder therefore have hydraulic pipelines to the pump or the hydraulic accumulator, and to the pump suction tank. Two inter-connected three-way cocks will suit the purpose, so there will be no wrong connexion. When an accumulator is used, generally of the sand or ballast loaded type, the pump throttle is automatically operated when the accumulator reaches its lowest position and stops when the highest position is attained.

To prevent cane sticks from falling between the carrier and the excavation walls, a hinged plate f is attached to the platform side towards the feed carrier.

Many of these lateral car dumps following different systems are in use and they have all given reliable operating performance. Selection depends on local conditions and first costs.

3.—Cane Carriers.

The cane carrier has to be designed for the cane it has to handle, and therefore in those countries where the cane is cut at full length as in Java, the construction necessarily has to be different from the one for countries where the cane is cut in short lengths of approximately 2 to 3 feet, as is the case in Cuba and many Latin-American countries. Moreover, the system of unloading has a marked influence upon the cane carrier construction.

In *Fig.* 117 is shown a *Java Cane Carrier* in actual use, having 5 ft. interior width between the carrier walls and designed for a grinding capacity of about 30 short tons of cane per hour. The carrier feeds a vertical crusher of the Krajewski type, 26 in. × 60 in., and the feed table and cane rake are located at a level flush with the top boards of the lower carrier part.

This carrier has the advantage of being well above the factory floor, thus making costly excavations and foundations unnecessary, and it, moreover, makes inspection and maintenance easy. It has wooden slats, and two strands of carrier chains, 40 in. apart, are well supported by guide beams and guide rollers, so that friction is reduced to the minimum. There is a horizontal or feeding part, with an upward slope of about 20°—in Java 22° is to be considered as the maximum slope, whereas in British design 16 to 18° is taken as standard

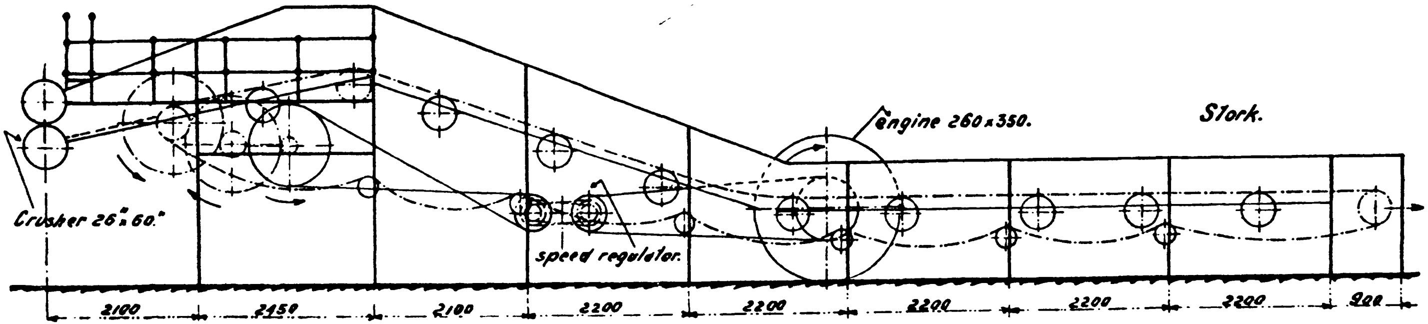


Fig. 117.—Cane Carrier (Java).

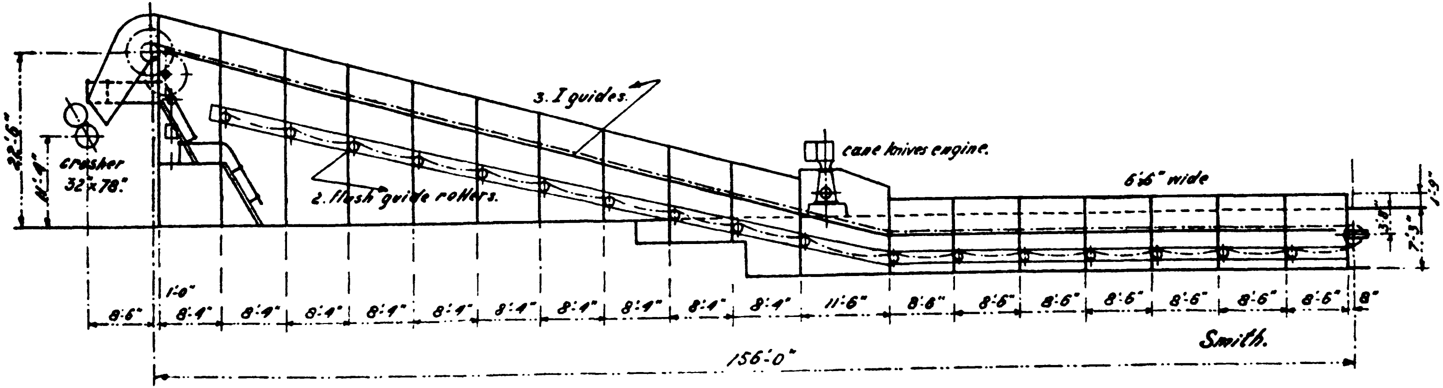


Fig. 117A.—A British Cane Carrier.

—and a downward slope of about 13°. It is obvious that the angle between upward and downward slopes has to be large (147° approximately in this case), as the long cane sticks do not allow any abrupt change of direction. The downward sloping serves the purpose of pushing the cane over the cane chute (which has the same inclination) into the crusher.

The driving of the carrier is effected by a horizontal single-cylinder steam engine with flywheel, having cylinder dimensions of 260 × 350 mm. which develops about 25 h.p. The hoisting machinery for unloading the cane is also driven by this same engine.

The cane carrier can also be driven from the lower crusher roll by a chain drive, in which case a friction clutch is arranged for this purpose on the main carrier shaft. This arrangement is only to be used in an emergency, as the cane carrier will have a speed proportionate to the crusher speed and as a matter of fact generally about 10 per cent. less to avoid chokes. With irregular feeding of the cane carrier, the crusher feed will be irregular also.

The return apron is carried on six sets of guide pulleys and a chain tightening arrangement is provided at the lower end.

The steam engine drives a speed regulator of the trapezoidal belt type by means of a belt, and the driving pulley of this speed regulator drives the double reduction gear of the carrier. This carrier has no abrupt changes in speed, which indicates that the loading is done quite regularly using native labour, which is contrary to Cuban practice, where special constructional details have to be embodied to produce this result.

It will be obvious that the repair cost of such a carrier as shown in *Fig.* 117 will remain low, as there is a nearly constant speed, few stops and no acceleration forces of importance acting on the chains.

The return apron does not move as regularly as the carrying apron and a jerky action takes place. This is due to the guide roller resistance and the sagging of the chain. As explained previously with *Fig.* 86 (ropeways) the sag or deflection has a direct bearing upon the stress in the chain and the bigger the sag, the smaller the chain stress. The convex bending of the carrier chains will increase with the sag and the wooden slats might touch each other, causing chain stresses several times higher than the normal ones. The happy medium, therefore, has to be sought for and touching the slats must be avoided.

The *power input* to operate a carrier can be derived as follows : Consider the length of the loaded apron of the carrier just mentioned as 50 ft., the width 5 ft., and the carrier speed 15 ft. per minute, then the height of the cane will be, at 30 short tons hourly capacity and 25 lbs. per cub. ft. cane weight :—

$$\frac{30 \times 2000}{60 \times 15 \times 5 \times 25} \simeq 0{\cdot}6 \text{ ft.}$$

The cane weight on the carrier, therefore, amounts to :—

$$50 \times 5 \times 0{\cdot}6 \times 25 = 3750 \text{ lbs.}$$

The weight of the carrier apron, having a total length of about 120 ft., can be figured at 10 lbs./ft. for each chain and 50 lbs. for the wooden slats, including the attaching bolts, making a total of 70 lbs. per ft. and for the whole apron 8400 lbs. The total friction weight therefore amounts to 12,150 lbs. The efficiency of this carrier is very low and from measured data the author has at hand, a friction coefficient from 0·5 to 0·8 has to be reckoned on, including the friction of the driving gear. The friction resistance will thus amount to :

$$\frac{12{,}150 \times 0{\cdot}8 \times 15}{60 \times 550} = 4{\cdot}4 \text{ h.p.}$$

and the formula reads :—

$$N_{fr} = \frac{W \times \mu \times V}{33{,}000} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (51)$$

The friction is high because proper lubrication of the chains is a difficult proposition. The heavy wear on the carrier pins and bushings indicates this very clearly.

Moreover, the weight of the cane has to be lifted about 7 ft. and at the rate of 60,000 ÷ 3600 = 16·9 lbs./sec. The necessary power input for this item, with a friction coefficient of 0·7 of the transmission gear, amounts to :—

$$\frac{60{,}000 \times 7}{3{,}600 \times 550 \times 0{\cdot}7} = 0{\cdot}3 \text{ h.p.}$$

and the formula reads :—

$$N_{lift} = \frac{W_{sec} \times H}{550 \times \mu_g} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (52)$$

It is seen that the friction resistance is far in excess of the net resistance caused by lifting the cane load. The total driving resistance in this case will amount to approximately 4·7 h.p. for which the driving engine is of ample size.

Another cane carrier, of a type in operation in the British Colonies, is shown in *Fig.* 117*a*, having three strands of roller chain 6 in. of pitch. The feed carrier is about 60 ft. long, whereas the sloping part with an inclination of about 15° will bring the cane which has just passed the knives well above crusher level, and a steep chute can be provided for efficient crusher feed.

The carrier is driven by a vertical twin engine, attached to the main carrier columns by means of double gearing. The returning apron is supported at short intervals by flush guide rollers, so as to allow a very reduced sag, and boards are attached on the inside of the columns for guiding the returning apron.

The spanning device is arranged at the horizontal end for taking up the stretch of the chain wear. On the upgoing carrier part, a set of revolving cane knives, directly driven by a vertical high speed twin-cylinder steam engine, will assist in proper cane preparation.

The small slope of the carrier will be beneficial for reducing chain stresses; it is a sound construction, especially when combined with revolving knives.

In *Fig.* 118 is shown a *Cuban Cane Elevator* in operation at a factory designed for 2250 short tons of cane per tandem per 24 hours but actually grinding 3400 short tons on this tandem. The elevator receives the cane from an unbalanced end-discharge dump and an excavation of 26 ft. 3 in. had to be made. The inclination is 37°, which is low compared with other elevators, where inclinations up to 60° with the horizontal are in use. These steep elevators, nevertheless, have to possess projecting arms beyond the slat surface to hold the cane, else it will slide backwards. This explains why cane elevators need to have chains with a large pitch, generally about 12 in., although in the elevator shown the pitch of the steel roller chain is 9 in.

The rising apron is supported on two 6 in. joists, 4 ft. centre to centre, whereas the falling apron is supported on the slat side by two rails of 25 lbs. per yard section. This support of the downgoing apron is very favourable to the chain wear and the reduced friction resistance of the carrier. The chain slack is taken up by a tightening arrangement at the bottom end of the elevator.

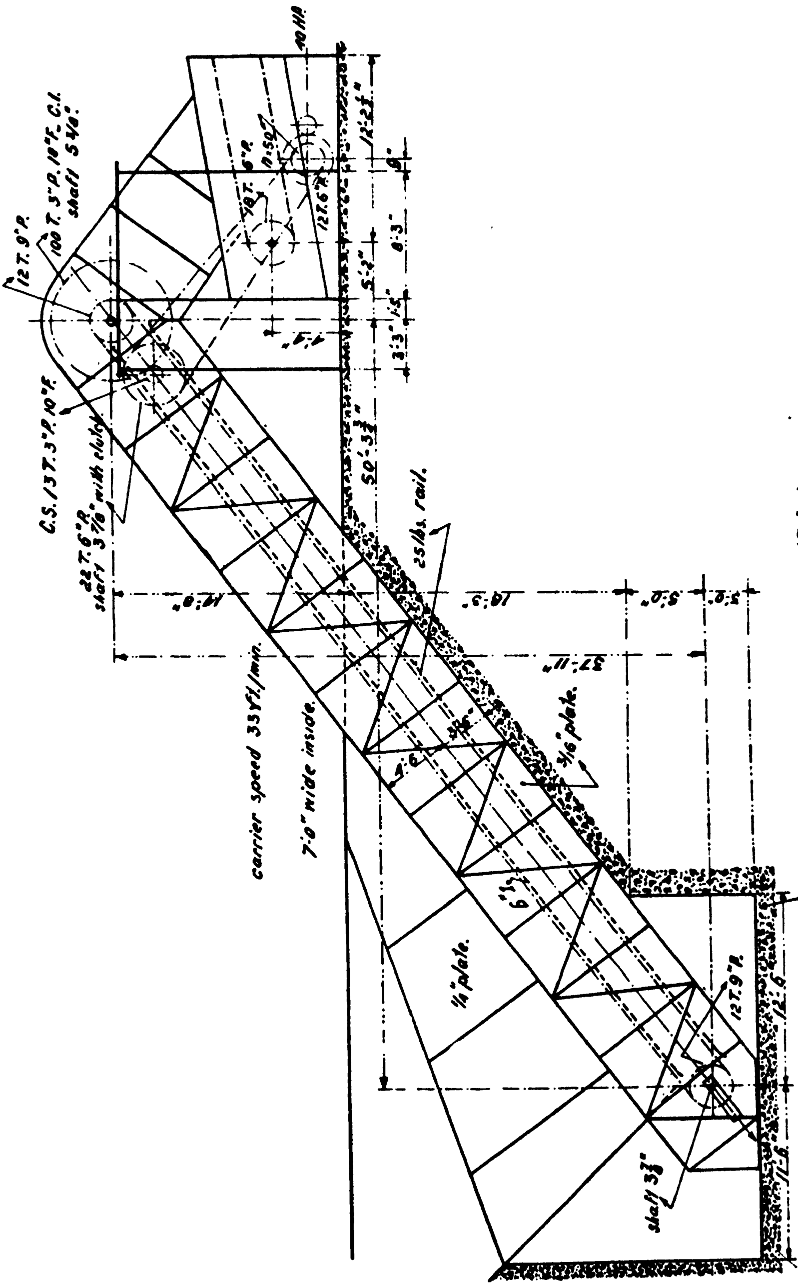


Fig. 118.—Cane Elevator (Cuba).

The sprockets have 12 teeth and are made of cast iron. The author has supplied many cast steel sprockets, having teeth completely machine-cut. These wear better than machine-moulded teeth.

The cane hopper is made of $\frac{1}{4}$ in. steel plates reinforced with angle irons, whereas the elevator walls are of reinforced $\frac{3}{16}$ in. plates. The elevator unloads on the cane conductor, which brings the cane to the crushers. The maximum speed of the apron is 33 ft. per minute, but normally the elevator is not run over 20 ft. per minute. Following the calculations as per formulae (51) and (52), the results are :—

Average height of cane on elevator	1·35 ft.
Average weight of cane on elevator	14,150 lbs.
Weight of apron	9,100 lbs.
Average friction resistance	11·25 h.p.
Average lifting power	7·5 h.p.
Average total power input	18·75 h.p.

The acceleration forces are so small that they can be neglected. It should, moreover, be recollected that the starting resistance is considerably higher than the average running resistance. As the engine is operated by a throttle, wiredrawing will take place and full boiler pressure will not act on the pistons. This explains why the engine h.p. is generally taken as double the calculated one.

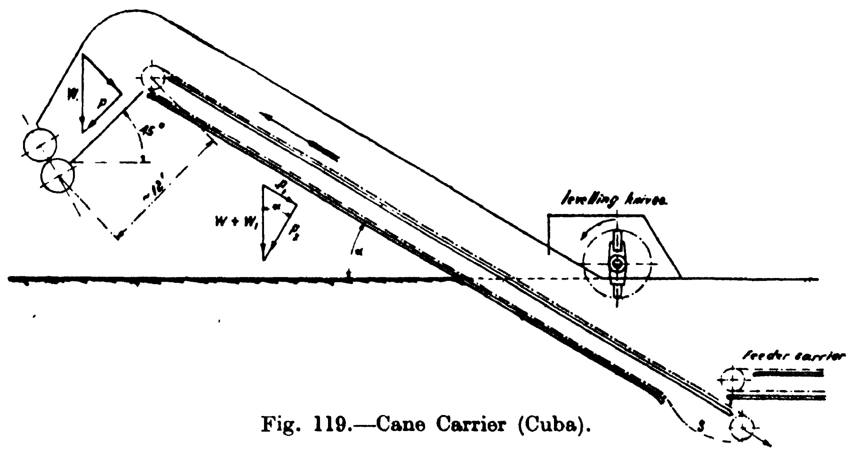


Fig. 119.—Cane Carrier (Cuba).

In *Fig.* 119 is shown the standard arrangement of a *Cuban Cane Carrier with Feeder Carrier*. The inclination angle α has been executed up to 35°. The cars are unloaded on a lateral or side car dump and it will be obvious that the feeder carrier sometimes is covered with heaps of cane far in excess of the normal cane carrier load and with empty spaces between. The feeder carrier, therefore, has the definite purpose of distributing the cane load evenly on the cane carrier going to the crusher. The feed carrier has to have a maximum speed of about 35 ft. per minute, as the gaps between the cane heaps should be traversed as quickly as possible.

The feed carrier is driven by a non-reversible twin cylinder engine. The twin arrangement is necessary as the engine has to pick up at any position. Electric drive is also used in many instances and the motor has to have a high initial torque, and slip ring motors or special squirrel cage motors are appropriate when A.C. current is available or a series wound motor for D.C. In case of a steam engine a governor for maximum allowable speed is sometimes provided, but this is not common practice, as the operator has the throttle under his control. Moreover, these engines should have a good cylinder drainage, as they are located at the end of the factory steam lines and often stopped and started, so the presence of condensed water is not illusory. Friction clutches are not used for this hard service. A double reduction gear is provided with a silent chain drive. The sprockets of the latter drive are of cast or mild steel and have to have machine-cut teeth. Cast steel sprockets for the carrier chains with machine-cut teeth are also used to advantage.

The slats are generally of wood, 2 in. thick, as they are subject to rough service and three strands of chain should be provided with guides of I-joists under the rollers, so that they can withstand the impact of the falling cane. The chains are—of course—of the same pitch, but should also be of the same construction for even wear. The author has come across instances where different chains were used, and this has caused considerable trouble.

The sides of the feeder carrier slope about 10 to 15° with the vertical and the transfer from the sloping walls to the vertical walls of the upgoing carrier should be made very smooth, as otherwise the cane will choke at this spot.

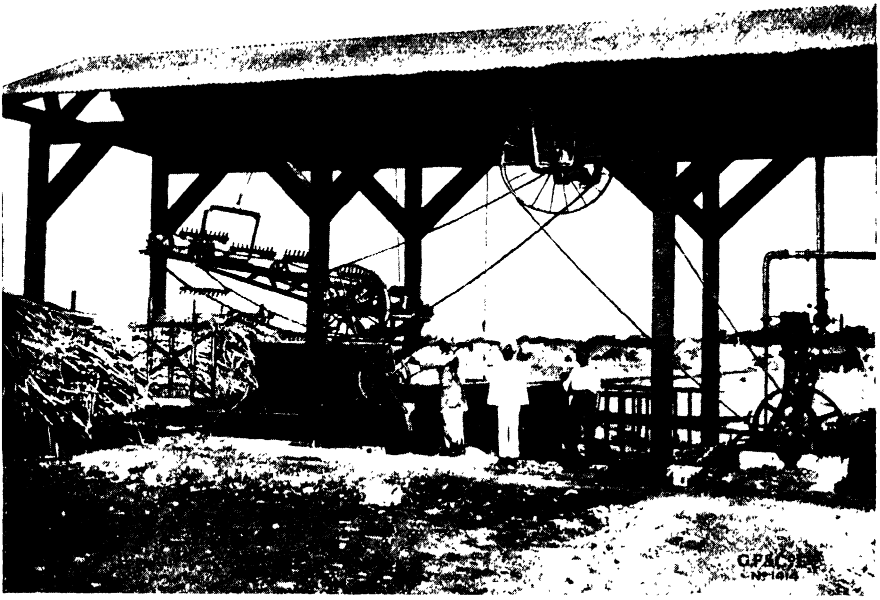
The return apron of the feeder cannot be supported on sliding guides, as the weight reaction to overcome the sliding friction is zero on the horizontal line, and guide rollers have to be used. As the chain strand bends, when passing over the guide roller, there is a sliding friction on the roller surface and as the load is proportionate to the guide roller distances, these should not be made too large, as the slats will soon indicate any heavy wear upon these. A chain tightening arrangement is applied at the free end of the feeder carrier.

The cane elevator has also three strands of chain and the sprockets on the shafts have to be arranged in such a way that the tooth flanks of the three sprockets on each shaft are exactly in line. The upgoing apron is supported on joists as mentioned before and the downgoing apron is also provided with them. Sometimes wooden beams are used for these guides. As the top part is subject to wear, it is good practice with wooden or steel guides to have a renewable flat iron on the top flange. For wooden beams this will be a necessity.

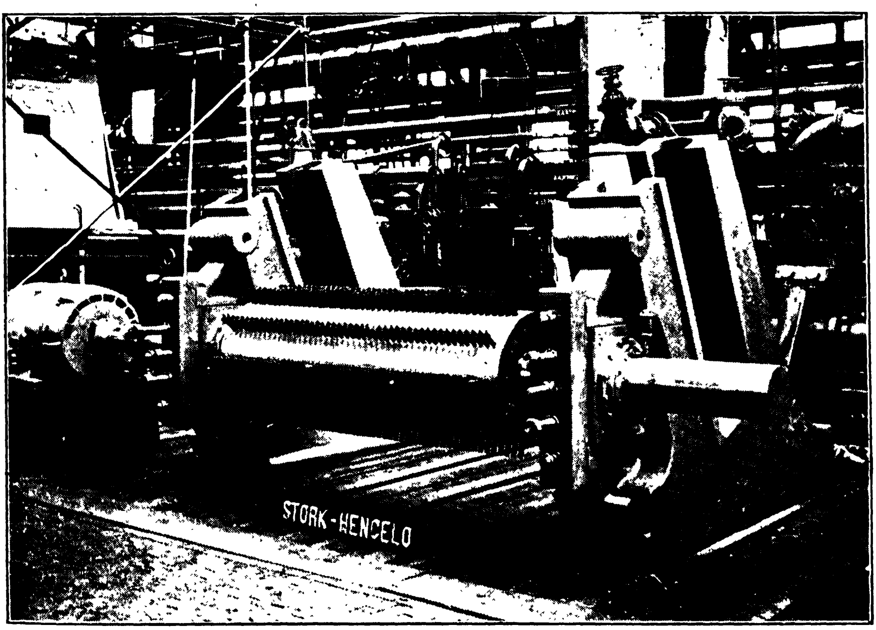
The sag S should be at the lower end of the cane carrier, as the weight component will be sufficient to overcome friction on the return guides and the total friction of the carrier will be reduced by this construction. The upgoing apron obviously does not jerk on the lower sprocket and smooth running is achieved.

A set of levelling knives is also shown in *Fig.* 119; these are applied as a further means of equal distribution of the cane on the carrier. The direction of rotation has to be opposite to that of the carrier movement, so that the excess cane is thrown backwards. Instead of levelling knives, a heavy shaft with attached arms about 6 in. apart, helicoidally arranged around the shaft, is sometimes used. It is obvious that the knives not only throw the canes back, but actually cut these, and the levelling action is therefore improved on. The levelling knives and especially the so-called "gallego" (the Cuban name for a revolving shaft with arms) increase the carrier resistance to a certain extent, to be calculated from the spherical force, acting on the arms or knives.

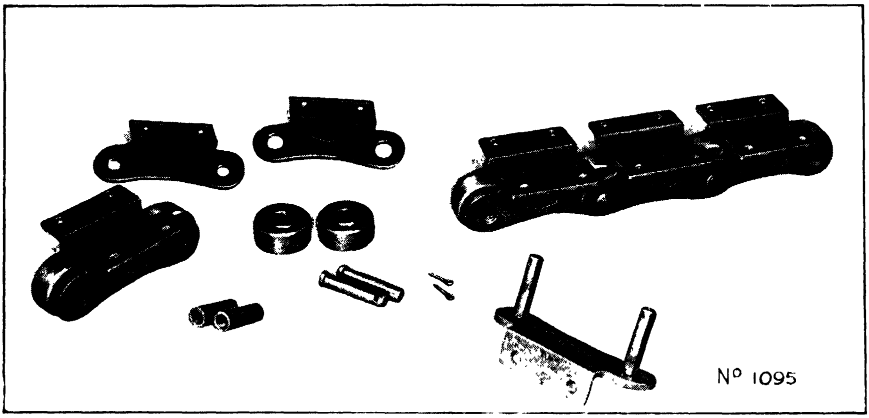
The cane is dumped into the chute to the crusher and as it is cut in short lengths there is no inconvenience in so doing. The chute angle is generally at 45° to the horizontal and a length of 12 ft. or more (according to the head room above) is allowed.



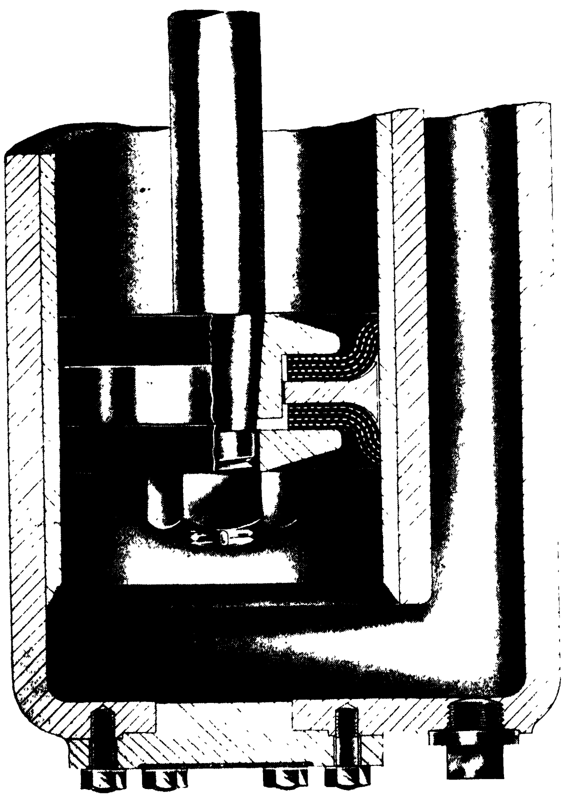
DRAG CARRIER RAKE FOR UNLOADING CANE.
(*Geo. Fletcher & Co., Ltd.*)



CRUSHER HOUSINGS WITH MAXWELL SHREDDER.
(*Gebr. Stork & Co.*)



All-Steel Machined Cane Carrier Chain.
(A. & W. Smith & Co., Ltd.)



Hydraulic Ram Cylinder with "S. E. A." Ring Cups.
(Ronald Trist & Co., Ltd.)

Taking the chute with cane filled over a length of 10 ft. and 3 ft. average height, each foot of roller length of the crusher will receive a ***feeding pressure***, which is of paramount importance, when heavy amounts of cane have to be ground.

From *Fig.* 119 at the chute end, W is taken as the cane weight in the chute per foot-roller length, thus :—

$$' = 10 \times 3 \times 1 \times 25 = 750 \text{ lbs.}$$

The component P parallel to the chute bottom at 45° has the value :—

$$P = W \div \sqrt{2} = 0{\cdot}7\ W = W \times \sin \beta$$

and as the friction coefficient between the cane and the flush chute walls is low, the feeding force will amount to about 0·6 W and the formula can be written :

$$P_{feed} = L \times H \times 25 \times (\sin \beta - \cos \beta \times \mu) \ldots\ldots \text{(53)}$$

and for $\beta = 45°$

$$P_{feed} = L \times H \times 25 \times \cos \alpha \times (1 - \mu) \ldots\ldots\ldots \text{(53a)}$$

where : P_{feed} = feeding pressure per foot roller length.
L = cane covered chute length.
H = average height of cane in chute.
β = sloping angle of chute ($\alpha = 90° - \beta$).
μ = friction coefficient (about 0·15).

The *chain stress* needs also to be investigated. As a matter of fact, the 6 in. pitch chains on these carriers have often been ruptured and manufacturers have gradually increased the ultimate strength from 39,000 lbs. about 20 years ago, to 100,000 lbs. to-day. Of course, the increased capacities call for heavier chains, but operating engineers prefer additional strength in this detail of the carrier as the breakage of a chain on a cane-loaded carrier is no small nuisance.

The apron is considered to be balanced, as both upgoing and downgoing parts are nearly of equal weight, but a stress equal to half the apron weight prevails at the upper sprockets.

From *Fig.* 119 let us take W, the weight of cane on the carrier, and W_1, half the apron weight, then the net stress in the chains derived therefrom will amount to :—

$$P_1 = (W + W_1) \times \sin \alpha$$

when α is the sloping angle of the carrier.

As already mentioned, the acceleration forces can be neglected but the friction stresses are considerable.

The friction coefficient of the apron, that is without the driving gear, has to be taken at about 0·6, and the friction stress will amount to :—

$$P_2 \times 0{\cdot}6 = (W + W_1) \cos \alpha \times \mu$$

and the total chain stress will amount to :—

$$P_{chain} = (W + W_1) \times (\sin \alpha + \cos \alpha \times \mu) \ldots\ldots \text{(54)}$$

From a cane carrier designed by the author, the cane weight amounted to a maximum of 27,000 lbs., the half apron weight to 6400 lbs. and the sloping angle 22°. The chain stress therefore proved to be 31,100 lbs., and three strands of chain of 39,000 lbs. ultimate strength each had to be used, giving a safety factor of 3·76, which is a low figure, so heavier chains had to be provided a few years later.

The cane carriers are often driven from a twin steam engine vertically arranged against the main carrier columns ; this is a very good arrangement, as no additional floor space is needed for this engine and the throttle connexion to the operating platform above is short. Moreover, a chain drive from the engine shaft is not needed, the latter being provided with a driving pinion and triple reduction gear.

4.—Cane Carrier Details.

The standard design of a 6 in. pitch *Cane Carrier Chain* is shown in ***Fig.*** 120. The links of the heaviest chain are now $\frac{1}{2}$ in. thick and the pins 1 in. in diameter. These chains are punched or completely machined. The machined method is more expensive, but gives the strongest chain, as incipient shearing stresses or microscopic cracks are inherent in the punching method. The latter is now almost completely superseded by so-called heat treatment or annealing. The pitch is made in both cases to a tolerance of $\frac{1}{8}$ in. per 100 links.

The pins are nowadays made of high shearing special alloy steels, which are case-hardened, whereas the bushings, which are split, when manufactured by the stamping method or completely machined in the other case, are of case-hardened high carbon steel. The links are of ductile mild steel of high tensile strength.

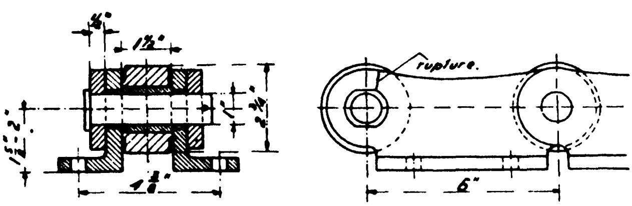


Fig. 120.—Cane Carrier Roller Chain.

The strength of the chain depends on the material used and good wearing qualities should be embodied. Most of the breakages occur on the inner links as shown in the drawing and the spot indicated is the weakest of the whole chain on account of the large bore for the bushing. The pin bearing is also relatively small, the projected bearing area of the heaviest chain not being over 1 sq. in. The pins do not rotate in the links, but in the bushings.

When lubrication is applied, the pin is sometimes provided with a centre bore, but this is not general practice. For cleaning the chain, a small steam or compressed air jet is arranged over the returning strands and a lubricating dripper is sometimes fitted. The average life of the pins is 4 to 6 crops, or in some cases longer when under less heavy service conditions.

A very recent design is shown in ***Fig.*** 120a, where the chain links are made from special angle sections, so as to allow for a heavier link section, without increasing the thickness of the flange attachment. The chain is completely machined and no punching or bending of the material is required, the links being profiled by a patented method of oxy-acetylene cutting.

The links are drilled and reamered to jig, and the material used is of high quality, having a minimum tensile strength of 100,000 lbs. per sq. in. The bushes are case-hardened, forced into the inner links with a 0·001 in. fit, being ground to exact dimensions. The pins of special material have 123,000 lbs. tensile and a corresponding high shearing strength, and are forced into the outer links. The rollers are also case-hardened and provided with an oil chamber for retaining the lubricating oil.

The 6 in. chains mostly used for cane carriers are made in three sizes, having ¾ in., ⅞ in, or 1 in. pins for ultimate chain strengths of 50,000, 75,000 and 100,000 lbs. respectively, all rollers having 2¾ in. outside diameter and the links up to ½ in. thickness. The chain, drawn in *Fig.* 120a, is of the 50,000 lbs. type.

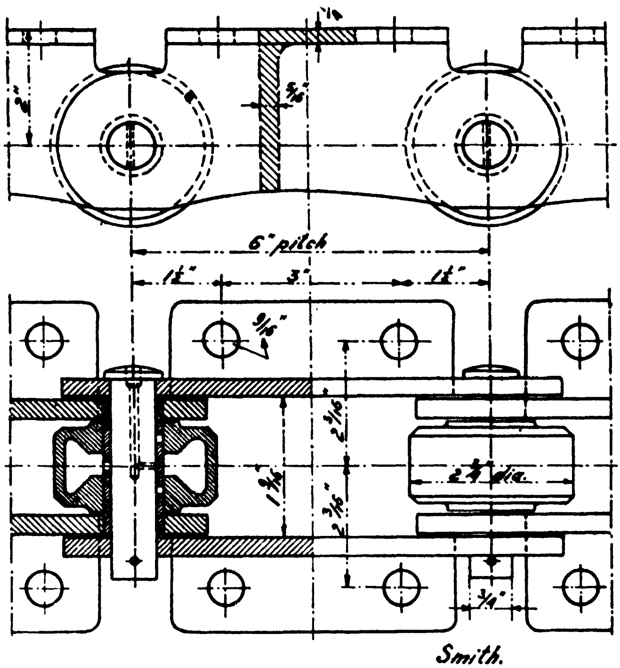


Fig. 120A.—Cane Carrier Chain.

An *Overlapping Corrugated Steel Slat* for cane carriers is shown in *Fig.* 121. The slat has a corrugation in the centre for increased strength and the overlapping is curved, according to a radius having its centre in the pin centre. On the left end the slat is bent at right angles with cuttings at the place of the three chain strands, which gives additional strength at the locus of overlapping, ensuring a good performance of the hingeing action.

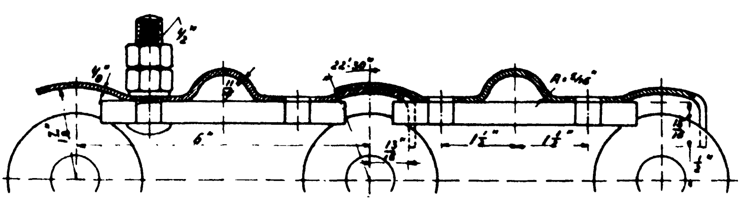


Fig. 121.—Corrugated Steel Slats.

It will be unnecessary to mention that the apron has to move to the right. The long projecting bolts for attachments to the chain links will help to hold the cane on top and prevent slip; sometimes pieces of angle iron are attached to the slats for this same purpose. The overlapping has the great advantage that little or no dirt from the cane can fall on the carrier chain, which materially increases the life of the latter. For shredded cane the overlapping is of paramount importance, as also where revolving knives are used. The reinforcement below the overlapping is of special advantage where the knives mentioned are mounted over the carrier, as the action of the knives causes an impact on the carrier slats.

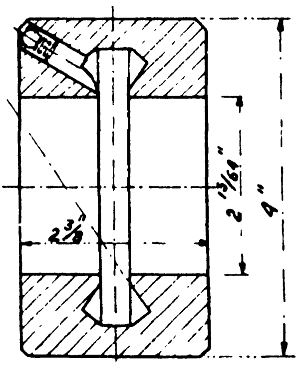


Fig. 122.—Oil Chamber in Roller.

In *Fig.* 122 is shown a *Chain Roller with an Interior Oil Chamber* for a 12 in. pitch chain for a cane elevator of heavy construction. The oil chamber is produced by a triple cut in the mild steel roller bore. This chamber is filled with heavy oil or grease by means of a grease gun through the spring-closed oil hole. This hole is at a position where there is no load on the roller and is easily accessible. Care has to be taken that the oil holes are on the outside of the carrier for accessibility. This method of chain lubrication is to be preferred over the oil hole in the pin and will be as cheap.

The author has had unfavourable experience with rollers of malleable iron having the oil chamber cast in. They have broken under operating conditions and have had to be replaced by the mild steel ones referred to above.

In *Fig.* 123 is shown a *Main Sprocket* for a cane carrier. This sprocket has to have the same pitch as the chain ; driving and driven sprockets have ere now been designed with a different pitch, the former smaller and the latter larger than the chain pitch, but this practice has been discontinued. When

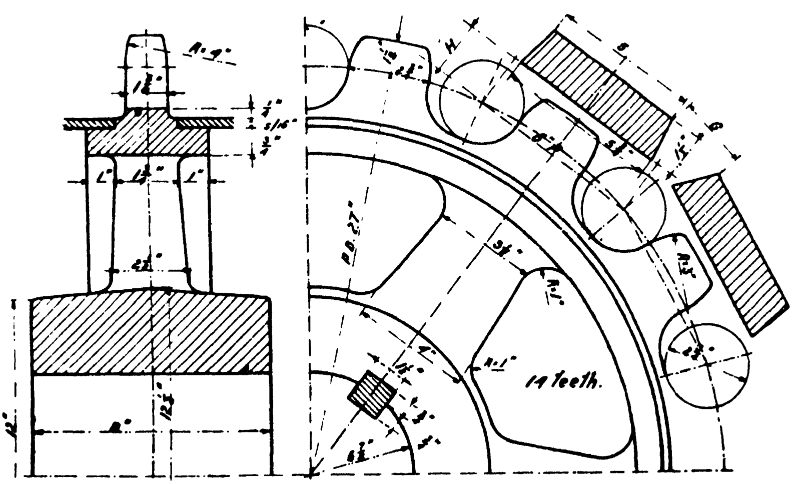


Fig. 123.—Main Sprocket.

wear occurs on the chain, the pitch will increase and the chain will mount on a larger pitch diameter. The sprocket allows an increase in pitch of about 5 per cent. or $\frac{5}{16}$ in. before top mounting is to be feared. Generally, new pins and bushings will restore the original chain pitch.

The shaft is $6\frac{7}{8}$ in. in diameter, reduced by the $\frac{3}{4}$ in. keyway. The American type of square key has been used, having a large lateral bearing. As the shaft material generally will allow a larger bearing than the sprocket material, the keyway in the shaft can be less deep. Flat keys, as used on the European continent, will do equally well. The bore should be of a good fit and a maximum tolerance of 0·002 in. should not be exceeded. The tapering of the keys should be $\frac{1}{8}$ in. per foot or 1 per cent. and the tapering should be applied in the bore.

From the drawing it will be seen that the gap G between the wooden slats will increase when the chain height above centre H increases. The inconvenience of wooden slats exists in the fact that small cane sticks, etc., may be caught between the boards and by convex bending or straightening may cause heavy stresses in the chains. This applies especially in those countries where the cane is cut in short lengths. Overlapping wooden slats are not used to any big extent because these will not stand the very heavy service expected of them when used on feeding carriers. The overlapping steel slat has a great advantage in this respect.

Between the sprockets is mounted a steel drum of $\frac{5}{16}$ in. plate material, butt-welded on the seam ; this serves the purpose of preventing canes winding or twisting around the shafts. For overlapping steel slats, these drums are omitted.

CHAPTER V.

DEFIBRATORS AND DISINTEGRATORS OF THE CANE.

Revolving Knives and Shredders.

The extraction of sucrose is a function of de-fibration or disintegration of the cane tissue, thus the proper extraction of the juice is only possible when this tissue is ruptured to the greatest possible extent and the juice-holding cells are opened up. As a matter of fact, repeated pressings between crusher and mill rollers will produce this defibration and the cane will be disintegrated to an advanced degree. Nevertheless, increased grinding capacities and the limited number of rolls in the milling train put a limit to defibration by roll pressure and it is a logical evolution that an economical preparatory arrangement has been sought, as the pressing rolls then are partly relieved from the defibrating performance and can be used to the fullest extent for extraction purposes; and so that this extraction can be enhanced by repeated imbibition or soaking of the bagasse blanket, it will be obvious that the sooner the bagasse blanket is fit for receiving this maceration or imbibition, the better the leaching process will be achieved. This indicates clearly why the defibration has to be completed at an early stage or before the milling performance. On the other hand, it is obvious that the advantages of disintegrating might be neutralized, where proper maceration or imbibition is not applied; and contrarily, advantages might be exaggerated, when the increase in extraction is not only due to the defibrating action of the revolving knives or shredders, but, to a big extent, to improved grooving, and better application of imbibition.

1.—Revolving Knives.

Revolving knives will be treated first because they are always arranged in front of the pressing rollers, be these a crusher or a three-roller mill.

The actual purpose of the revolving knives is to cut the cane into small chips, so that the hard rind and nodes are broken up and the soft pith, which contains most of the juice, will be more effectively treated by the following crusher or mill pressure.

For those countries where the cane is cut in full lengths and laid parallel to the longitudinal axis of the cane carrier, the cutting effect is handicapped, as the distance between the revolving knives generally is not so close that each cane will be cut, and therefore, though their application is limited in such countries, small pitch revolving knives are here to be preferred.

On the other hand, where the cane is cut in short lengths, full advantage is to be derived from revolving knives, as the rind is efficiently broken up. Only the bottom layer of cane, which can wholly escape under the clearance between the knife points and the carrier slats, is not cut, but serves as a mattress for the chopped cane and assists in dragging it through the crusher, an advantage in those cases where a long chute in front of the crusher is not available. Of late, revolving knives have found application to a big extent, as not only will the extraction increase by proper selection, but also the capacity of the milling station will be raised by about 25 to 30 per cent. It should, nevertheless, be recollected that, with the installation of revolving knives, the mill settings and the grooves generally are revised also, so the increase in capacity may not be attributable to the knives alone.

The original type of *Knives* is shown in *Fig.* 124, measured on an existing installation by the author. These knives are placed 8 in. above the carrier slats on a 7 ft. carrier, for a grinding capacity of 1400 metric tons of cane per 24 hours.

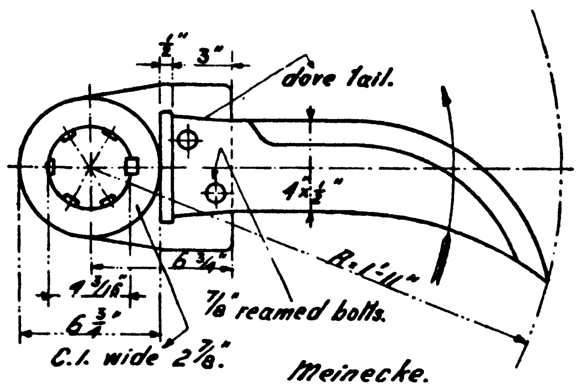


Fig. 124.—Original Type of Knives.

The knives are run at about 100 r.p.m. or a peripheral speed of only about 20 ft. per second. The work done by these knives has been favourable, and for the time being the mill concerned ranked high in sucrose extraction.

The hubs are of cast iron and the blades are long, so a heavy impact has to be resisted by the hubs. Some 28 blades are placed in a helicoidal line around the shaft. It is obvious that for larger capacities and higher speed this construction would prove too weak and it has therefore been discontinued.

A second type, of which there are many installations, is shown in *Fig.* 125. These *Swinging Blades* are pivoted to cast steel hubs on the shaft and the principal advantage lies in the easy removal of the blades. The sideward swinging action must prove illusory when running at full speed as the centrifugal force of well over one ton for each knife will need quite an obstacle to divert the knife from its straight course.

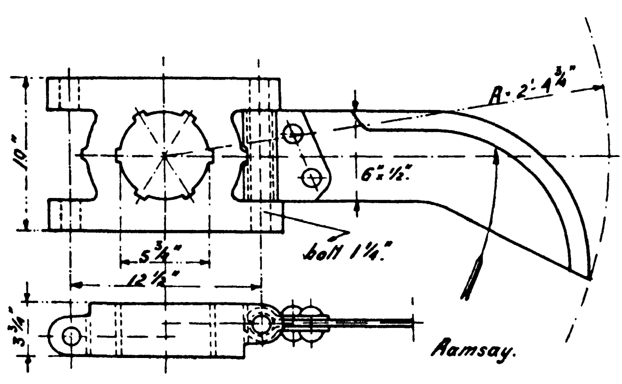


Fig. 125.—Swinging Blades.

These knives are mounted over a 6 ft. cane carrier, running at an average speed of 21 ft. per minute and have about 3 in. clearance from the carrier slats. There are 36 blades and at 500 r.p.m. a peripheral speed of about 125 ft. per second is obtained. The grinding capacity is about 1500 metric tons cane per 24 hours and the power input amounts to 75 h.p. or about 1·2 h.p. per ton cane ground per hour.

The number of blows per second obviously is :—

$$500 \times 36 \div 60 = 300 \text{ per second.}$$

As the mean power input is 75 h.p. or 75 × 550 ft. lbs. = 41,250 ft. lbs. per second, for each blow there will be available an average momentum of 41,250 ÷ 300 = 137·5 ft. lbs. With an average radius of about 27 in. or 2·25 ft., the average impact or force of each blow amounts to 137·5 ÷ 2·25 = 61 lbs. The formula for the force of the blow may thus be written :—

$$P_{blow} = \frac{N \times 550 \times 60}{i \times n \times R_m} \quad \ldots\ldots\ldots\ldots (55)$$

where N = power input in h.p.
i = number of blades.
n = number of revs. per min.
R_m = mean cutting radius in ft.

With n = 550 and a mean cutting radius of 2 ft., the formula will be reduced to :—

$$P_{blow} = \frac{30\,N}{i} \quad \ldots\ldots\ldots\ldots (55a)$$

The force of the blow required depends on how far the knife has to penetrate into the cane blanket and especially upon the sharpness of the knives. Blunt knives will increase the power consumption with no advantage to the extraction. In a case which came to the author's notice, the knives were left blunt, to achieve more of a hammering or splitting effect than a straight cutting one ; the result was a higher power consumption and broken hub pieces. Any appreciable increase in extraction was not noted.

When hitting a hard obstruction, the knife has to cut through it or else it will break, or fracture the hub ; the hub pieces of *Fig.* 125 have shown an average good performance, except for the blunt knife experiment mentioned above.

As the knives are staggered on the shaft, a blow will take place each full revolution, as the diametrically opposed blades do not hit on the same spot. At 500 r.p.m. the time between two consecutive blows is 60 ÷ 500 = 0·12 seconds, and at an average speed of 21 ft. per minute of the cane carrier apron, the linear advance of the carrier in this time limit is only :—

$$21 \times 12 \times 0{\cdot}12 \div 60 = 0{\cdot}5 \text{ inch.}$$

The cuts are as irregular as the cane is loaded on the carrier and, as the author has observed many times, a very good performance is obtained when the knives are sharp and the cane thus is really splintered.

From a very good installation with swinging blades, having 36 blades on a 6 ft. carrier, driven by a high-speed twin vertical engine, $14\frac{3}{16}$ in. dia. × 10 in. stroke, running at 350 r.p.m. and driving the revolving knives at 500 r.p.m. by a wide belt with belt tightener, the author has taken indicator cards for maximum and average power input. The carrier is of the Cuban type and the speed is irregular as the operator keeps an eye on the loading of the cane chute before the crusher and the carrier is run at a variable speed according to these requirements. The cane loading is also irregular and although there were no open gaps, a large variation in power input would appear from time to time. In *Fig.* 126 is shown a *Normal Diagram* of this high-speed engine at the left side, whereas on the right side are given the diagrams when a maximum feed of cane was cut by the knives. The shaded part gives the fall in power during 40 revolutions of the engine and the time limit was 40 × 60 ÷ 350 = 7 seconds, which indicates the very good regulating performance of the shaft governor on the throttle valve.

The clearance between the knife tops and the carrier slats was 2 in. The normal power input at an M.E.P. of 26 lbs. per sq. in. amounted to 146 i.h.p., whereas the maximum at an M.E.P. of 62 lbs. per sq. in. amounted to 348 i.h.p. When taking the efficiency of the engine, belt drive and shaft *plus* air resistance of the knives at 0·80, the power input varies between normally 117 b.h.p. and 278 b.h.p. for maximum conditions. As the grinding capacity of this mill is 70 long tons of cane per hour, the necessary power input is established between 1·67 b.h.p. and 3·96 b.h.p. per ton of cane per hour for this particular installation, where about 90 per cent. of all the cane is cut. The maximum average force of the blow, calculated according to formula (55), is about 220 lbs. with a lever arm of 2 ft. 4 in., thus being far in excess of the normal blow; and as this is only an average, a considerable safety factor has to be embodied in the calculation. The shear of the $1\frac{1}{4}$ in. bolts is low, considering the reacting couple of the blow and the centrifugal force combined. The author does not know any instances where the bolts were actually broken and the blades thrown away.

From another installation, having 42 knives over a 7 ft. carrier, driven by a high-speed engine of the same size and speed, 130 long tons of cane were ground per hour. The engine developed 200 b.h.p. and a power consumption of 1·54 b.h.p. per ton of cane per hour resulted.

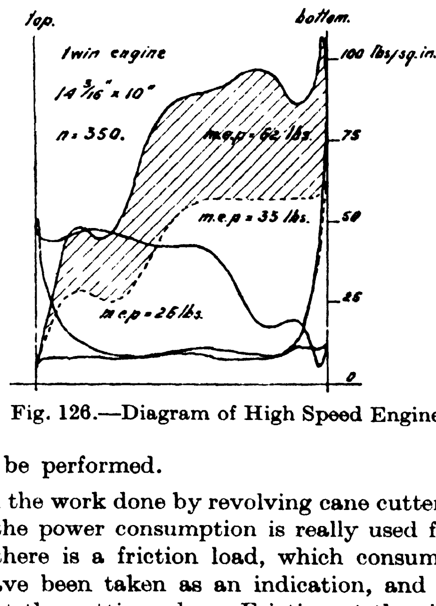


Fig. 126.—Diagram of High Speed Engine.

The power consumption, therefore, is normally taken at 1·5 b.h.p. per ton of hourly ground cane, but 2 h.p. will be more conservative. Much depends on the clearance of the knives above the carrier, considering the sharpness of the knives as a necessary condition. In some instances up to 90 per cent. of the cane is actually cut, whereas in other instances only 60 per cent. is disintegrated and it is no good lowering the power consumption by reducing the mechanical work to be performed.

The author has frequently studied the work done by revolving cane cutters, so as to reach a conclusion whether the power consumption is really used for disintegrating purposes, or whether there is a friction load, which consumes power ineffectively. Worn blades have been taken as an indication, and as a rule they all wear at the top, just at the cutting edge. Friction at the side of the blades is of much lesser importance, so the author is inclined to believe that by far the biggest share of the power input is used for defibration.

The speed of the knives might accelerate the movement of the cane particles and consume power by throwing these away, but from actual performance it will be found that only very few pieces of cane and then of small size only are scattered, and so only a very small power consumption can be attributed to this acceleration performance.

In several countries a double set of knives is arranged for; the first having a clearance from 8 to 24 inches from the carrier, and the second set as close as possible above the apron. The number of the *levelling knives* is half the amount of the cutting ones, and about half the power input is necessary. The work to be done by the cutting knives is somewhat reduced, but the total power input will be higher than where one set of knives is used. In countries like Cuba one set only of knives is accepted as standard, and a cane kicker, consuming from 10 to 20 h.p., on a 7 ft. carrier does the levelling performance. It should, nevertheless, be observed how the crusher is being fed and if the feed can be regulated at will after having passed the cane cutting knives.

Ball bearings are not used on the knives shaft and white metal bearings with automatic lubrication and large bearing surface are well adapted for this service, where shocks are not always avoidable. The vibration of the shaft is small, as may be ascertained by touching the bearings of a set of knives in operation.

The main bearings are normally mounted on horizontal slide rails, so that the distance between knife tips and the apron can be changed by pushing the shaft horizontally. The belt tension gear will take care of the difference in centre distance between engine and knives shaft. With electric motor drive, the motor has to move also on slide rails along with the shaft, which rule applies also for turbine-driven knives.

A patented construction is now on the market, where the guide beams of the upgoing carrier chains are raised at the locus of the cane knives, to adjust the clearance. The knives shaft therefore will remain in a fixed position.

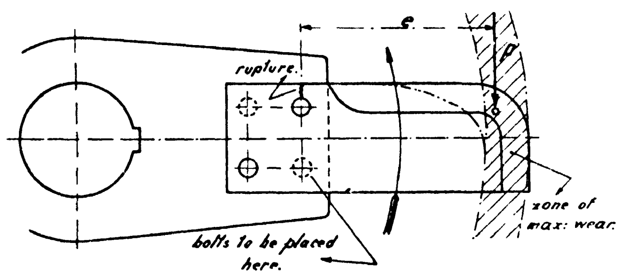


Fig. 127.—Wear and Bolting of Blades.

A flywheel is placed at the free end of the knives shaft to ensure smooth operation. The inertia of the knife material must also not be overlooked.

In *Fig.* 127 is shown the *Wear* and the *Bolting* of the blades. Several worn blades have been inspected by the author and the dotted line indicates their average wear. The zone of maximum wear is also indicated and it will be seen that this takes place at the top of the knives.

With long blades, there is a danger of rupture as indicated, when the bolt holes are not properly placed; and manufacturers now have accepted as standard practice the drilling of the bolt holes as indicated by the dotted circles. The bolts are sometimes reamed, but this requires very close tolerances in drilling.

The attached end of the blade is finished (machined) on both sides, so that it will make a tight fit in the milled notch in the hub arm.

The knives are made of high-class plough steel, having a natural Brinell hardness of about 225°.

As the blades exert their mechanical performance at the point, it is obvious that long blades and short hubs are out of place, as the momentum $P \times e$ is increased. Moreover, the worn knives will have a greater weight of material to be thrown away, and although this material might be used for other purposes, economy points to short blades.

A *Standard Short Blade* of modern manufacture is shown in *Fig.* 128; the point is forged to give a longer wear.

In *Fig.* 129 is shown a *Boltless Type of Blade.* The blade is dovetailed in the cast steel hub and held by a dovetailed key, so there are no bolts to remove when replacing the blades. The blades are pushed in as indicated by the arrow, after the key has been removed. The key is arrested by a set screw, so that it will not work loose during operation.

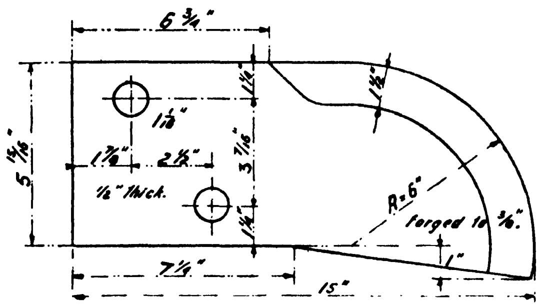


Fig. 128.—Short Blades.

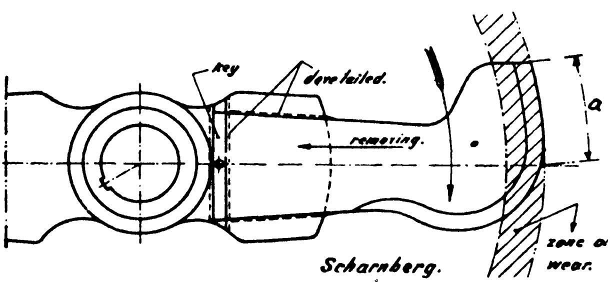


Fig. 129.—Boltless Blades.

The inventor has devised a long wearing surface over the length *a* at the cutting line, so to ensure long life for the blade. The blades have a few recesses not shown, to reduce the weight of the blades and the corresponding centrifugal force.

In *Fig.* 130 is shown a *Knife Construction* having four cutting edges, here also with the object of long life. These cutting edges are serrated, to increase the grip on the cane, and to produce a shredding effect. The area of the blade in the wearing zone is also shown.

In *Fig.* 131 is shown the arrangement of a set of *Small Pitch Knives.* Four blades are attached to one hub piece, the other designs having only two. The pitch, therefore, is reduced to $\frac{7}{8}$ in. and it is obvious that a very good cutting performance results. The power input for two sets of such knives for a grinding capacity of 50 tons cane per hour has proved to be about 3·4 h.p. per ton of cane per hour. The 64 blades of the cutting knives are arranged in four helicoidal lines around the shaft, as indicated by the consecutive numbers in the drawing.

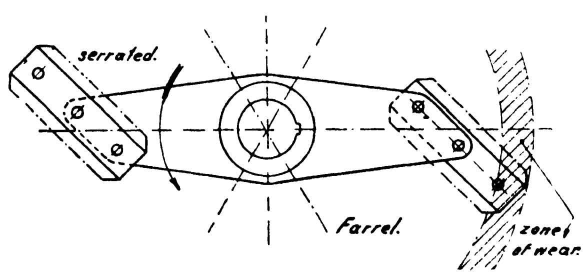


Fig. 130.—Blades with Four Cutting Edges.

The blades are reversible, but care should be taken not to have bolt connexions within the wearing zone, as they will be worn down rapidly and increase the power consumption.

The scope of this book does not permit mention of all the knife designs now on the market, and only fundamental data have been given, so as to assist in the proper selection and construction of the knives. There are hubless blades, circular blades, blades with the tops bent at right angles, etc., which all have found application in practical factory operation.[1]

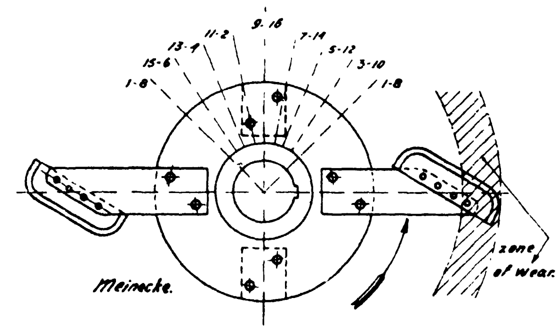


Fig. 131.—Small Pitch Knives.

As to the defibration effect, little was known till lately, but the Java Experimental Station has published valuable data as to the fineness of the cane chips produced by different methods.[2] From these data the *Bagasse Fineness Diagram* shown in *Fig.* 132 is drawn. The tests were made by putting a certain amount of bagasse in an 8 in. vertical tube, the bottom part of which

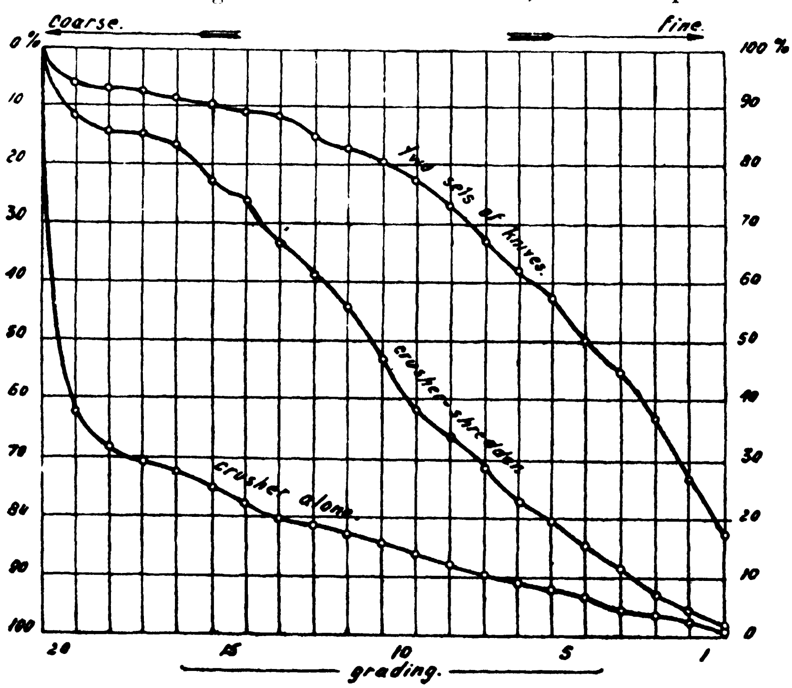


Fig. 132.—Bagasse Fineness Diagram.

had been connected to a blower and around the top of which a dust arrester was arranged to collect the particles drawn up by the air current. By applying different air pressures, such as the blower might produce by varying speeds,

[1] See F. MAXWELL, "Modern Milling of Sugar Cane," pages 53 to 59.
[2] See *Het Archief*, 1934, pp. 917-964; and *I.S.J.*, 1935, pp. 188-189.

different sized particles were floated up and the lightest particles were separated first by the lowest pressures of the wind-making equipment. The collected particles at each definite air pressure were weighed, and the diagram is made according to these weights. There were 20 different grades of fineness, No. 1 being the finest and No. 20 the coarsest size.

From these tests it was found that on preparation with a single Krajewski crusher about 60 per cent. of the cane is not finely disintegrated ; the preparation with a crusher-shredder shows a far better performance ; while a double set of small pitch revolving knives showed the best results.

As already mentioned in Chapter IV (Cane Carriers), overlapping steel slats have to be used when the cane is disintegrated on the carrier. Wooden slats with gaps between have proved unsuitable for this work, as the fine particles will fall through. The modern construction of these steel slats can be taken as fully reliable.

In *Fig.* 133 the *general arrangement* (side view or elevation) of a set of revolving knives is shown. The knives are covered by a hood of steel plates and hinged doors are fitted at front and rear ends. These doors have to keep clear of the knives, whatever swinging may take place. Instead of one sheet, the swinging doors are divided into strips about 1 ft. 6 in. wide, which can swing individually.

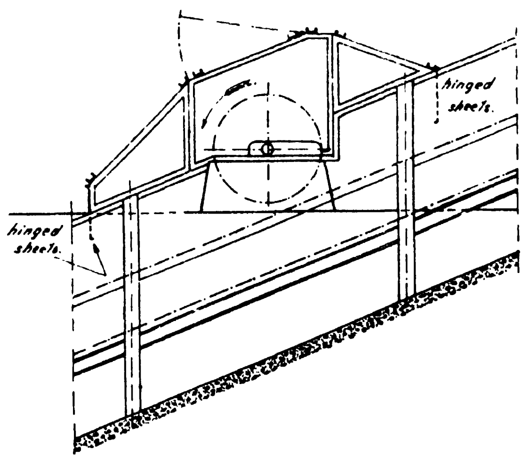


Fig. 133.—Arrangement of Revolving Knives.

The rotation of the knives has to be as indicated by the arrow, thus *with* the carrier travel.

When two sets of knives are arranged for, the first levelling set is generally placed on a concrete foundation, whereas the second or cutting set is placed on a steel structure. When one set of knives is used, preference should be given to the concrete foundation, as it gives a solid anchorage for the bearings. The driving pulley for the belt or V-ropes should have an outer bearing when possible. The engine and the knives shaft have of course to be mounted exactly parallel in all planes.

The knife hub pieces are held together by tightening nuts on the main shaft, having threads against the direction of rotation. The main bearings are placed as close as possible against the carrier walls.

The hood is a protection against small particles of splintered cane which fly around, and also against the spattering of juice. The author has many times observed that this spattering acquires the form of a dense fog under the hood.

2.—Shredders.

Shredders have long been used as a means of preparing cane before milling, and of late years very interesting developments have taken place in this kind of equipment.

The first shredder, introduced and invented in Louisiana, was the *Abrasion Shredder*, as shown in *Fig.* 134. The author has seen the last ones in operation in Cuba, where they are now no longer employed. In Australia they still form part of the standard equipment.

This shredder consists of two 5 in. shafts, on which are mounted cast iron tubular bushings of 16 in. diameter, which in turn support the shredder discs made of chilled cast iron, each 8 in. wide, and of double conical form, with a sloping angle of 45° with the axis. The discs have a mean diameter of about 22 in. and are provided on their circumference with 40 teeth of undercut profile which are about ⅜ in. deep. The two shafts are rotated in opposite directions, so that the cane will pass between the shredder rollers, which are each composed of a number of discs, held together by six through-going bolts with end plates.

Due to the sloping of the shredder discs, the effective shredder length is increased to the amount :—

$$L_{eff.} = L_{norm.} \div \sin \alpha \quad \ldots\ldots\ldots\ldots \quad (56)$$

where $L_{norm.}$ = normal shredder width and

α = half the top angle.

In our case the sloping angle is equal to half the top angle or 45°, and the effective shredder length is increased 1·41 times. For a 72 in. shredder, this value amounts to 101·5 in.

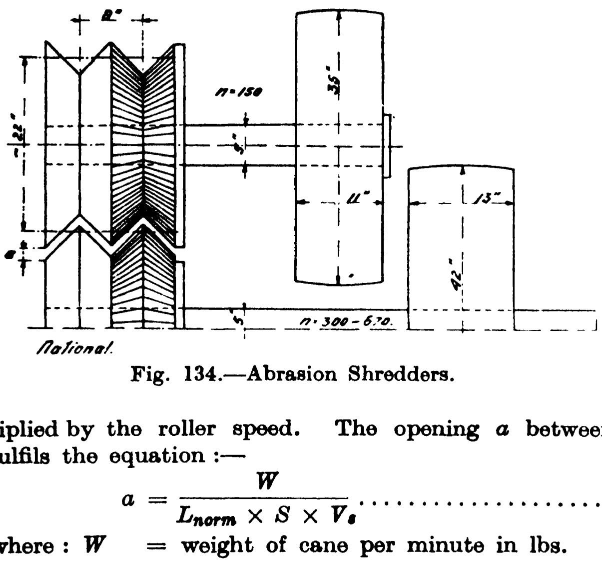


Fig. 134.—Abrasion Shredders.

Similarly, as is the case with the mills to be dealt with later, the rotating shredder rollers produce a volume equal to the total shredder passage multiplied by the roller speed. The opening a between the rollers therefore fulfils the equation :—

$$a = \frac{W}{L_{norm} \times S \times V_s} \quad \ldots\ldots\ldots\ldots \quad (57)$$

where : W = weight of cane per minute in lbs.

L_{norm} = normal roller length in ft.

S = specific gravity of shredded cane in lbs. per cub. ft.

V_s = spherical roller speed in feet per minute.

The opening a will be obtained in feet, and the specific gravity of shredded cane is about 67 lbs. per cubic foot, but an allowance has to be made for air gaps between the canes, when passing the shredder. The spherical roller speed has to be taken as the arithmetical average of the sum of top and bottom roller speeds. The top roller rotates at 150 r.p.m., whereas the bottom roller rotates at 300 to 670 r.p.m. The average speed, therefore, amounts to :—

$$V_s = D \times \pi \times (n_t + n_b) \div 2 \quad \ldots\ldots\ldots\ldots \quad (58)$$

where : D = mean roller diameter in feet.

n_t = number of revs. per min. of the top roller.

n_b = do. of bottom roller.

In *Fig.* 135 this *Abrasion Principle* is graphically shown. The teeth of the top roller *a* operate in the opposite direction to those of the bottom roller *b*, as they have to hold the cane and prevent any slip, and thus both rollers will participate in the shredding action. This shredding is done by the differential relative speed between both rollers, but the fast rotating bottom roller will receive the biggest share.

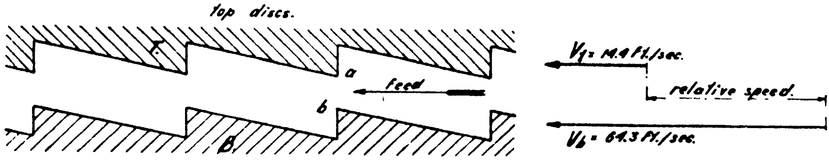


Fig. 135.—Abrasion Principle.

As the discs which compose the shredder rollers are chilled, the wear is reduced, and the average life amounts to a capacity of about 80,000 tons of cane. After this quantity has been shredded, the chilled discs have to be renewed.

In *Fig.* 136 is shown an improved *Belt Drive* for this shredder, as made by the manufacturers. Formerly a crossed belt drive was used for the top roller drive, but the wear on this belt proved excessive, and led to the construction as shown. The reversing of rotation is achieved by a simple gear drive, and both belts are now of the open type.

The centre line of the shredder rollers is inclined at 45° with the horizontal to ensure efficient feeding. The top rollers are spring loaded as a means of safety under excessive load or when tramp iron goes through the shredder.

The manufacturers give the capacity of the different sizes as between 5 and 100 tons of cane per hour, whereas the power consumption is given as 0·75 to 1 h.p. per ton of cane per hour.

The work of the shredder is obviously improved when the lower roller rotates at a higher speed, as the average spherical roller speed is increased and the shredder opening *a* thus reduced. The power consumption will increase, as more mechanical work has to be delivered.

Reference should be made to the abrasion defibrator used by the "Vazcane" process of making insulation board out of bagasse.[1] In the trial plant working according to this process at Marianao, Cuba, which the author visited, the cane is ground on the surface of a 20 in. × 42 in. silicon carbide grindstone, driven at 1200 r.p.m. by an electric motor of 100 h.p. The cane is fed to a vertical hopper 15 ft. high on top of the grindstone and provided with drag chains to push the cane down.

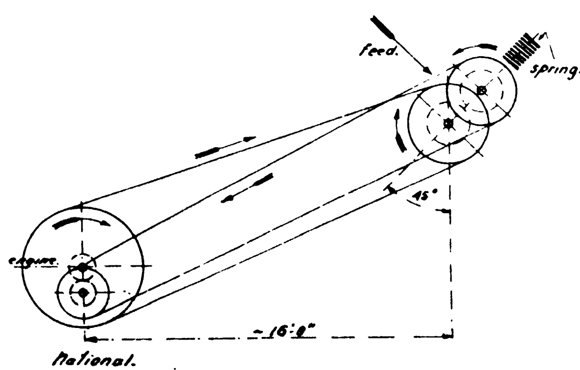


Fig. 136.—Belt Drive for Shredder.

[1] U.S. Patents 1,688,904/5 1928; and see *Industrial and Engineering Chemistry*, Vol. 22 pp. 765-1930.

The power consumption is very high, being 1·83 kw. per 2000 lbs. cane per 24 hours, or about 60 h.p. per short ton cane per hour. Although the stone has to be re-ground or sharpened every day and the power consumption appears far too high for economic sugar production, the cane fibres produced are of unsurpassed fineness and the sucrose is extracted by a maceration process without the use of any mills whatsoever. That the defibration is very complete may be proved by the fact that, operating with this leaching process, a sucrose extraction of 99 per cent. is obtained.

Rasping machinery, such as is used in the potato, starch and manioc (tapioca) industries, which also is based on the abrasion principle, has not been applied in the cane sugar industry.

Slicing machinery similar to that used in the beet industry has been tried out in the cane factory, to be followed by a diffusion process, but the experiment has been discontinued.

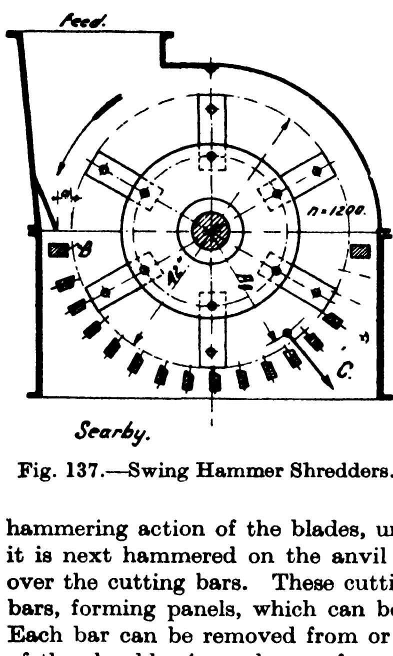


Fig. 137.—Swing Hammer Shredders.

The type of shredder mostly used in Hawaii is the *Swing Hammer Shredder* shown in *Fig.* 137. The construction is as follows: On the main shaft are mounted a number of mild-steel discs, having a pitch of about 1¼ in., and carrying six swing hammers of flat iron between each two adjacent discs. These swing hammers revolve on long bolts or pivots, having bushings for each hammer. As soon as the shaft starts to rotate at the desired speed, the flat iron hammers will be thrown out radially by the centrifugal force.

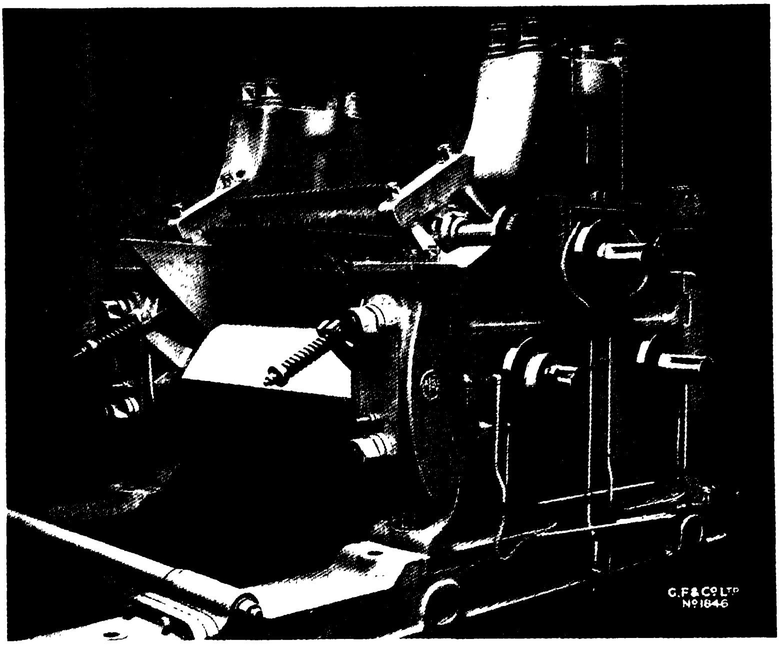
The whole shredder is contained in a cast iron frame having double row ball bearings for the main shaft and provided with a hinged cover on top.

As soon as the cane is fed into the receiving hopper, it is subjected to the hammering action of the blades, until it has been reduced to the space *a* and it is next hammered on the anvil bar *B* and the disintegration is completed over the cutting bars. These cutting bars are mounted in curved slotted side bars, forming panels, which can be easily removed from the shredder casing. Each bar can be removed from or reversed in these panels. The total width of the shredder is made up of several panels, so that the unsupported length of the cutting bars will not be too great.

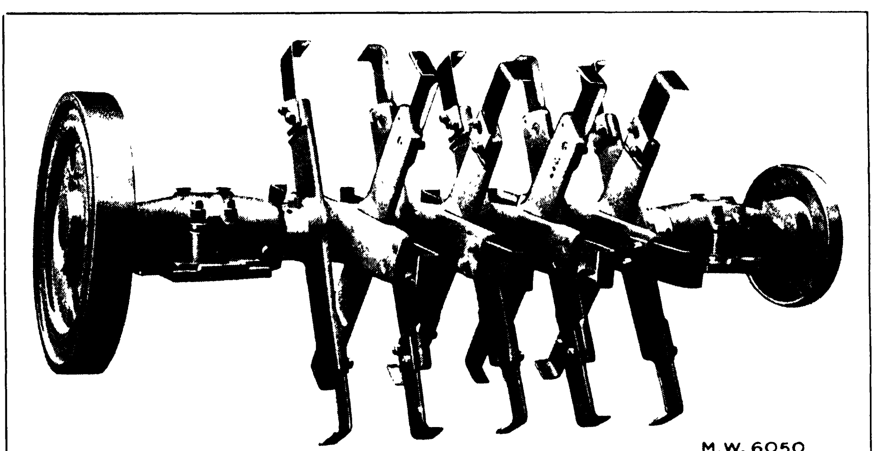
The shredder is made in different sizes, all having the rotor of 42 in. outside diameter and 36 in. to 84 in. wide. The capacity as given by the manufacturers is from 6·6 to 18 tons of cane per hour per foot shredder width, the smallest size having the smallest capacity.

The power consumption is given as about 2 h.p. per ton of cane shredded per hour, whereas the driving motors have a normal power output from 8·5 to 2·8 h.p. per ton cane per hour, the smallest sizes having the highest specific power input.

The rotor revolves at about 1200 r.p.m. and is coupled by a flexible coupling to the motor shaft.

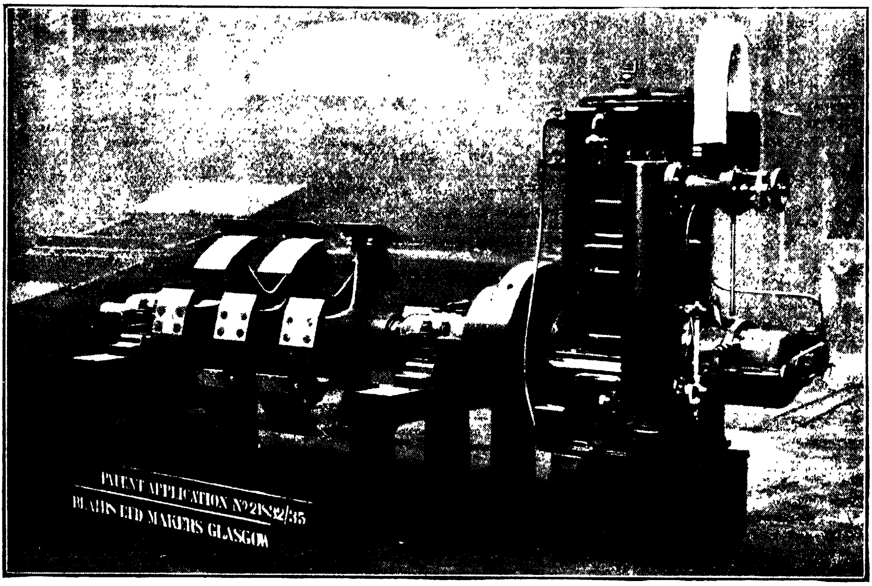


THREE-ROLLER MILL EQUIPPED WITH MAXWELL SHREDDER.
(*Geo. Fletcher & Co., Ltd.*)

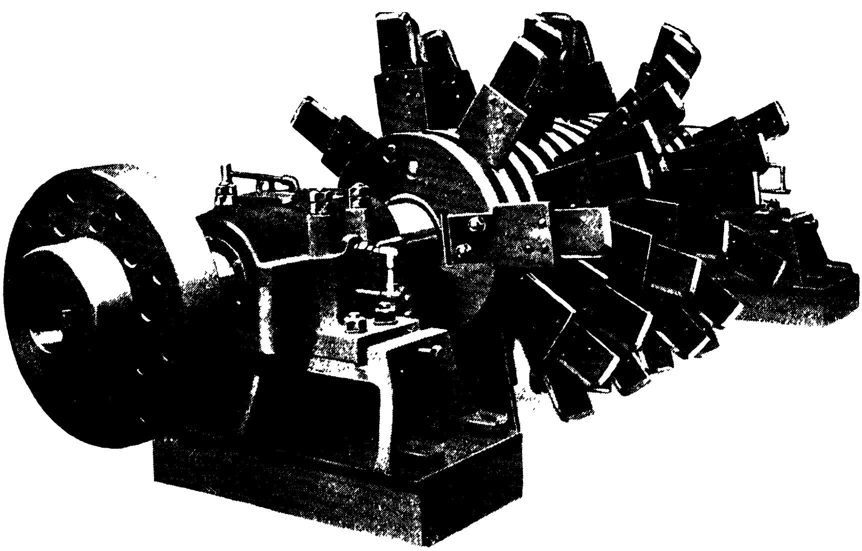


REVOLVING CANE KNIVES WITH BENT BLADES.
(*The Mirrlees Watson Co., Ltd.*)

PLATES 23 & 24.



HORIZONTAL BLADE CANE CUTTER.
(*Blairs, Ltd.*)



SMALL PITCH REVOLVING CANE KNIVES.
(*Fawcett, Preston & Co., Ltd.*)

The hammers are pieces of flat iron or mild steel, about 8 in. long and 2½ in. wide, weighing 1·75 to 3·5 lbs. each. The blades are provided with two bore holes for the bar pivots, so that they can be reversed and the four cutting edges be used until completely worn, ensuring a long life. Moreover, the shape of the blades is very simple, so that they can be produced on the spot, which is an advantage for remote locations.

The cane is pressed against the cutting bars by centrifugal force, and therefore the spaces between must not be too large. As these openings are wider on the outside than on the inside, there is no danger of clogging them with shredded cane.

At a mean radius of 20 in. = 1·67 ft., each lb. of cane receives a centrifugal force of :—

$$C = \frac{M \times V^2}{R} \quad \dots\dots\dots\dots\dots\dots\dots\dots \quad (59)$$

where M = mass = weight in lbs. ÷ 32·16

V = $2\,\pi \times n \times R$, n being the number of revs. per second.

R = Length of radius in ft.

As the spherical speed is about 210 ft. per second, the cited centrifugal force for each lb. of cane amounts to 820 lbs.

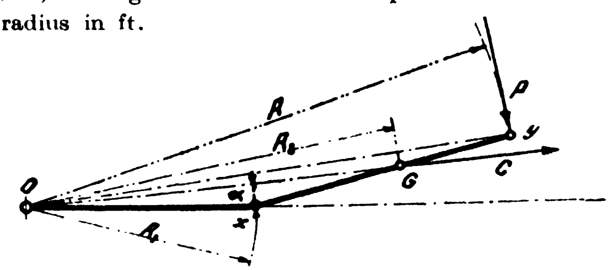


Fig. 138.—Swing Hammer Principle.

Foreign matter, like pieces of "tramp iron," will have a heavy impact on the bars, and as a safeguard and to protect the cutting bars from becoming blunt, a few bars are left out at the end of the shredding course.

The swing hammer shredder is located generally between the crusher and the first mill, and special arrangements have to be made for its installation on an existing milling plant.

The inertia accumulated in the shredder rotor is sufficient to ensure smooth running, and a flywheel is not necessary.

The *Swing Hammer Principle* is shown in *Fig.* 138. 0-x is the radius, where the pivots for the hammers are located. This radius R_1 is taken to be 14 in., whereas the centre of gravity of the bar x-y at G is located on a radius R_2 taken at 17 in. With a hammer weight of 3 lbs., the acting centrifugal force C amounts to about 2100 lbs. as derived from formula (59).

Due to the obstructing cane it is assumed that the bar x-y is thrown back, so the centrifugal force C forms an angle $\alpha = 10°$, in which case the following couple of forces will prevail, revolving around x :—

$$C \times R_1 \times \sin\,\alpha$$

and the hammering force P (which is different from the blow) on the extremity of the blade will be :—

$$P = C \times R_1 \times \sin\,\alpha \div R \quad \dots\dots\dots\dots\dots\dots \quad (60)$$

With the given dimensions, the centrifugal force C of one bar amounts to about 2100 lbs. when the rotor is running at 20 revs. per second, and the hammering force P to about 254 lbs., when the blade is thrown 10° back.

For a 54 in. shredder, having 240 blades or hammers and 150 h.p. assumed power input, the force of the blow, according to formula (55) amounts to :—

$$P_{blow} = \frac{150 \times 550 \times 60}{240 \times 1200 \times 1{\cdot}75} = 9{\cdot}8 \text{ lbs.}$$

It will be seen that as soon as the hammers start to swing out, the hammering force assists the blow very efficiently and in actual operation the hammers are nearly in a radial position.

The pivots, blade sections, etc., have to be calculated to these combined forces.

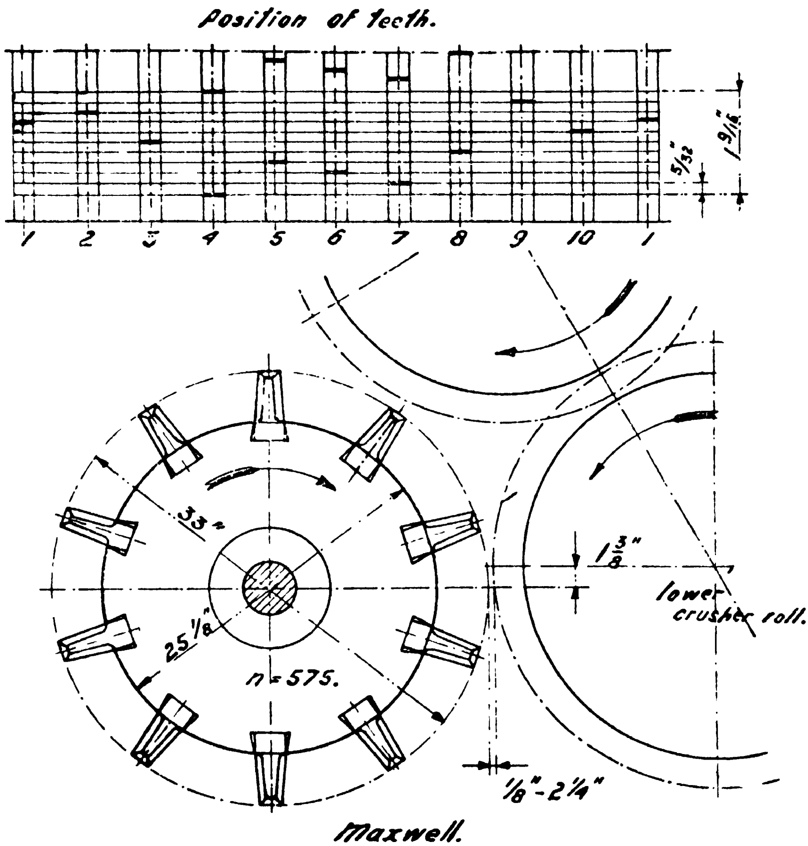


Fig. 139.—Splitting Shredder.

An interesting new type of *Splitting Shredder* is shown in *Fig.* 139 ; it is composed of a shaft on which is mounted a solid steel drum of 33 in. diameter, on the periphery of which are ten grooves running parallel to the axis which hold the knives, about 4 in. long. The knives or teeth make a radial cutting performance just as do revolving knives. To prevent heavy wear the cutting or top angle of the knife edge is 90° or more.

The teeth are separated by distance pieces, which fit into the conical grooves, whereas the teeth proper have a dovetailed bearing against these distance pieces. This makes removal and re-placing easier as the locking nuts and plates at the ends of the grooves have only to be loosened in order to lift out the teeth. The latter are made of special steel, having a Brinell hardness of about 500°, achieved by oil hardening.

The first installation seen by the author had the shredder roller at the locus of the discharge roller in the first mill, so the bagasse was cut as soon as it emerged in a compressed state from between the top roller and the trash turner plate. The inventor subsequently abandoned this idea and the shredder-roller is now located at the spot where the crushed cane emerges from the crusher and this combination is called a *crusher-shredder*. This arrangement is shown in *Fig.* 139, and the distance between the top of the shredder teeth and the outer periphery of the crusher can be as little as $\frac{1}{8}$ in. or even $\frac{1}{16}$ in.

The crushed cane is split by the shredder teeth and thus shredded, and the cutting or splitting performance is based on the same principle as with revolving knives, but with an increased number of cutting blades (teeth) and a remarkably small pitch of $\frac{5}{32}$ in. only.

The power consumption of the latest arrangement is very low, amounting to about $\frac{1}{2}$ h.p. per ton of cane ground per hour, but exclusive of the crusher power input. The capacity of the driving motor is greater, as is customary in all power drives of kindred equipment. A crusher-shredder of 33 in. × 78 in. has 485 teeth and is coupled by a flexible coupling to a 70 h.p. electric motor running at 575 r.p.m. The average maximum force of the blow, therefore, amounts to :—

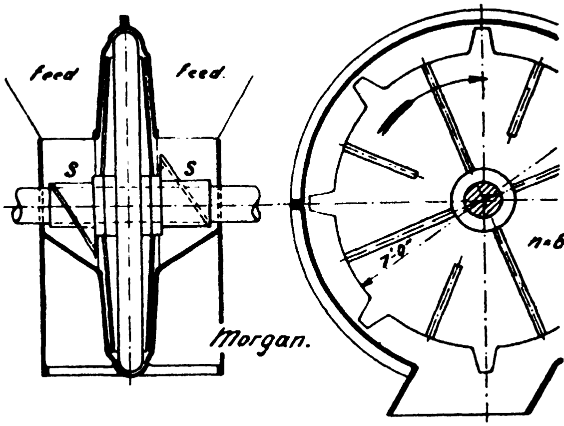


Fig. 140.—Cane Disintegrator.

$$P_{blow} = \frac{70 \times 550 \times 60}{485 \times 575 \times 1{\cdot}375} = 6{\cdot}0 \text{ lbs.}$$

according to formula (55), and this is the lowest value of the shredders treated.

The staggered position of the teeth, which have a pitch of $1\frac{9}{16}$ in., is shown in the top sketch of *Fig.* 139.

These crusher-shredders are in actual operation in Java and British India, and the Java Experimental Station has published a report of the favourable results obtained from the installation of these shredders, when combined with proper mill grooving and the efficient application of imbibition.

As the shredder roll has a large inertia, flywheels are not required for smooth running. The friction resistance is very small, as the rotor shaft is mounted in a pair of double row barrel-shaped roller bearings. The inventor states that after 60,000 tons of cane are ground, the whole shredder roller should be reversed.

The latest development in cane shredding is the *Cane Disintegrator*, diagrammatically shown in *Fig.* 140.

A heavy disc of mild steel 7 ft. in diameter and 6 in. wide[1] is mounted on a shaft with large white metal, automatically lubricated bearings and revolves at 600 r.p.m. In milled grooves on both sides of the disc are attached strips of special steel. The cast steel housing, which resembles the housing of a large centrifugal pump, is also provided on the inside with these steel blades and on each cutting front there are four long blades and four small ones, so that the disintegrator contains in all 32 steel blades.

The revolving and fixed blades, respectively *D* and *F* in *Fig.* 141, pass each other with a small clearance *a* of about $\frac{1}{16}$ in., the maximum being $\frac{1}{4}$ in.

The cane is fed bi-laterally to the feeding bins, which are provided with scrolls *S* on the main shaft and which push the cane centrically between the disc and the sides of the housing. The cane has to be cut by two sets of revolving knives before it reaches the cane disintegrator, as it cannot be fed efficiently in the form in which it arrives from the fields. It is obvious that the cut cane is actually hammered and the cells ruptured and very good shredding is obtained, as there are no pieces which can escape the disintegrating performance. The author has seen such a disintegrator at work at Central Hershey in Cuba ; the shredded cane has a hay-like structure, only surpassed by the grinding on stones in the Vazcane process.

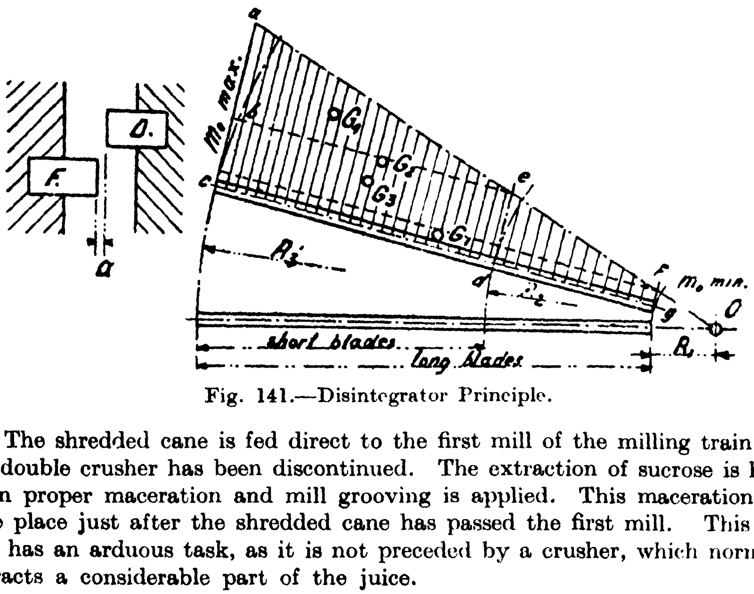


Fig. 141.—Disintegrator Principle.

The shredded cane is fed direct to the first mill of the milling train and the double crusher has been discontinued. The extraction of sucrose is high, when proper maceration and mill grooving is applied. This maceration can take place just after the shredded cane has passed the first mill. This first mill has an arduous task, as it is not preceded by a crusher, which normally extracts a considerable part of the juice.

The capacity of this disintegrator is given as 280 short tons of cane per hour, but actually at Central Hershey 160 short tons per hour have been ground. The main shaft is connected by flexible couplings to a 300 h.p. motor at each end and actual power consumption averages 3·75 h.p. per short ton of cane per hour.

That the hammering effect is tremendous may be assumed from the fact that each revolving blade delivers eight blows per revolution on the fixed blades. As there are eight revolving blades, a total of 38,400 blows is delivered per minute on each side of the disc. This shows clearly that there is no escape from the disintegrating action. The shredded cane is guided along channels

[1] U.S Patent No. 1,804,797, May 12th, 1931.

cast in the housing to the outer periphery of the disc by centrifugal force and then expelled by the lugs on the outside of the disc.

The *Principle of Disintegrating* is shown in *Fig.* 141. The long blades have a length of $R_3 - R_1$, whereas the short blades have a length of $R_3 - R_2$.

Of the 64 blows per revolution of the rotor on each side, there are 16 blows on the long blades and 48 on the short ones. As the rotor shreds on both sides, the total figures will be 32 and 96 respectively.

If the striking force per unit of length of the blades is taken as P, the momentum on the radius R_3 will amount to $P \times R_3$ for each blade, thus :—

$$M_{o3} = P \times R_3$$
$$M_{o2} = P \times R_2$$
$$M_{o1} = P \times R_1$$

The aggregate momentum for a long blade therefore amounts to :—

$$M_{oal} = \frac{P \times R_3^2}{2} \times 2/3\ R_3 - \frac{P \times R_1^2}{2} \times 2/3\ R_1$$

or : $$M_{oal} = 1/3\ P \times (R_3^3 - R_1^3) \ldots\ldots\ldots\ldots\ldots\ldots\ldots (61)$$

Likewise for the short blades the aggregate momentum has the value :—

$$M_{oas} = 1/3\ P \times (R_3^3 - R_2^3) \ldots\ldots\ldots\ldots\ldots\ldots (62)$$

For easier calculation it is assumed that $R_1 = \frac{1}{4} R_3$ and $R_2 = \frac{1}{2} R_3$, then the total momentum per revolution of the rotor amounts to :—

$$M_{oal} = 10{\cdot}5\ P\ R_3^3$$
$$M_{oas} = 28{\cdot}0\ P\ R_3^3$$

$$\text{Total } M_o = 38{\cdot}5\ P\ R_3^3$$

As $R_3 = 3{\cdot}5$ ft. in our case, the total momentum per revolution—assuming there is no side friction on the disc to absorb power—amounts to 1650 P ft. lbs.

The power input per revolution is :—

$$\frac{600 \times 550 \times 60}{600} = 33{,}000 \text{ ft. lbs.}$$

and by simple division the average blow per foot of blade length amounts to 20 lbs., which cannot be considered an excessive figure.

The newest design of *Cane Disintegrator* is shown in *Fig.* 141a ; here it will be seen that the shredding performance is unilateral, thus only on the feeding side of the disc *a*. This disc is held firmly by the cone *b*, which bears against a shoulder on the shaft, the connexion between the disc and the cone being effected by bolts. Any thrust acting upon the disc is therefore transmitted to the shaft and at *c* is mounted an adjustable heavy thrust bearing of a conical roller design, similar to that shown in Fig. 41, Chapter II, but having the rollers at a reversed and more pronounced inclination, specially designed for carrying the thrust.

The bearing *d* is of the normal conical roller type and has very good qualities for such vibrating performance at high speed as occurs when shredding cane.

Eight rotary knives or bars *f* co-act with eight stationary ones, marked *g*, which are easily changed. The casing, as well as the feed hopper, the latter of large size for the efficient feeding of coarsely cut cane, is split on the horizontal centre for quick dismantling. The lower halves of the casing and the hopper are attached to the bearing supports, which rest on the foundation.

A liberal space is left between the feeding scroll and the hopper. At the vertical casing joint an annular ring *h* is inserted to ensure a round corner ;

this will reduce abrasion and the corrosive action of the cane shreds or juice. This ring can be made of any material suitable for the purpose.

The disintegrator shaft is provided with a half coupling *e*, fitted on the tapered end and secured by a lock nut and feather key, as is the custom with high speed shafting.

The discharge (as indicated by chain dotted lines in the Fig.) of the shredded cane is effected in the top casing in a horizontal direction, the rectangular opening having the same width as the inside of the casing.

Lately, a new type of *rotary cane cutter* or shredder has been put on the market; it consists of several sets of four short blades, which are mounted parallel to the shaft on hub pieces at a radius of about one foot and driven from a high speed steam engine or electric motor. Data about power consumption and defibration are not yet available.

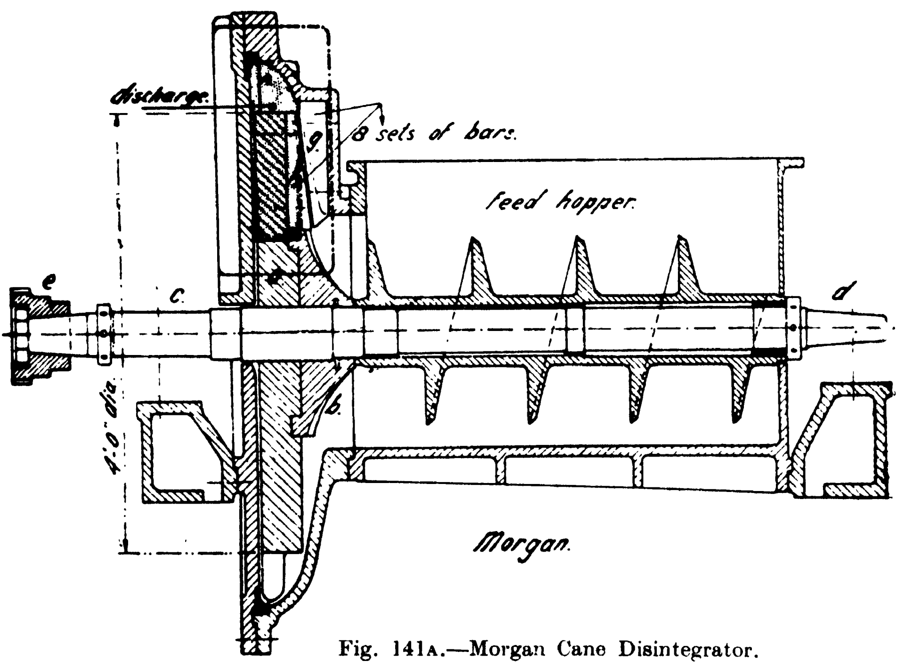


Fig. 141A.—Morgan Cane Disintegrator.

Shredders and revolving knives will allow about 25 per cent., and sometimes more, increase in grinding capacity of the milling plant to which they are attached, as the mills proper are partly or totally relieved from the task of disintegrating and are exclusively employed for purposes of extraction.

The total power input per ton of cane ground, therefore, does not differ generally from the previous power input, before installation of the defibrating equipment, as the mechanical work to be done by the mills is reduced.

The author knows several installations where no additional fuel was required after the installation of revolving knives or a shredder. In those sugar factories which have already a high live steam consumption, caused by uneconomical steam engines, the design of the steam distribution of these engines should be improved, before installing shredding equipment.

3.—Tramp Iron Separators.

Tramp iron in the cane mass is always a nuisance and may cause damage to the mill roller grooving, and even heavy breakages have occurred at times. Revolving knives are less subject to damage, as there is generally a clearance of a few inches underneath, so some of this tramp iron will not be hit by these blades. Shredders, on the other hand, having a very small clearance, will be greatly damaged by tramp iron.

The author has seen sledge hammers, car pins and links and such like obstacles that had passed the mills, and the efficient separation of these pieces is of great importance for the maintenance costs of the equipment.

The separation of such tramp iron from the cane is difficult; the only place where electro-magnetic separators for untreated cane have been employed is in the cane chute in front of the crusher. In those instances where this arrangement is not feasible the separator is placed in the chute for crushed cane between the crusher and the first mill.

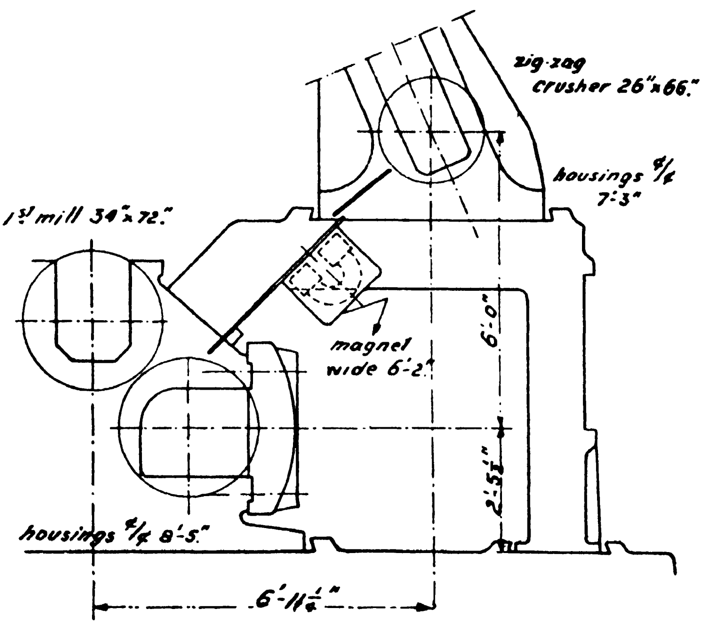


Fig. 142.—Tramp Iron Separator.

The author has installed a fixed *Electrical Tramp Iron Separator*, 6 ft. 2 in. wide, as shown in *Fig.* 142, in the chute of the first mill. The preceding crusher is of the zig-zag type and this piece of equipment did not suffer from any passing pieces of iron. The magnet is built for D.C. current of 110 volts and the starting power consumption is as much as 4 k.w. As the mill is partly electrified with A.C. of 440 volts, a rotary converter of 4 k.w. was installed for supplying the necessary D.C. current.

This separator did not remove all the tramp iron, although when tried with the chute empty any of the usual car pins or links were easily held by the magnet, when they were dropped at the locus of the crusher discharge opening. The actual removal of the tramp iron caused some difficulties, as sometimes the current was switched off, in which case the iron pieces slid into the first mill. There are now electric separators on the market which can be turned 90°, and the current is cut off when in reversed position, so that the tramp iron will drop into a receptacle under the chute.

The cover plate of the magnet is made of non-magnetic material like brass or yellow metal. As the protecting cover of the magnet might accumulate drippings of juice or condensed water, the lowest part is drilled for drainage.

A magnet having a revolving brass cylinder mounted around it is shown in *Fig.* 143, in which the tramp iron is automatically brought beyond the magnetic influence and will then drop underneath the chute. The author has seen such a separator at work and it has a disadvantage similar to the

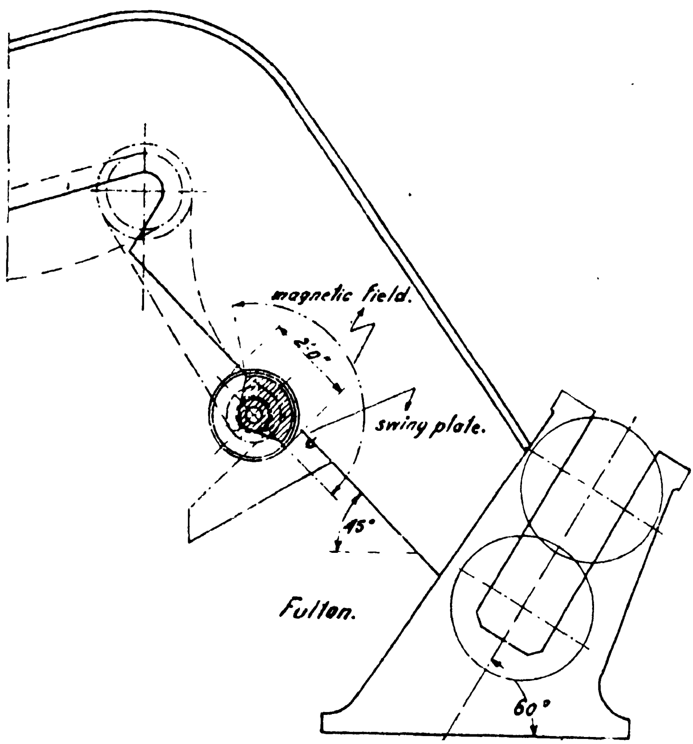


Fig. 143.—Revolving Tramp Iron Separator.

fixed one, that not all the tramp iron is caught. When a layer of cane comes between the iron and the magnet cover, the magnetic force is reduced in proportion to the square of the separating distance between the magnet and the iron. A 75 per cent. protection may be considered as the limit to be obtained.

With the shredder shown in *Fig.* 140, the cut cane is transported on a high speed belt conveyor 48 in. wide and running at a speed of 450 ft. per minute. The magnet is arranged over the belt, so that the tramp iron is lifted. The magnet is 52 in. in diameter and consumes 6 k.w. when starting and 5 k.w. under normal operating conditions. The current is D.C. at 110 volts.

It is obvious that a thin layer of cane is ensured by the high speed of the conveying belt, so the magnet can be placed at a short distance over this conveyor, but the inertia forces of the tramp iron will be greater and the time limit when under magnetic influence is short, so the total efficiency is lessened, although about 65 per cent. of all the tramp iron has been caught.

A 100 per cent. protection cannot be expected of any of the existing electric separators, but as the magnetic forces increase with the weight of the piece of tramp iron, the heaviest pieces generally are removed and a considerable reduction in eventual damage done to the shredders or mills is obtained.

There are also *Centrifugal Tramp Iron Separators* on the market and they are used for crushed or cut cane. Tramp iron has a specific weight about ten times as much as crushed cane, and as the centrifugal force is in direct proportion to the specific weight, the iron will be thrown out with about ten times the force attained by a piece of cane of equal weight. It is obvious that heavy pieces of metal, through their increased weight, will attain a bigger centrifugal force. Hence the tramp iron can be caught in a receptacle out of the path of the cane particles.

The power consumption generally is higher than for magnetic separators, as all the crushed cane has to be accelerated to obtain the centrifugal effect. Where a rotary movement of the cane or crushed cane is already at hand, the centrifugal separation can be embodied in these revolving apparatus.

Since this Chapter was written a very interesting piece of equipment for *indicating the presence* of tramp iron in the cane blanket has been satisfactorily tried out at Central Aguirre in Puerto Rico.[1]

It is composed of a detector coil, mounted above the centre of the carrier and the magnetic field is distorted when a piece of iron comes within its operating radius. Relays are now affected, which control the carrier motors and the cane is dumped sideways, by a side-carrier arrangement.

Per 100 tons cane/hr. about 200 lbs. cane will be dumped sideways and the apparatus has worked so positively that not a single piece of tramp iron has passed during the whole crop.

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1 See *Int. Sugar Jl.*, 1935, p. 448.

CHAPTER VI.

CRUSHERS.

The crushers are considered as preparatory milling equipment, but contrary to the revolving knives and shredders, they combine crushing and extraction operations. Crushers are used in all countries where cane is ground and they belong to the older cane mill preparatory devices.

Extraction of juice from the cane stalks is only possible when the cane cells and tissue are ruptured and this rupturing action can be advanced by shredding or using knives, but has to be completed under a compression exercised between grinding rollers. Only very fine shredding, such as is performed in the Vazcane process, will make the extraction process by rollers under heavy pressure superfluous. The purpose of the extracting rollers, therefore, is to pass the fibre and to hold back the juice contained therein. This extraction performance obviously can only be achieved to a high degree, when the compressed bagasse between the co-acting rollers forms a juice-tight seal, so that the extracted juice cannot be re-absorbed by the bagasse which has passed the maximum pressure between the rollers. The completeness of the extraction depends in the first instance on the fineness of the bagasse and the quantity of juice therein, as the capillarity of the bagasse will always hold a certain amount of liquor or juice, and a 100 per cent. extraction is impossible to obtain. In the case of crushers, the degree of fineness is not very great, but the quantity of juice is from five to six times as large as the fibre content and normally a 40 to 60 per cent. extraction is achieved, which may be considered the average performance of a good crusher installation. The first mill, of course, is greatly relieved by this amount of juice extracted by the crusher.

The compression performance of the crusher is the same as that effected by the three-roller mills and there are now three-roller crushers on the market, having two compressions with only three rollers.

The two-roller construction apparently came into vogue because it was considered only as a crushing agent and one compression was considered sufficient to achieve the said purpose for the time being.

There are different types of crushers made, but as their main difference consists in the grooving of the rollers, this should be dealt with first.

1.—Crusher Roller Grooving.

According to manufacturers' data, the *Zig-Zag Crusher Grooves*, as shown in *Fig*. 144, were first put into practice in 1883. The crusher rollers are provided with intermeshing ridges, integrally cast on the main roller body in a zig-zag line at an angle of 45° with the longitudinal axis of the roller. The lengthwise pitch is about 12 in., where the circular pitch is from 1/14 to 1/20 of the roller periphery, varying according to the roller diameter, and amounting to about 5½ in. as an average.

In *Fig*. 145 is shown a cross-section of these rollers. The shape of the teeth is normally of triangular form, although there are rollers with razor-blade-like teeth; but as wear is heavy on the teeth, those should be preferred which have the largest wearing section.

The ridges are from 2 to 2½ in. high, and the higher the ridge the better the crushing performance. The crusher rollers do not actually touch each other; when running idle a small back lash is applied and the working clearance is subject to a hydraulic or spring pressure on the top roller journals. The crusher opening is in direct proportion to the capacity and the work to be performed.

The intermeshing ridges have to be constructed to operate in such a way that the cane on the extremes of the rollers will not be pushed against the roller flanges, but away from them, as indicated by the arrow in *Fig.* 144. The author has seen an installation where this was not adhered to, and this led to heavy wear on these flanges and even to breakages.

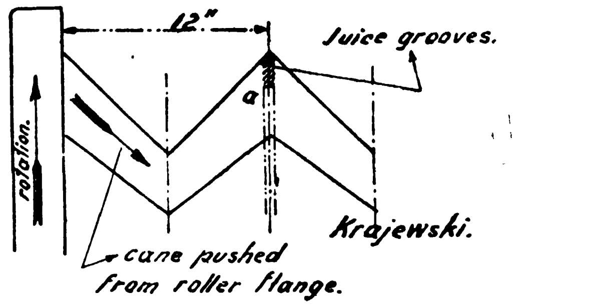


Fig. 144.—Zig-zag Crusher.

Another inherent inconvenience of the zig-zag crusher is the juice drainage from the bottom rollers. The juice from the bottom roller is not released spontaneously, since it lies stored between two adjacent ridges, and as the crusher opening is not sealed hermetically, due to the coarseness of the crushed cane, a part of the juice passes through with this cane. Crushers with slanting housings will be more subject to this inconvenience than vertical crushers. The discharge chute from a crusher is always wet and a part of the juice flows down to the first mill roller, where it is readily drained.

To overcome this inconvenience, the lower roller of the zig-zag crusher is sometimes provided with juice grooves, about ⅜ in. wide, as shown in dotted lines in *Fig.* 144. These grooves must not be deeper than the height of the ridges, as they otherwise will drain the juice to the discharge side, which is contrary to their purpose. Scrapers are generally not used in these grooves.

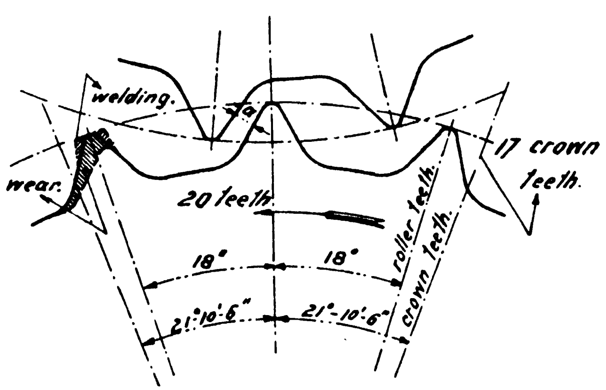


Fig. 145.—Zig-zag Intermeshing.

In *Fig.* 145 it will be seen that the squeezing action depends on the distance *a* between the co-acting ridges. The smaller this distance, the better the squeezing or cutting performance, but then the wear on the teeth is increased and so-called "spitting" will occur, which sometimes soils the mills and running platforms. The rollers of *Fig.* 145 are 36 in. × 84 in. and have 20 ridges on the periphery. As the crown wheels have 17 teeth, they can be arranged in different positions for a variable distance *a*. Manufacturers generally mark the corresponding crown teeth with the dimension of the opening.

The wear on cast steel zig-zag rollers, as noted by the author, is shown shaded in *Fig.* 145. The top of the ridges wears more quickly than the bottom part, as the wearing section is smaller on the top. Special provision has to be made for reversal of the rollers and this is not always provided. This unequal wear has induced manufacturers to chill the top part of the ridges, when made of cast iron and a different hardness between top and bottom results, giving a more equal wear, without affecting the height of the teeth.

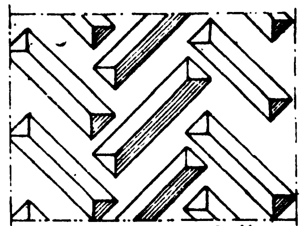


Fig. 146.—Open Flow Crusher.

In times of economic depression, such as have prevailed lately, the author has seen cast steel zig-zag rollers with the ridges built up by electric welding, as shown by the shaded area. This welding has to be done by an expert man, as otherwise the built up material will very soon break off. The thickness of the teeth does not matter so very much, where a squeezing action is wanted, thus by shortening the distance *a*. Good jobs have been done in this way and although the roller is inferior to a new one, it has helped out many a sugar factory in bad times. After each crop a thorough retouching of the worn spots has to be done. It should, nevertheless, be mentioned that frequent welding will cause heat tension stresses, which crystallize the material.

In *Fig.* 146 is shown an *Open Flow Type Crusher Grooving.* Here a continuous ridge is replaced by separate teeth of prismoidal shape. The juice-draining capacity is improved as compared with the standard zig-zag grooving.

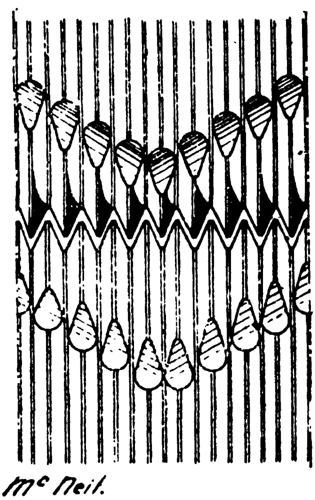


Fig. 147.—Meshing Grooves.

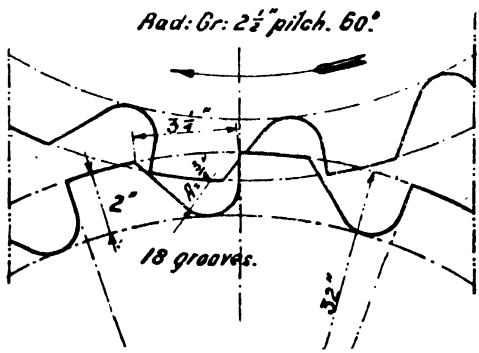


Fig. 148—Radial Grooving.

Rollers provided with ridges, as shown in *Figs.* 144-146, are sometimes called *splitting rollers,* as the cane is crushed and split in lengthwise direction. The author has seen very good work done by the zig-zag crusher, but this depends on a large roller diameter, high teeth, sufficient acting pressure on the top roller, and the necessary power input.

The gripping effect of the ridges is great and where cane is ground at full length, it will be efficiently dragged into the crusher; and these crushers are widely used in Java for this reason. In Hawaii the zig-zag crusher also finds a nearly universal application, whereas in Cuba and the Philippine Islands it has almost completely disappeared.

In the same year, 1883, when the zig-zag crusher was put into practice, *Meshing or Radial Grooves* for crusher rollers were invented in Great Britain. These grooves, sometimes called V-grooves, are turned on a lathe on rollers of special coarse-grained cast iron. The top angle is normally 60° and the pitch may vary from $1\frac{1}{2}$ in. to 4 in., according to size and capacity of the crusher.

The rollers are furthermore provided with zig-zag cross grooves, as shown in *Fig.* 147, having the same depth or nearly so as the radial grooves. The number of longitudinal grooves depends on the diameter of the roller and the circular pitch is about 8 in. as an average.

In *Fig.* 148 is shown a cross section of a roller with radial grooving, as measured by the author, having $2\frac{1}{2}$ in. longitudinal pitch and a top angle of 60°. There are 18 longitudinal grooves, 2 in. deep, and one side—the gripping side—is cut radially, whereas the other side is slanting. The rollers rotate as shown by the arrow and the cross grooves are staggered on top and bottom rollers, to ensure a good gripping effect.

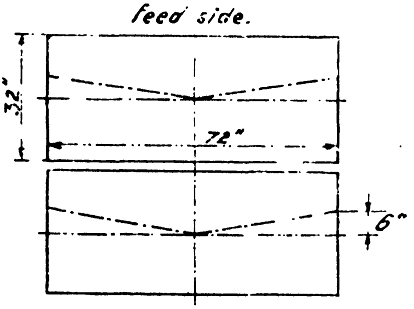


Fig. 149.—Inclined Axial Grooves.

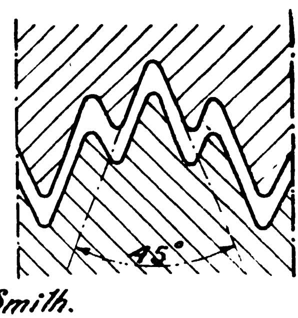


Fig. 150.—Compound Radial Grooving

The slope of the longitudinal grooving is shown in *Fig.* 149; it slants towards the centre line of the rollers, when seen from the feeding side. As the grooves run parallel on top and bottom rollers as shown, it is easily seen that the grooves will cross at the intermeshing position.

The radial or V-grooves will create an increased roller length, as according to formula (56) with a top angle of 60°, the effective roller length will be :

$$L_{eff.} = L \div \sin 30° = 2L.$$

The blanket, therefore, is increased twice in length when under compression, whereas the thickness is reduced to half the measure of the ungrooved roller.

To combine the effect of coarse grooving with one of fine grooving, a *Compound Radial Grooving*, as shown in *Fig.* 150, has been designed. Such rollers will give a better performance, although the damage done by passing pieces of tramp iron will be heavier.

According to publications of the Java Experimental station both crusher types, viz.: zig-zag and radial grooved, have equal advantages and neither need be preferred.

In Cuba the general adoption of the latter type was due to different reasons, one of the first being the increased diameter of radially grooved crusher rollers, whereas zig-zag rollers had a smaller standard diameter corresponding with mills of the same roller length. Thus for obtaining an increased grinding capacity, which was essential at the time, the radial grooved type had the preference in Cuba. Moreover the criss-cross loading of the cane on the carrier gave equally good gripping effect for the V-grooves, so there was no inconvenience in this direction. It should also be mentioned that the V-grooves could be easily brought to the original shape and sharpness on the factory roller lathe or kindred equipment, and although the lifetime of a cast iron roller is no greater than that of a cast steel one, the purchase price is lower.

In practice steel rollers become polished while cast iron rollers, made of a coarse grained cast iron, similar to the mill rollers, retain their rough surface. The depth of the zig-zag grooves is 2 in. on an average, whereas with radial grooves 3½ in. is the limit with the largest pitch, so a better crushing is obtained when grinding heavy amounts of cane.

More recently many mills have been fitted with revolving knives and the crusher is now doing a larger share of the extraction performance. As the V-grooves have a decidedly better drainage capacity than the zig-zag grooves, there have been many advantages from their adoption in Cuba.

2.—Crusher Chutes and Scrapers.

The cane chute for short cut or knifed cane should be as long as possible, as it will materially increase the easy feeding of the crusher, as explained before in Chapter IV.

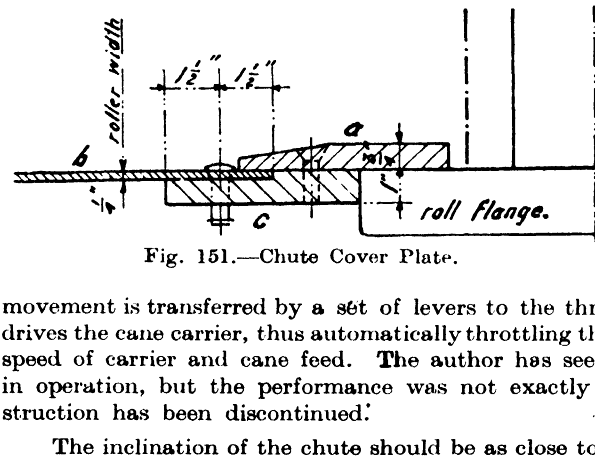


Fig. 151.—Chute Cover Plate.

There is a patented construction, having the cane chute *pivoted at the lower end,* outbalanced by a counterweight. When the chute becomes loaded to its limit, the weight of the cane will tilt the chute a few inches, and this movement is transferred by a set of levers to the throttle of the engine which drives the cane carrier, thus automatically throttling the engine and reducing the speed of carrier and cane feed. The author has seen several of these outfits in operation, but the performance was not exactly as desired and the construction has been discontinued.

The inclination of the chute should be as close to 45° as possible to ensure a positive sliding of the cane down to the crusher. As the cane exercises a lateral pressure on the walls of the chute, the lower end is provided with *cover or thrust plates* of sufficient strength, as shown in *Fig.* 151. The lower crusher roller is provided with cast steel flanges and the ¾ in. coverplate *a* is located inside these flanges. The walls *b* of the chute are ½ in. thick, and to give a firm connection a backing plate c of 1 in. thickness is arranged on the outside of the chute.

The wear on the inside coverplate is heavy, and countersunk bolts are used for attaching the plates together. The feed side of the inside cover plate is tapered to prevent choking.

The *Position of a Cane Chute* for a 32 in. crusher is shown in *Fig.* 152; the distance *x* depends on the fineness of the cane and it can be reduced when revolving knives or similar preparers are used. It is nevertheless of no paramount importance, as the author knows several instances where revolving knives have been installed without changing the cane chute, with no difference in feeding performance. Contrariwise, it is of much more importance when the space between the top roller and the chute bottom is too small, as this will choke the crusher feed.

The end of the chute y has to remain clear of the crusher bottom roller by about $\frac{3}{8}$ in. when zig-zag rollers are used, as the juice has to pass this clearance. For radial grooving, especially when the teeth are of large pitch, the chute end should have corresponding teeth to clear about $\frac{1}{2}$ in. from the roller periphery. This applies particularly when cane knives are used, as otherwise a large amount of fine cane will pass and accumulate on the juice pan or screening plate below the crusher rollers. It is advisable to have a plate with slotted holes, which can be adjusted to the bottom roller as it wears, and thus make the clearance of the chute bottom as small as possible.

So as not to cause too heavy pressure against the roller flanges, the *End Grooves* of the rollers are executed as shown in *Fig.* 153. The distance a is made larger than the normal vertical crusher opening, so the pressure exercised by the cane on the flanges will be less. The clearance between the flanges and the top roller is about $\frac{1}{16}$ in. on each side. Through wear of the flanges, this clearance will increase, and the flanges have from time to time to be faced on the lathe and, therefore, should have sufficient original thickness.

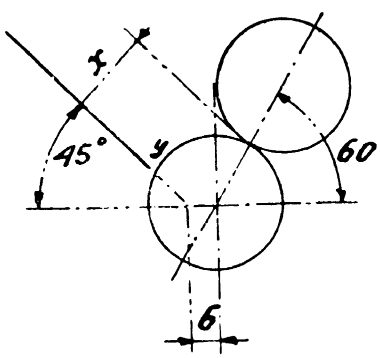


Fig. 152.—Cane Chute.

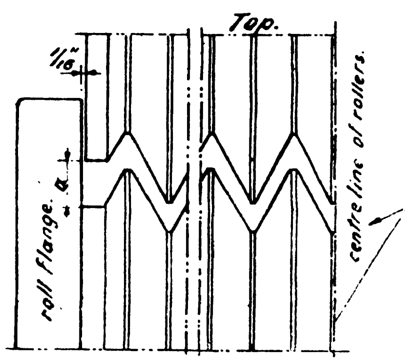


Fig. 153.—End and Centre Grooving.

The *centre groove* is generally on the lower roller, whereas the top roller has a ridge at the mid centre of the roller. In Cuba and U.S.A. the top roller is called the male and the lower roller the female one.

In *Fig.* 154 is shown the executed arrangement of the *Chute and Bottom Roll Scraper* between a 25 in. × 48 in. crusher and a first mill of $26\frac{1}{2}$ in. × 48 in. The radial grooves have to have scrapers as otherwise they would clog with bagasse.

The scraper plate is mounted on a shaft 3 in. square, laid in two bearings against the crusher housings and held in position by a spring-loaded lever on each side. The scraper plate is of $\frac{5}{8}$ in. sheet material, provided with several rows of countersunk holes for adjustment by bolts on the scraper shaft. The scraper tip is of cast iron, so wire-edging does not take place, and it is easily removable as being subject to heavy wear. The cast iron tips do not cause any polishing of the crusher roller. The spring pressure on the scraper levers should be gentle, about 150–175 lbs. on the lever end.

The scraper on the top roller is not curved but straight, and has also a cast iron scraper tip. There is a patented construction having the scraper shaft bearings attached to the crusher top roller bearings, thus moving up and down with these. This construction is of especial value, when uneven lifting of the top roller is frequent. For normal crusher performance the scraper shaft bearings are attached to the housings.

Pieces of tramp iron, when passing through the crusher, will produce a wire edge on the roller grooving and damage or breakage of the scraper tips may result.

The crushed cane emerges from the V-grooved rollers as a continuous blanket and this has led some operating engineers to apply a hinged cover plate on top of the cane in the chute to the first mill, so that the blanket will be slightly compressed and choking of the first mill will thus be prevented.

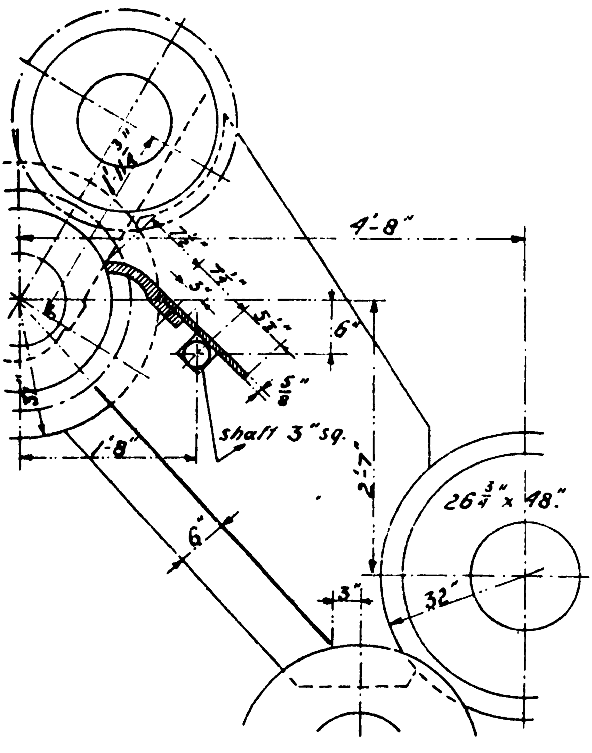


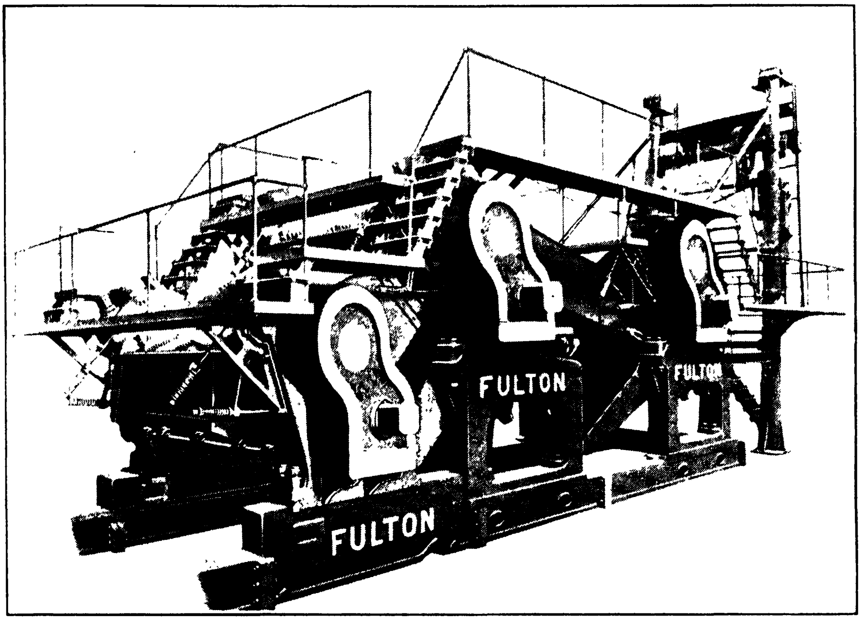
Fig. 154.—Chute and Scraper for 25 in. × 48 in. Crusher.

This choking of the first mill is due in first instance to insufficient drainage of the front roller of this mill, as the large amount of juice acts as a lubricant and counteracts the gripping of the crushed cane by the rollers. Moreover, the distance between the top roller and the chute should be of sufficient size. For 36 in. roller diameter a distance of 12 in. is common practice. The author has seen cane grinding at the rate of 230 tons per hour with a fibre content of about 11 per cent. on a 36 in. × 84 in. tandem without any anti-choking or feeding equipment.

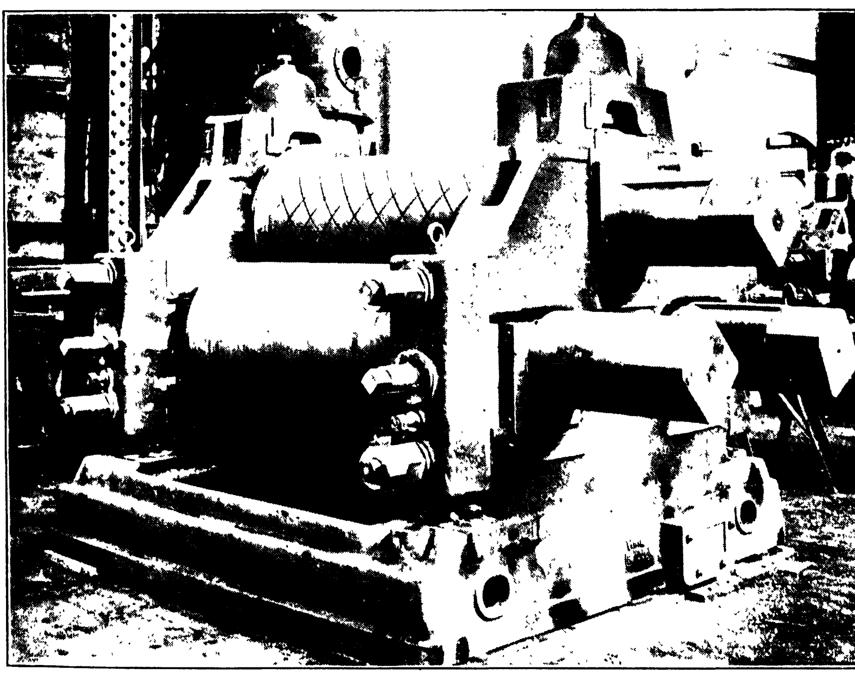
3.—Crusher Arrangements.

The crusher, being a two-roller mill, is placed in front of the following three-roller mills. An attempt has been made by a leading engineering firm to grind the cane by two-roller mills only, to avoid the power input for the trash bar friction. This arrangement has not found adoption in practice so far.

In *Fig.* 155 is shown the *General Arrangement* of a crusher and a first mill. The crusher is supported by two columns on the factory floor and the slanting crusher housings are firmly supported by inclined cast iron brackets on the first mill housings. The two vertical columns are well braced, so as to avoid vibrations during operation.

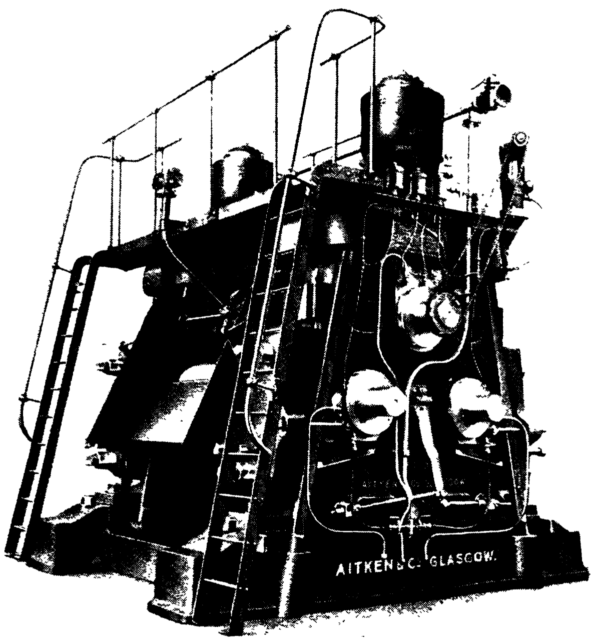


TRIPLE CRUSHER FOR CENTRAL " GUIPUZCOA," CUBA.
(Fulton Iron Works Co., Inc.)

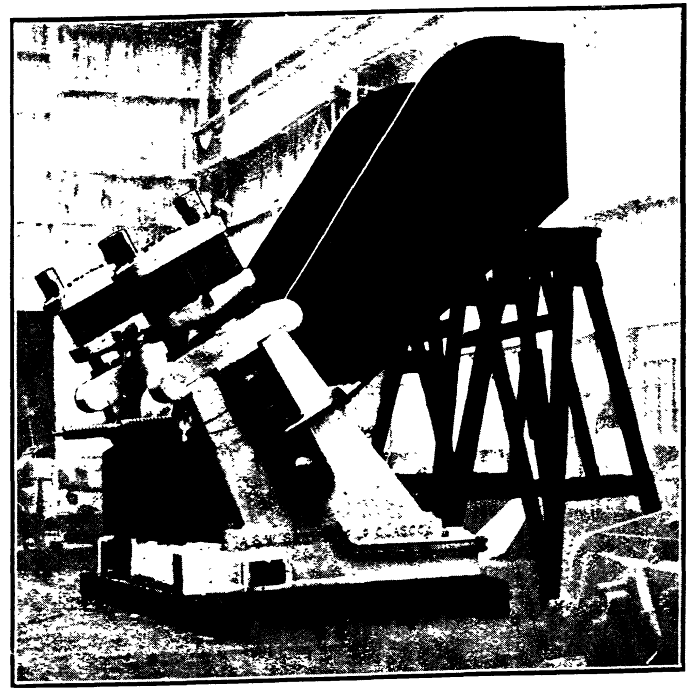


CANE MILL WITH VAN RAALTE GROOVES ON TOP ROLLER.
(Hallesche Maschinenfabrik)

PLATES 27 & 28.



CANE MILL WITH "LA CORONA" HYDRAULIC REGULATOR
30 in. x 60 in.
(H. W. Aitken Co., Ltd.)



30 in. x 72 in. CRUSHER-SHREDDER
(A. & W. Smith & Co., Ltd.)

A cast iron pan is bolted between the two crusher housings and the juice discharge flows down a steep chute to the first mill juice pan.

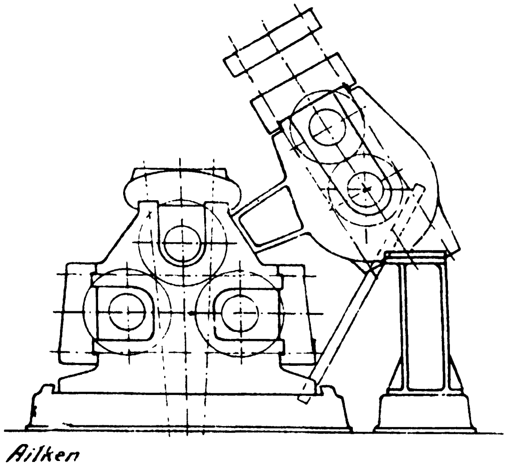


Fig. 155.—Crusher and First Mill.

In *Fig.* 156 is shown a *Rebuilt Crusher* designed by the author. The old installation comprised cast iron crusher housings with short bolts, having nuts imbedded in the former. The rollers were of the zig-zag type, 30 in. in diameter, and hydraulic caps were added as an improvement in lieu of the existing springs. The result was that the housings cracked along the indicated line and the whole top roller fell on the first mill, causing considerable damage.

The former arrangement is shown by shaded lines in the *Fig.*, whereas the new arrangement is shown in ordinary lines.

The housings are made from cast steel, as is now the general custom, and are provided with through-going bolts secured to the supporting brackets below. The right hand bolts are threaded on both ends, whereas the left hand

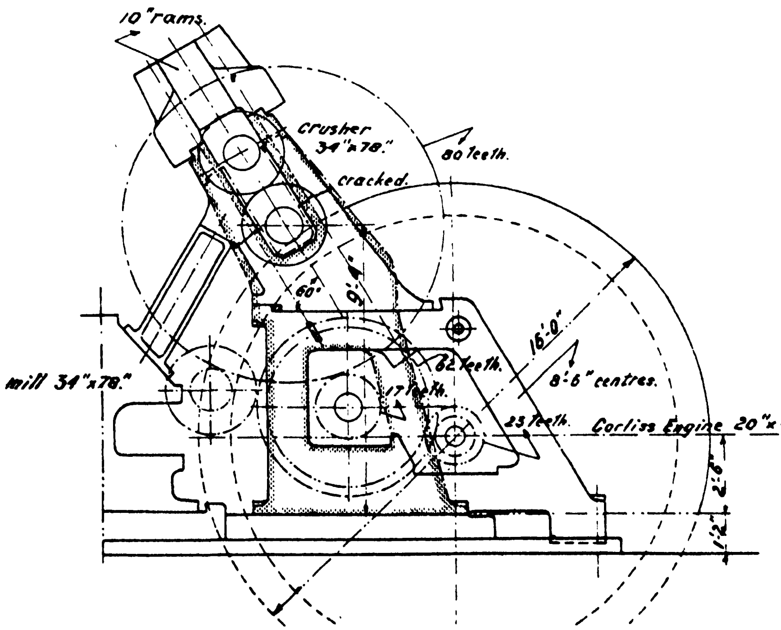


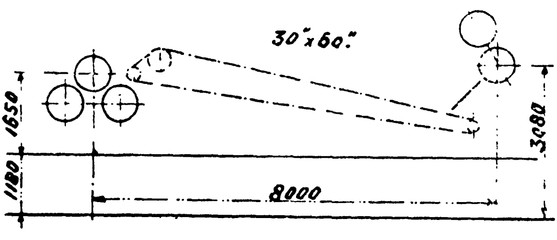
Fig. 156.—Rebuilt Crusher Arrangement.

ones are keyed into the brackets, as the bearings for the intermediate gearing were placed in openings in these brackets and did not allow sufficient clearance for threaded construction.

The supporting brackets are also made of cast steel, firmly anchored to the existing base plate and interconnected by a cast iron distance piece with a throughgoing bolt. On the extreme right hand a firm anchorage to the foundation has been provided.

The juice pan, of cast iron, is firmly bolted between the crusher housings, as it has to keep clear of the intermediate gear. It has a good slope towards the juice gutter, so all the trash or small pieces of cane can be flushed to the first mill juice pan.

The crusher housings are, moreover, firmly supported on the mill housings by removable slanting supports of H-section. The front roller of the first mill can be removed as indicated by dotted lines.



Stork.

With the old equipment about 2000 metric tons of cane per 24 hours was ground; but the power input was very low as the patched housings did not allow the crusher opening to close well and the normal engine load was 127 i.h.p. at about 56 r.p.m.

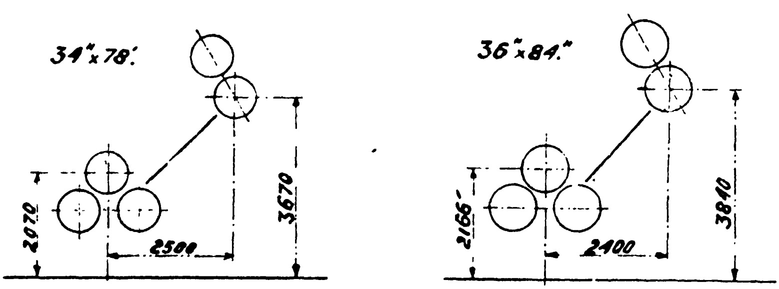


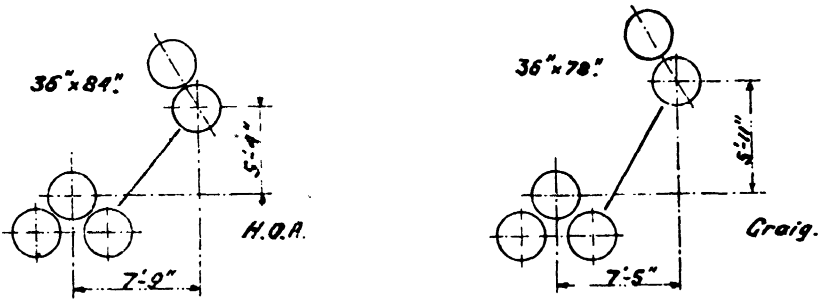
Fig. 157.—Java Crusher Arrangements.

The engine has now been reinforced and a new 16-ft. flywheel has been added, as well as a new intermediate gear. All the gear teeth were and are machine-cut to reduce friction, and the new bearings are provided with automatic lubrication.

With 34 in. rollers, having V-grooves, the capacity was brought up to 2,800 metric tons of cane per 24 hours, applying about 280 tons hydraulic pressure on the top roller. The crushing performance has been very satisfactory, with a normal power input of 256 i.h.p. at about 70 r.p.m. of the same engine.

In *Fig.* 157 are shown different *Crusher Arrangements* as put to work in Java. The top one, being a 30 in. × 60 in. installation, has a low level crusher, which discharges the crushed cane on a carrier to the first mill. The other two are 34 in. × 78 in. and 36 in. × 84 in. installations, having gravity chutes to the first mill. As will be seen, the tendency prevails to bring the crusher as close as possible to the first mill.

In *Fig.* 158 are shown a few *Crusher Arrangements* operating in Cuba. The heavier sizes prevail here. The lowest installation is of the older type with short vertical distance between the top roller of the first mill and the bottom roller of the crusher. The other two have the crusher at a higher level. The horizontal distance from centre of lower roller to centre of the top mill roller is in most cases around 7 ft. 6 in.



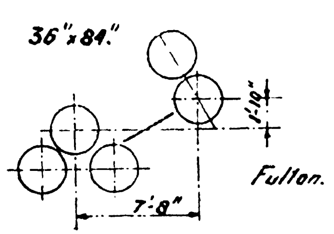


Fig. 158.
Cuba Crusher Arrangements.

In Cuba the double crusher has found many applications, and even the triple crusher has been installed in several sugar factories. The merit claimed for these latter installations was their increased milling capacity, and at that time, when revolving knives were not widely used, favourable results were reported; but at present the heaviest grinding capacities in Cuba are achieved with single and double crushers, the triple ones lagging behind.

4.—Crusher Details.

Many details of the crusher are similar to those belonging to the mills, and therefore some features only will be dealt with in this chapter, the rest being found in the one on mills proper.

The fitting of the roller shells (which are made from special coarse-grained cast iron having initial roughness, but generally improving through the action of the juice) to the steel roller shafts has been accomplished in different ways, the oldest method being a loose fit with a number of keys (eight or more); but present shop methods are of such accuracy that a tight fit can now be achieved, so keys are superfluous. Nor are juice rings, shrunk on the shaft to hold the shell in its place, any more used to-day. This has greatly reduced the equivalent manufacturing cost of the rollers.

In *Fig.* 159 is shown a *Press Fit Roller* of size 36 in. × 84 in., having 19 in. × 28 in. journals. The shaft has five steps of different diameters, stepping down by $\frac{1}{32}$ in. each. As there will be considerable abrasion when the shaft is pressed into the shell by hydraulic pressure, it is obvious that the best results are obtained by using as many steps as possible, varying between 15 steps with no cores for highest class manufacturing, and five steps as shown in *Fig.* 159, which is standard practice in many good shops. The largest diameter is 21$\frac{1}{2}$ in., stepping down by intervals of $\frac{1}{32}$ in. to 21$\frac{3}{8}$ in. The crown wheel trunnion has to be of smaller diameter than the fitting part, so that it will pass freely through the shell. On the left side is a collar to enable the shell to be pressed exactly to the desired place.

The centre part of the shell bears on the shaft, to increase the section modulus, and a centre core, once used, has been abandoned, as it has been the cause of many a shell breakage at this spot. The two small cores next to the centre fit are bored and as the shaft is covered with white lead paste when it is inserted, this lubricant can accumulate there. On the outside, it is released by grooves cut in the shaft for this purpose.

The hydraulic pressure needs to be watched closely when the shaft is pressed into the shell, and should increase gradually. Any abrupt rise in pressure indicates something wrong and the shaft should be taken out again. The maximum pressure for a 36 in. roller is about 1200 short tons.

In the diagram at the bottom of Fig. 159 is shown the intensity of the fitting to the shaft. The shaft as well as the shell fit will suffer by abrasion and this abrasion is more pronounced on the shaft at the end of each fit and on the shell at the commencement. Due to the fact that the shaft material is of a closer structure than the shell material, abrasion will be less on the former, as indicated in the diagram.

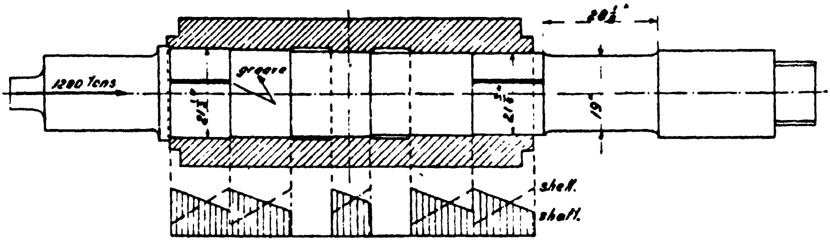


Fig. 159.—Press Fit Roller.

The press allowance is 0·018 in. for this shaft or a little less than one thousandth of an inch per inch mean shaft diameter. It is, of course, of paramount importance that the shell as well as the shaft be finished to true dimensions. The shafts can be removed from worn shells also by hydraulic pressure, when close to a local engineering works, as otherwise the freight abroad would be prohibitive. Sometimes the shell is heated before the hydraulic pressure is applied for removal, as the shaft and shell might have oxidized.

Operating performance with press-fitted rollers is usually a good one, as it seldom happens that a shell will turn loose on the shaft or will crack from too great a fit allowance. A press-fitted roller should not be heated when inserting the shaft.[1]

In the case of *Fig.* 159 the total fitting surface of the shell is 4700 sq. in. so the friction resistance per sq. in. will amount to :

$$2{,}400{,}000 \div 4700 = 510 \text{ lbs./sq. in.}$$

The friction coefficient being taken as 0·2, the normal peripheral pressure on the shell will then amount to : $510 \div 0{\cdot}2 = 2550$ lbs./sq. in.

The cylindrical shell has a mean inside diameter of 21·4 in. and a mean outside diameter of, say, 32 in. through the radial grooving, and the adhesive shaft length is 70 in., giving a total adhesive shaft projection of : 21·4 × 70 =

1 See also Proceedings A.T.A. Cuba, 1928, M. J. GALAINENA, p. 58. 1928, H. J. B. SCHARNBERG, p. 93.

1498 sq. in., and the total inside pressure within the shell will be 1498 × 2550 = 3,720,000 lbs.

The area of the shell material to withstand this force is obviously: 90 × (32 — 21·4) = 954 sq. in., and the stress in the cast iron shell is therefore :

$$3{,}720{,}000 \div 954 = 4000 \text{ lbs./sq. in.}$$

The formula, therefore, can be written :—

$$t = \frac{P}{\pi \times \mu \times (D - d) \times L} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (63)$$

where : t = tensile stress in the shell material in lbs./sq. in.
P = hydraulic inserting pressure in lbs.
D = mean outside diameter of shell
d = mean inside diameter of shell bore } in inches.
L = length of shell
μ = friction coefficient between shaft and shell.

This indicates that with an increasing friction coefficient the stress in the shell material will decrease and rough surfaces therefore will increase the adhesion. There is a limit, as abrasion will increase and the hydraulic pressure at the end may drop as the adhering material gets worn away.

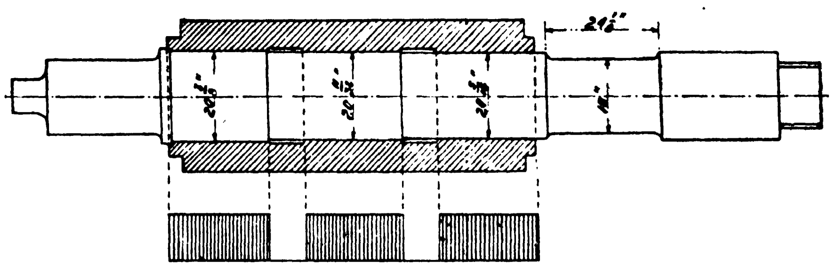


Fig. 160.—Shrunk Fit Roller.

In *Fig.* 160 is shown a *Shrunk Fit Roller* of size 36 in. × 84 in., having 18 in. × 24 in. journals. The shrunk fit has no abrasion, so three fits only will be necessary. The shaft diameter and the shell bore have to be finished as accurately as is the case with press-fitted rollers. The shell is heated to about 200°C., generally in a vertical position and the shaft lowered into the shell bore. The shrunk allowance depends on the coarseness of the material, as the shell material will be compressed at the points of contact. The shrunk allowance for the roller shown is 0·015 in. and the maximum applied by manufacturers is 0·025 in. The shell material around the bore is sometimes of a softer structure than the outside material and a larger shrunk allowance can be applied. From formulae (21) and (21a) in Chapter II the shrinkage performance is explained and as allowable stresses in cast iron are far below those of steel, it will be seen that the penetration or compression of the shell material requires an allowance far in excess of the true fitting allowance. Moreover, tensile stresses in cast iron are variable and manufacturers know by experience the shrinkage to be allowed.

Shrunk-fitted rollers have been known to crack but this occurrence need not to be more frequent than with press-fitted rollers. The adhesion between roller and shell material, under equal manufacturing conditions, is more perfect with the shrunk-fitted roller.

Cast steel crusher rollers are much more ductile, and will allow much higher tensile stresses than cast iron and, therefore, the danger from cracking is less.

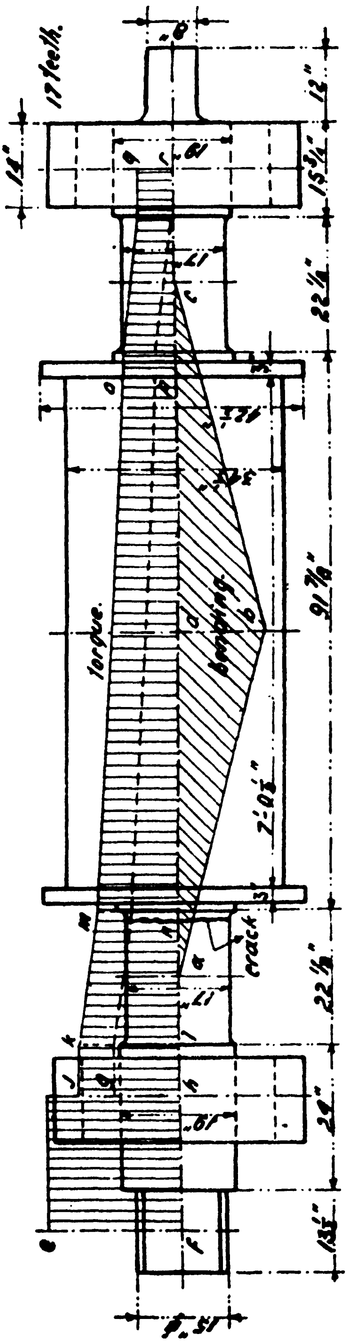


Fig. 161.—Crusher Roller with Double Crown Wheels.

The *casting shrinkage* for cast steel is larger than for cast iron and for unfinished crusher rollers the pattern for cast steel should have 2 per cent. excess dimensions, whereas for cast iron 1 per cent. only is sufficient.

A third method of attaching the shells on the shafts has been advertised lately. It is called the *bond lock* method and comprises the use of a bonding agent between the shaft and the shell. The shell, therefore, has an unfinished bore with longitudinal grooves to give a large bearing surface to the bonding agent. The diameter of the bore is about two inches larger than the shaft diameter and the shaft is provided with welded ridges to ensure a better bonding of the material, which has to be cast in between the shaft and the shell when placed in vertical position. The author has no data at hand, to show how this method has succeeded in practice. There is no doubt that it will, if successful, revolutionize this special operation. The bonding agent has a low melting point and the worn shell can be as easily removed as it has been attached. About the composition and shrinkage coefficient of the bonding material no data are as yet available.

In *Fig.* 161 is shown an executed *Crusher Roller with Double Crown Wheels*. There is a difference of opinion amongst operating engineers, whether double or single crown wheels should be employed; and as the author has supplied both types, he is able to give the viewpoints which can be taken into consideration when making a decision.

The total face of the double crown wheels is larger than the face of a single crown wheel, so wear will be less, when proper intermeshing is secured on both sides of the roller. The price of the double crown wheels is higher and they need a longer shaft.

Double crown wheels have to be machine-moulded and the tooth form has to be made accurately as well as the keyways, so that both crown wheels are in mesh. The author has seen cases where only the crown wheels on one side were in mesh and on the other side were running idle. Wear on the meshing set will of course then be heavy.

Moreover, tooth reaction may cause the closing or opening of the rollers on the crown wheel side, and different hydraulic rams should be used with single crown wheels. This theory is very old, but it is very difficult to make sure that uneven lift of the rollers is caused by the crown tooth reaction only, as uneven feed plays a paramount rôle too. The author has measured many rollers of the single crown wheel type, but conical wear due to uneven pressure or uneven feed has not been found to be of major importance.

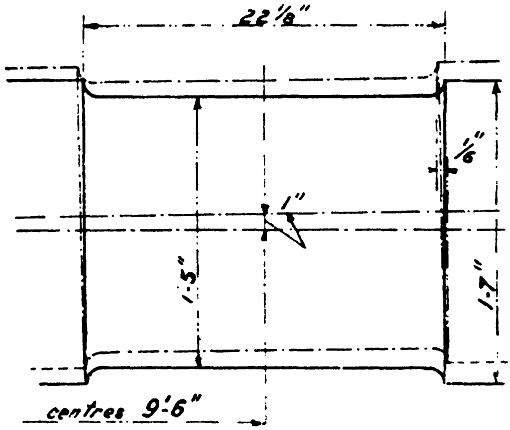


Fig. 162.—Lateral Play.

The shaft in *Fig.* 161 is subject to a bending moment, as indicated by the triangle *a-b-c*, the maximum being at the centre *d-b*. Here only the shaft bending moment is considered, and as the shell also carries a part of this moment, shaft breakages at the centre line seldom occur.

The roller shaft is also subject to a torque and for the driven roller this is composed as follows :

1.—Friction torque of the right hand journal.
2.—Dragging and friction torque of the shell.
3.—Friction torque of the left hand journal.

The total torque for this roller reaches the value *gh* and as there are two rollers in a crusher, it will be obvious that the torque *gh* is half the total torque *ef* when crown flank friction is neglected.

With the single crown wheel arrangement the total torque only acts over a roller length *fh* and at the spot *h* 50 per cent, is transferred to the crown wheel of the second roller. In the case of a double crown wheel arrangement and assuming that both sets are transmitting equal forces, half the torque of the second roller *jg* has also to be transmitted by the driven shaft.

The driven shaft of the one crown arrangement, therefore, is subject to the bending moment and the torque of one roller only, whereas with the double crown arrangement the driven shaft is subject to the same bending moment plus 1½ times the torque of a roller. In case the off side crown wheels are in mesh and the others are not, then the torque of the two rollers has to be transmitted by the driven shaft. For three-roller crushers, the case is even less favourable.

The shaft of *Fig.* 161 cracked and broke on the left inside journal as indicated. It had crystallized, due to fatigue of the material.

The *Journals* of the crusher and mill rollers have to have *lateral play* in the journal boxes, as shown in *Fig.* 162. This play is to prevent the journal bearing flanges from breaking, when the roller is lifting unevenly. Let it be assumed that the roller lifts at one end one inch more than at the other, this being about the maximum that can happen under practical operating conditions, then the shaft has attained an inclination of 1 in 114. As the crown wheel hub in this case is 19 in. in diameter, the horizontal play required will therefore amount to : $1 \div 114 \times 19 = \frac{1}{8}$ in. The shaft is given $\frac{1}{8}$ in. horizontal clearance, whereas the journal boxes when new have $\frac{1}{16}$ in. clearance between the flanges and the crusher housings.

The *Journal Fillet* of a measured shaft is shown in *Fig.* 163, the journal being 18 in. in diameter and the hub 20 in., so there is a possible fillet radius of 1 in., but generally these fillets are smaller. The shafts generally break on the locus of the inside journal fillet, due to the fatigue of the material. This fatigue, according to our present-day knowledge, is due to cyclic or continuous stresses, which are concentrated at certain spots and which change the molecular structure of the material, causing crystallization and surface brittleness, this resulting in microscopic cracks which penetrate until the carrying area is reduced to such an extent that breakage will occur. Only seldom will a shaft break abruptly; generally the cracking process has advanced a few inches deep before final rupture takes place.

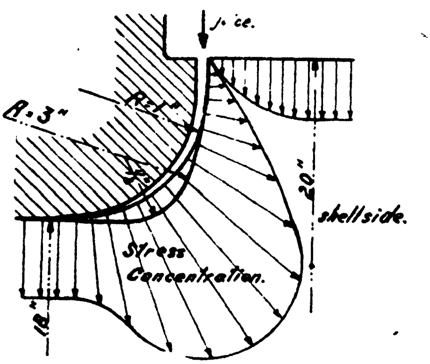


Fig. 163.—Journal Fillet.

This fatigue is due to several causes :—

1.—Bending stress fatigue.
2.—Torque stress fatigue.
3.—Corrosion stress fatigue.[1]

The bending fatigue limit is the last to be feared, as bending fatigue stresses take place beyond about 14 tons per sq. in., whereas the value for torsional fatigue lies at about 7 tons per sq. in. The corrosion fatigue is caused by the chemical action of the juice components on the shaft material or else by cooling water from the journal bearings. These liquids enter at the journal clearance or between the shaft and the shell. The author has seen several breakages of the shaft just inside the shell, thus in the strongest part of the shaft. This corrosional cracking is frequently noted when a shaft is reshelled. Care therefore should be taken to prevent the juice from having access to these places, although this is not always avoidable.

To counteract this phenomenon of fatigue, resort is now had to special materials, like nickel steel and the many alloy steels of to-day, and although these steels have a far higher tensile strength than common openhearth steel and have better wearing capacities, they nevertheless are less ductile and more subject to brittleness, and fatigue develops more quickly. Formerly, hollow shafts were used, as it was assumed that the cracking started from the centre or that microscopic cracks were present there from the forging or pressing process. But this practice has now been abandoned.

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[1] See *The Engineer*, 1936, Vol. 4147, p. 211

In *Fig.* 163 is also shown the magnitude of the stress concentration at the locus of the fillet and to overcome this inconvenience compound fillets are used in other engineering fields, having a large and a small radius fillet, which will give a more even distribution of stresses. This might be applied to crusher and mill roller shafts as well.

It is therefore of the utmost importance that the shaft material has a *high ductility*, and even material with a low tensile strength might prove to be superior for the purpose, as has been confirmed in many branches of present day engineering.

To repair worn shafts at the site of the journal or near to it, the author has frequently resorted to electrical welding, thus adding a fourth source of fatigue, viz., heat stresses, which indeed may be already present in the shaft. Welding on concave surfaces is only allowable when proper annealing of the shaft is effected afterwards, so as to restore the pre-crystalline structure of the material. This annealing is costly, as it requires specially constructed heating ovens. The author's experience of welding shafts without annealing has not been very satisfactory.

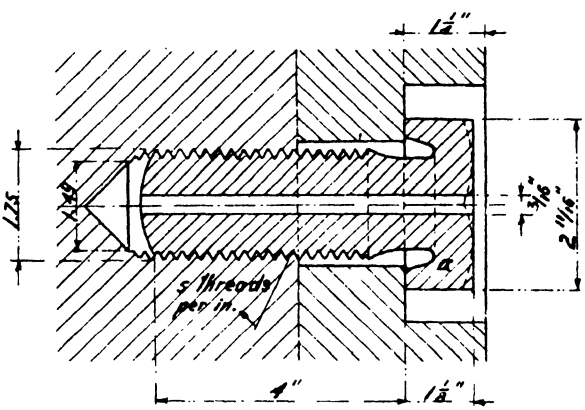


Fig. 164.—Flange Tap Bolts.

In *Fig.* 164 is shown the method of securing the flanges for crusher bottom or mill top rollers. The author has supplied many of these tap bolts or pins with a drilled core, to release air and especially oil that might have accumulated in the tap hole. The heads of the tap bolts or pins frequently break off and even special alloy steels do not give invariable satisfaction. Along the threaded part of the bolts, the material is already notched and fatigue will soon present itself. Material of low tensile strength will be good enough, or sometimes is to be preferred. The threaded part should end in a fillet and heads should be undercut as shown at *a*, so as to give good stress distribution. Dangerous fillet stresses are thus avoided.

The number of tap bolts in these flanges varies greatly and for 36 in. rollers from 12 to 16 or more are used with diameters from $1\frac{1}{8}$ in. to $1\frac{3}{4}$ in. The tapped holes for these bolts have to be placed in a pitch circle well within the smallest diameter the roller may obtain after it has been worn, as otherwise the shell material may break off at the locus of the tap holes.

The crusher top roller is loaded by spring or hydraulic pressure. Springs should be employed where irregular feed is to be feared, as too frequent action on the part of the hydraulic accumulator and rams will cause a heavy toll on the packing used therein. With short cut or knifed cane the hydraulic rams can be used to advantage. In Java both the spring and the hydraulic types are used, whereas in Cuba the latter is employed almost exclusively.

In *Fig.* 165 is shown a set of *Crusher Springs*, of which four are attached to each king bolt. To obtain initial pressure, the springs are tightened by pre-compression before they are inserted. When the pressure for each set has reached 17 tons, the top roller will lift and for one inch lift, the spring pressure increases to 25 tons. This is contrary to the hydraulic system, where the load on the top roller is nearly equal under all lifts, only slightly varied by inertia or acceleration forces of the accumulator or the oil friction within the hydraulic pipelines.

The spring-loaded crusher, therefore, will produce more abrupt bending stresses in the shafts than are caused by any equivalent hydraulic pressures.

It should be recollected that the empty crusher should not have the top roller load acting on the journal bearings; a stop has to be provided in the caps and a clearance on top of the bearing pressure plate of $\frac{1}{16}$ in. is normal practice, so that the spring or hydraulic load will come into action after the top roller has lifted $\frac{1}{16}$ in.

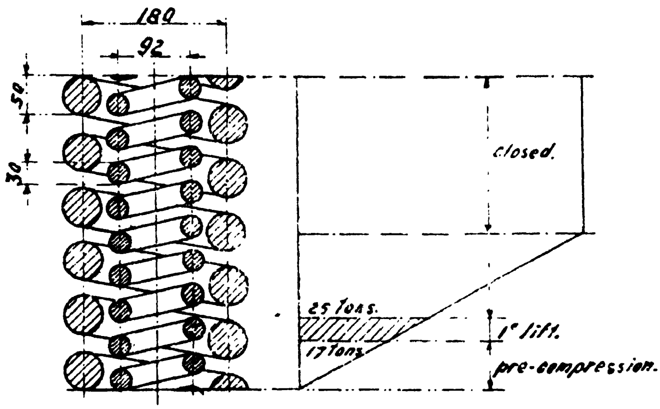


Fig. 165.—Crusher Springs.

In the diagram at the right of *Fig.* 165 it is seen how the spring pressure increases with compression. The material of the springs is subject to torsional stresses and they should have a sufficient number of free turns so as not to cause fatigue stresses. This especially applies to double springs as shown, and the normal stresses should be about equal for both springs, inner and outer, as otherwise the overstressed one will soon develop a breakage. The formulae for coil spring dimensioning can be found in any engineering hand book.

To adjust the crusher openings, packing plates were formerly used between the intermediate journal boxes. The adjustment could only be done by lifting the top roller, and present day crushers are provided with a *Wedge Adjustment* as shown in *Fig.* 166. Single or double wedges are arranged. The double wedge has the advantage that the same lift can be achieved with half the wedge slope and thus half the pushing force is required. The wedge bolts can also be arranged to act under tension instead of compression. The crusher housings have snugs cast integrally with the housings or attached by screws in the case of existing crushers. Bolts centrally arranged are used as well.

When adjusting the wedges, the space between the pressure plate and the hydraulic ram or spring bottom piece should be equally changed by adding or removing the packing plates *a*. These packing plates should be so arranged that they can be removed laterally. With varying grinding capacity or fibre content of the cane, as well as with wear of the crusher rollers, the application of these adjusting wedges is very convenient.

In *Fig.* 166 the lower half of the journal bearing *B* and the upper half of bearing *A* both receive the full pressure during operation, so they are water-cooled in the case of the larger sizes of crushers. Brass is mostly used for these water-cooled bearings.

The upper half of bearing *B* has only to carry the load of the top roller and rests on the lower half of *B*. To reduce friction there is a clearance provided between the lower shaft journal and the top of the upper half of bearing *B*, so it must be considered only as a spacer. The lower half of bearing *A* carries only the load of the top roller, when the crusher is empty, so it is generally provided with a few strips of white bearing metal. The intermediate half bearings (upper *B* and lower *A*) are as a rule made of cast iron.

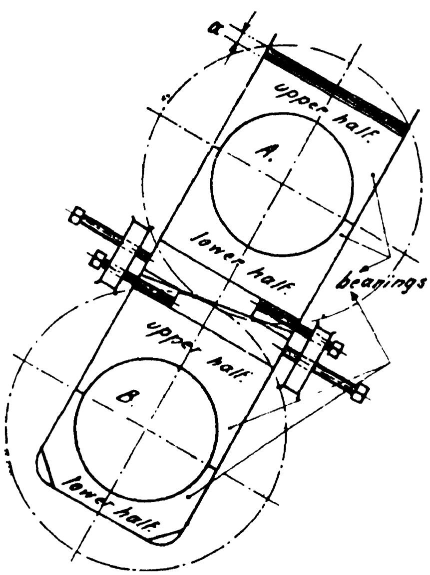


Fig. 166.—Crusher Wedge Adjustment.

With the zig-zag type of rollers, a spare roll should be kept with each crusher having two coupling ends and provided with detachable cast steel flanges, so that the roller can be used as either top or lower roller. For the V-grooved type interchangeability is only possible when the operating, as well as the spare, rollers have two coupling ends, as the whole set has to be reversed.

5.—General Data for Crushers.

The *capacity* of crushers is very flexible. The crusher grinds fibre and it is obvious that for grinding heavier amounts of cane a larger crusher opening has to be provided or the speed increased. It must, nevertheless, be borne in mind that with heavier amounts of fibre passing the crusher, the crushing performance will be less efficient, when larger pitch grooving or increased top roller load is not applied. This has led manufacturers to have double and even triple crushers, so as not to overload the milling train beyond. More recently the use of double and triple crushers has been less favoured by operating engineers and in Cuba a single crusher, 42 in. × 87 in., preceded by revolving knives, has ground up to 5000 tons of cane per 24 hours in an efficient manner.

The *power consumption* of crushers varies between 15 and 21 h.p./ton fibre per hour. The larger power consumption is due to the toughness of the cane and since such different varieties of cane as Uba, Cristalina, Coimbatore and the different POJ's and BH's can be met with at the mills, allowances should be made for the hardest canes in fixing the sizes of the driving engines or motors. The steam pressure at most tropical cane sugar factories varies between 90 and 125 lbs./sq. in., and the following table shows the crusher engines for those pressures[1]:—

Crusher. Number.	Inches.	Size of Executed Crusher Engines.
1	26 × 48	14 in × 30 in.
1	26 × 60	500 mm. × 900 mm. (20 in. × 36 in.)
1	26 × 72	20 in × 28 in.
1	30 × 60	575 mm. × 1000 mm. (23 in. × 40 in.)
1	32 × 72	600 mm. × 1000 mm. (24 in. × 40 in.)
1	32 × 72	400 mm. × 600 mm. triple gear (16 in. × 24 in.)
2	32 × 66	28 in. × 48 in.
1	32 × 72	22 in. × 42 in.
2	32 × 72	28 in. × 48 in.
1	32 × 78	24 in. × 48 in.
1	34 × 78	600 mm. × 1100 mm. (24 in. × 44 in.)
1	34 × 78	530 mm. × 550 mm. triple gear (21 in. × 22 in.)
2	34 × 78	28 in. × 48 in.
1	34 × 78	20 in. × 44 in.
1	34 × 80	22 in. × 40 in.
2	34 × 80	30 in. × 54 in.
1	36 × 72	22 in. × 48 in.
1	36 × 72	200 h.p. electric motor
1	36 × 84	26 in. × 42 in.
1	36 × 84	200 h.p. electric motor
1	36 × 87	24 in. × 42 in.
1	36 × 87	200 h.p. electric motor.
1	36 × 84	650 mm. × 1100 mm. (26 in. × 44 in.)
1	36 × 84	500 mm. × 550 mm. triple gear (20 in. × 22 in.)
2	37 × 84	30 in. × 54 in.
2	37 × 84	26 in. × 60 in.
1	42 × 87	26 in. × 60 in.
1	42 × 87	250 h.p. electric motor.

Journals on crusher shafts have gradually increased in size and the specific pressure between bearing and journal has reached 1100 lbs./sq. in. (80 kg./cm.²) The maximum journal dimensions of actual crusher installations in different countries, as gathered by the author, are :—

26 in. × 60 in.	crusher,	journals	13 in. × 15 in.
26 in. × 72 in.	,,	,,	13 in. × 17 in.
30 in. × 60 in.	,,	,,	14 in. × 18 in.
32 in. × 72 in.	,,	,,	16 in. × 22 in.
34 in. × 72 in.	,,	,,	16 in. × 22 in.
34 in. × 78 in.	,,	,,	16 in. × 22 in.
36 in. × 84 in.	,,	,,	17 in. × 24 in.

It is obvious that a larger roller diameter will allow a heavier journal diameter and as the larger roller has a better gripping effect, this indicates that large roller diameters are to be preferred.

The *top roller load* under spring or hydraulic pressure is as follows, but it should be recollected that as the hydraulic pressure can be easily changed by reducing the accumulator weights, all operating engineers do not strictly adhere to these data :—

26 in. × 60 in.	crusher,	150	tons	spring	pressure.		
26 in. × 72 in.	,,	170	,,	,,	,,		
32 in. × 72 in.	,,	170	,,	,,	,,	rams	11 in.
34 in. × 78 in.	,,	250	,,	,,	,,	,,	12 in.
36 in. × 78 in.	,,	250	,,	,,	,,	,,	13 in.
36 in. × 84 in.	,,	250	,,	,,	,,	,,	13 in. or more.

[1] See also : GILMORE, "The Cuba Sugar Manual" 1925, and *Het Archief*, 1925, No. 10, p. 356, etc.

The *peripheral speed* is in direct proportion to the capacity of the grinding tandem, and therefore an independent crusher drive is to be preferred. The crushing performance, moreover, will be obviously more efficient when a thinner blanket is passing through the rollers, as less cane has momentarily to be crushed by the same roller load.

In Java crusher speeds generally do not exceed 18 to 24 feet per minute, whereas the author has measured operating peripheral velocities on 10 Cuban installations, varying between 20 and 53 feet per minute. Second crushers are generally operated at a lower speed, but the author knows an installation where a higher speed was arranged for, thus avoiding the previously occurring chokes.

The *weight* of the crusher rollers without crown wheels amounts approximately to :

30 in. × 60 in.	9,500 lbs.
32 in. × 72 in.	18,000 lbs.
34 in. × 78 in.	24,000 lbs.
36 in. × 84 in.	29,500 lbs.

CHAPTER VII.

CANE MILLS.

MECHANICAL PERFORMANCE OF MILLING—MILL TYPES—
MILL DETAILS—GENERAL MILL DATA.

The completion of the extraction process, which has been initiated by revolving knives, shredders and/or crushers is accomplished by the cane mills; but it should be mentioned at the outset that finely shredded cane does not need to undergo the extraction performance by mills, but can be exhausted by a thorough maceration. The Vazcane macerator, used in combination with the Vazcane process (see Chapter V) achieves a continuous compound maceration with a sucrose extraction well over 99 per cent., without the use of cane mills. This process, nevertheless, is confined to fibre board manufacture from bagasse and the latter is discharged in a wet state, as it is not used as fuel.

In some sugar factories in Egypt the cane is subject to a partial milling process by two sets of mills and thereafter the bagasse (or megass, as it is called in some countries) is transferred to a diffusion battery, similar to those used in the beet sugar industry, for further extraction and is finally dried between mill rollers so as to reduce the moisture content to about 50 per cent., which is the optimum limit when the bagasse is to be burnt. But this practice is only adopted in a few isolated instances and the general custom in most cane growing countries is to grind the cane completely between mill rollers.

1.—Mechanical Performance of Milling.

From laboratory as well as by practical operating tests it has been found that by repeated compression of the cane tissue, more juice-containing cells are ruptured and thus the total extraction is increased. With a 10 to 12 per cent. fibre content in the cane, the total or normal juice extraction will amount approximately to :—

After the first mill and preparatory devices ..	60 per cent.
,, ,, second mill	67 ,,
,, ,, third mill	72 ,,
,, ,, fourth mill	75 ,,

without applying maceration or imbibition. The cane structure, peripheral speed and roughness of the rollers, hydraulic pressure on them, etc., have a bearing on the extraction performance.

The peripheral speed of the mill rollers affects the thickness of the blanket and there is a divergence of opinion among operating engineers in this respect, as in Java low speeds are preferred, from 24 to 15 feet approximately per minute, the last mill having the slowest speed, whereas in Cuba the tendency prevails to increase the mill speeds towards the last mill, and peripheral speeds of 50 feet per minute for that mill have been applied in some instances with good results. But it should not be overlooked that grinding capacities in Cuba are far in excess of those in Java per given unit and, therefore, the thickness of the bagasse blanket in both countries will not differ so greatly as sometimes is assumed.

Again, the physical properties of the cane play a large part, as a soft cane will be more easily squeezed than a hard one and therefore will be less subject to blanket thickness, whereas hard canes should preferably be ground at high speeds with a correspondingly thin blanket. The amount of fibre in the cane is also a predominant factor, as the mills *de facto* grind fibre, and higher peripheral speeds are required for high fibre canes, if capacities are to be maintained.

The true indication how far the mill speed for a certain mill setting can be raised can be observed from the hydraulic accumulators, which should be acting all the time. This applies to spring-loaded mills as well.

The author has had the opportunity of taking the power consumption of mill engines at different speeds in actual operation and has found that the power input generally is in proportion to the peripheral speed of the rollers and in some instances even more, which might suggest that more squeezing force was required with higher speeds, doubtless due to better extraction performance. The extraction figures increase and the moisture content of the bagasse decreases. In those countries where heavy amounts of cane are ground, the power input per ton of fibre is sometimes below the normal and a speeding up of the engines may be an advantage.

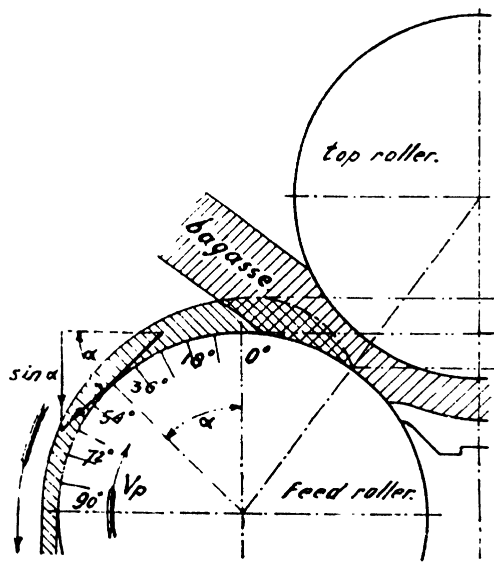


Fig. 167.—Juice Drainage on Front Roller.

In *Fig.* 167 is shown the *Juice Drainage* on the front roller of a three-roller mill. It will be obvious that the bagasse or discharge roller is more easily drained than the feed roller, as the extraction point of the latter is on the centre line of the rollers, and beyond the culmination point, and the juice therefore has to mount a vertical distance x before it will flow down the front side of the roller. Radial grooving of the roller as well as Messchaert grooving is of material assistance in draining, and for heavy grinding will be an indispensable factor.

Assuming g to be the gravity acceleration, the unobstructed falling speed of the juice will be : $V = g \times t$, where t is the time in seconds and $g = 32{\cdot}162$ ft. per sec.^{2} (9·81 metres per sec.^{2}).

On the roller surface the juice speed Vj is in proportion to the sine of the angle α with the vertical. At the culmination point 0° of the roller $\sin \alpha = 0$, and therefore a small hydrostatic head h is necessary to initiate the downward flow of the juice. At 90° the velocity will be equal to falling, the small adhesion between roller surface and the juice being negligible at this point.

Nevertheless this adhesion causes a friction when the juice runs down on the roller, and therefore the juice velocity will not reach the value $g \times t \times \sin \alpha$, but for any given point, the equation reads :

$$Vj = g \times t \times \sin \alpha (1 - \mu) \quad \ldots\ldots\ldots\ldots\ldots\ldots (64)$$

where μ is the friction coefficient between roller and juice.

The peripheral roller speed Vp acts against the juice flow, but is of too small importance (as is shown in *Fig.* 168) to interfere with the juice flow and only a very small hydrostatic head is required to overcome it. As the juice is pressed back by the milling performance, a positive juice flow is obtained at the mill speed found in our present day plants.

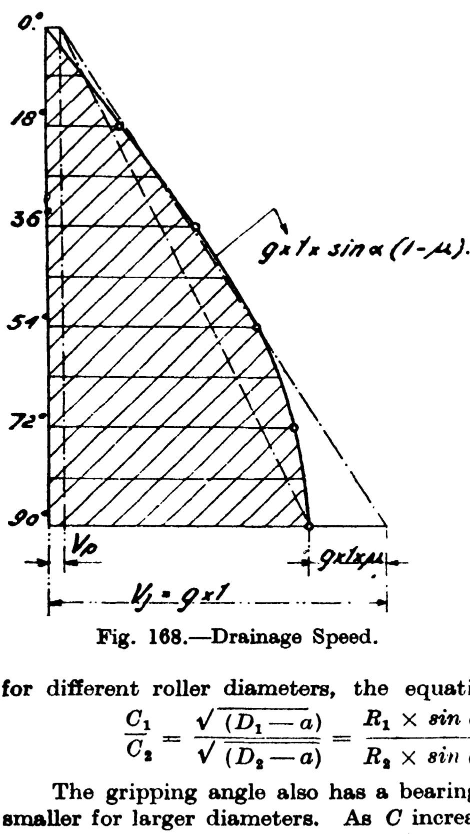


Fig. 168.—Drainage Speed.

The *Roller Grip*, as shown in *Fig.* 169, is obviously proportionate to the distance C, as there will be more adhesion between the bagasse and the roller, when C is larger, (C being represented in the Figure by C_1 and C_2 respectively).

From the Fig. is derived :—

$$C^2 = a \times (D - a) \text{ or}$$

$$C = \sqrt{a} \times \sqrt{(D - a)} = R \times \sin \alpha$$

where a is the compression depth and D the roller diameter, whereas α is the gripping angle.

For different roller diameters D_1 and D_2, the compression depth a is the same for the same mill setting and $\sqrt{a}$ therefore a constant figure, so the gripping effect depends only on the square root of $D - a$, and for different roller diameters, the equation prevails :—

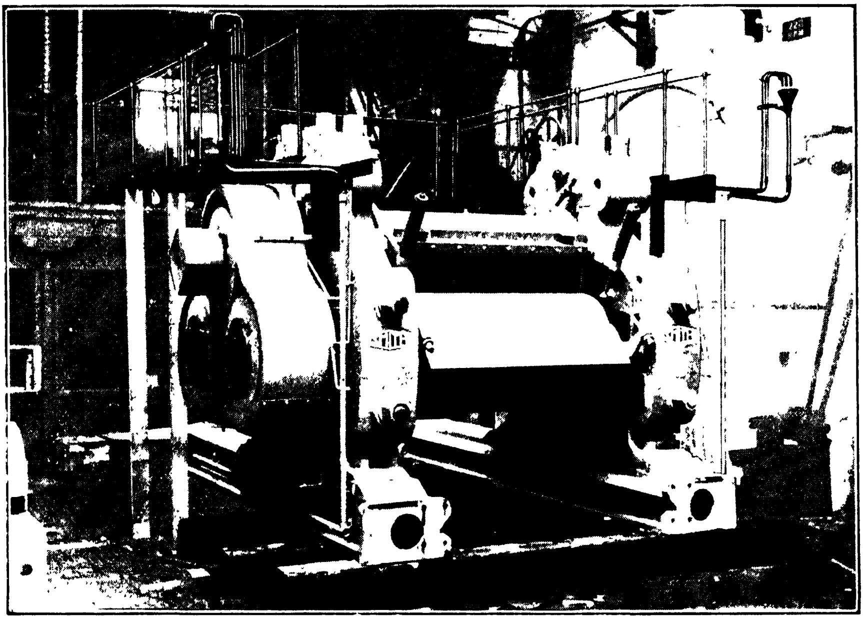
$$\frac{C_1}{C_2} = \frac{\sqrt{(D_1 - a)}}{\sqrt{(D_2 - a)}} = \frac{R_1 \times \sin \alpha_1}{R_2 \times \sin \alpha_2} \quad \ldots\ldots\ldots\ldots\ldots (65)$$

The gripping angle also has a bearing on the *roller slip* and this angle is smaller for larger diameters. As C increases with the diameter, it is obvious that larger roller diameters have a better grip on the cane.

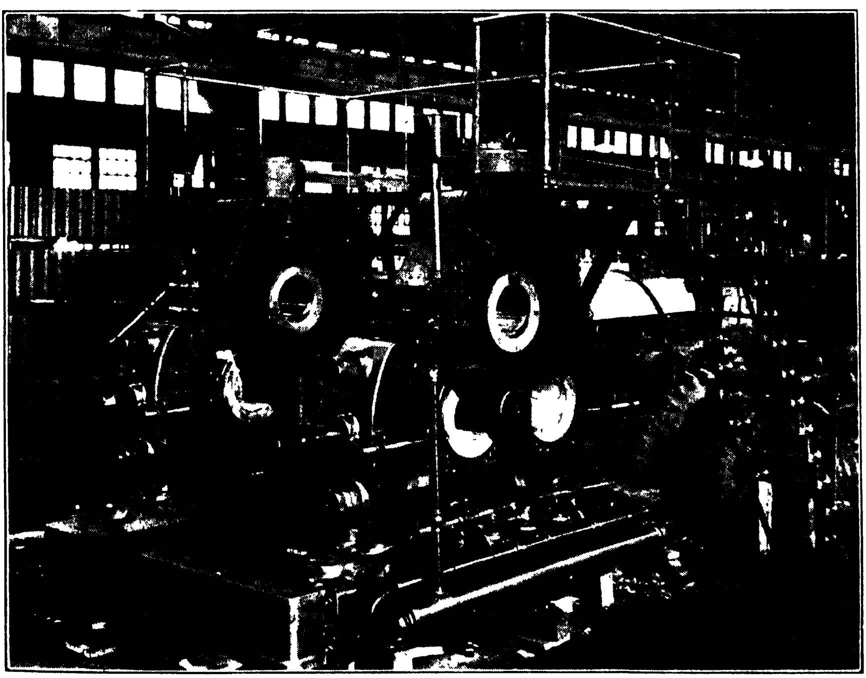
The slip depends on the friction between the bagasse and the roller and as the friction is proportionate to the pressure between these two, it is easily seen that the friction will gradually increase when the bagasse enters the mill, and slip is only feared at the moment when the bagasse is coming in contact with the rollers. As the receding juice acts as a lubricant, smooth rollers will more easily slip than grooved ones, where the juice can drain along the bottom of the grooves. The author on several occasions has avoided mill chokes by applying improved grooving on the front roller.

The force P tries to push the bagasse out of the rollers as shown in *Fig.* 170 and this force is equal to twice the force H, when assuming that the friction on top and bottom rollers is equal. The value of H can be derived from:—

$$H = D \times \sin \alpha = D \times \tan \alpha \times \cos \alpha \quad \ldots\ldots\ldots\ldots (66a)$$

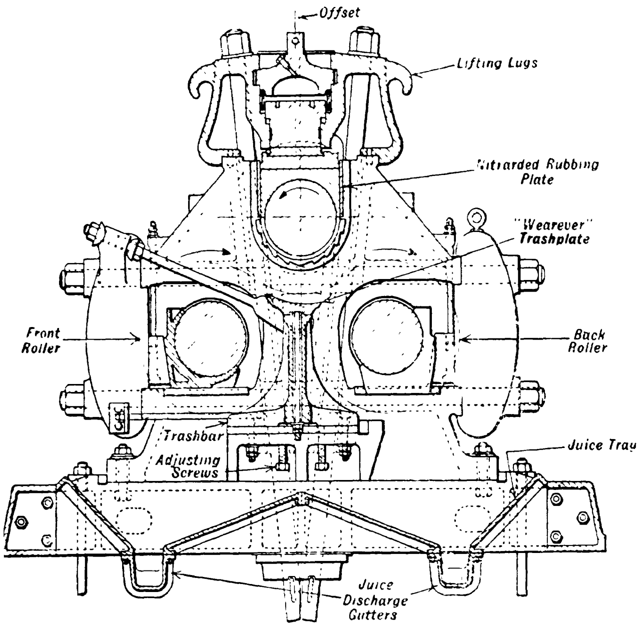


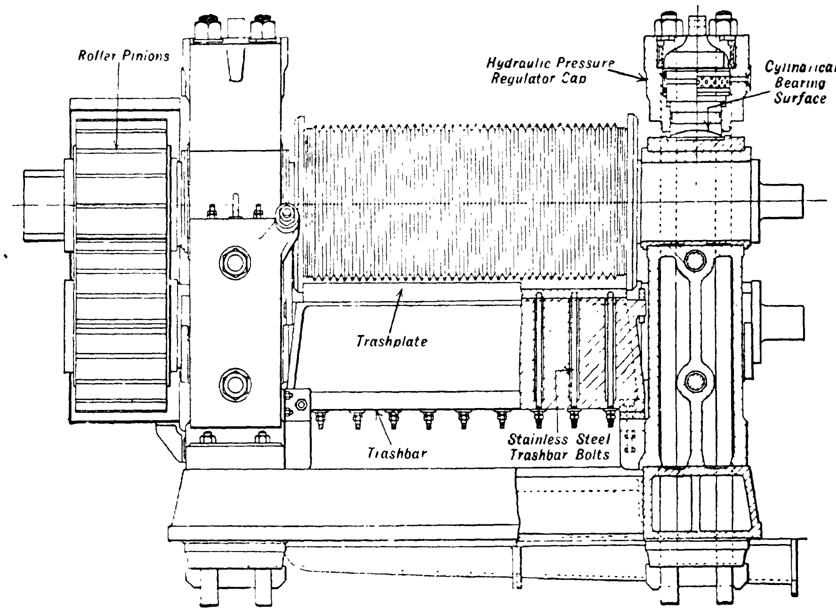
THREE-ROLLER MILL, 33 in. x 66 in.
(A. & W. Smith & Co., Ltd.)



"E" TYPE KINGBOLTLESS MILL HOUSINGS.
(Fulton Iron Works Co., Inc.)

PLATE 31.





SECTIONAL VIEWS OF 36 in. × 84 in. CANE MILL.
(*The Mirrlees Watson Co., Ltd.*)

where D is the diametrical force, acting at the bagasse entrance of the rollers. This radial force D obviously will produce a friction $D \times \mu$, and as μ is the tangent of the friction angle, the equation is written : $R = D \times \mu = D \times \tan \phi$, where μ is the friction coefficient and ϕ the friction angle.

The friction force R can be subdivided in two components V_1 and R_1, the latter having a value :—

$$R_1 = R \times \cos \alpha = D \times \tan \phi \times \cos \alpha \ldots\ldots\ldots\ldots (66b)$$

The friction therefore drags the bagasse into the rollers with the force $2R_1$ in perpendicular direction to the centre line of the rollers and this force has to be greater than the opposing force $P = 2H$, as otherwise the mill will choke.

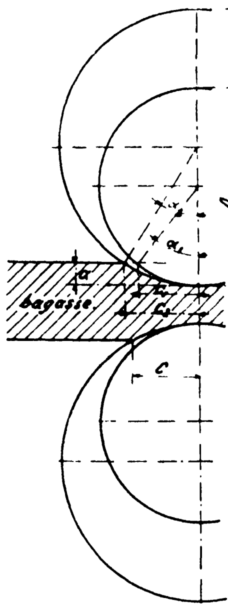


Fig. 169.—Roller Grip.

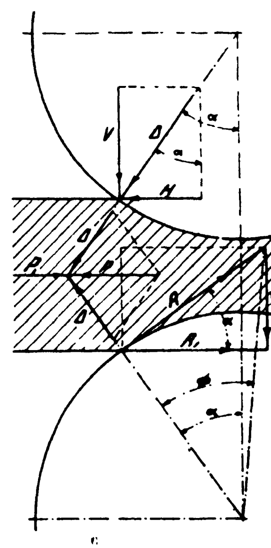


Fig. 170.—Roller Slip.

Comparing formulae (66a) and (66b) it will be noted that the friction angle ϕ has to be greater than the gripping angle α under optimum conditions. When this is not the case, a counter force P_1 has to be applied which forces the bagasse into the rollers, such as will be obtained by a slanting feed chute, a pushing bagasse carrier (Cuba type), a feed roller, or a set of mechanically operated pushers.

The roughness of the rollers is of the utmost importance in increasing the friction coefficient and it should be recollected that the rollers will become rougher through the action of the juice, and a brand new roller will need about three weeks grinding before full roughness is attained. There are manufacturers who "pickle" the rollers in a bath of dilute muriatic acid, to produce a rough surface before shipment. This "pickling" nevertheless is seldom applied. When the grooving permits one to do so, new rollers are sometimes used as discharge rollers until the "grain" is produced and then transferred to the front of the mill, as the cane or feed rollers have a more difficult task than the discharge rollers.

A much discussed factor in milling is the *Peripheral Speed of the Rollers.* This speed is generally in direct proportion to the roller diameter, so with larger rollers higher speeds are employed. The practical limit is not yet known and mill and crusher speeds up to 50 ft. per minute are employed, without causing trouble from juice drainage or choking. Crushers are run at a high speed, because the volume of cane is large, and thus a thinner blanket is obtained with better crushing performance. As very many plant cells have to be ruptured in the early stages of grinding, the first mills are run at a slow speed, whereas the speed of the last mills is increased in some countries.

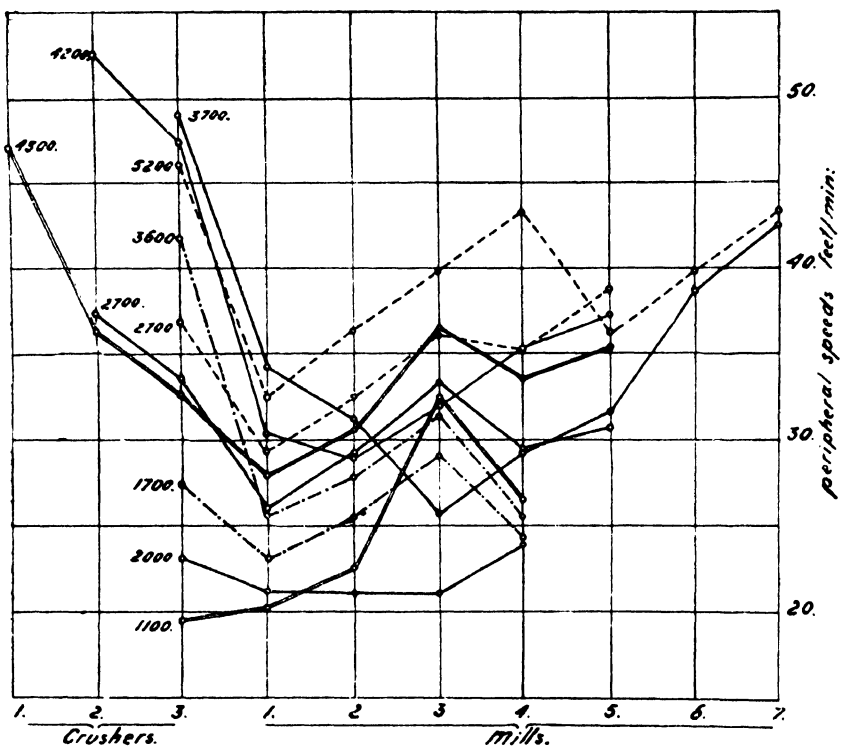


Fig. 171—Peripheral Mill Speeds.

There is nevertheless a divergence of opinion amongst operating engineers in this respect, as already mentioned, and in *Fig.* 171 the author gives from his own records a *Peripheral Mill Speed Diagram* from 10 Cuban mills having good operating performance. The numbers at the crusher end of the diagram show the capacities ground in tons per 24 hours (when not otherwise mentioned, long tons of 2240 lbs. are meant). As will be seen, there are big differences extant. In cases of doubt as to how far the peripheral speeds can be raised, tests should be made at different speeds and results noted, this being of special interest when different varieties of cane are ground. The driving engines or motors in this case should, of course, be equipped for varying speeds. The author has found that a peripheral speed in feet per minute of about 1·5 times the roll diameter in inches is allowable (i.e., a 30-in. roller would be operated at a maximum speed of 45 ft./min.). All the same, one should not attempt to run the mills at this high speed, when proper grooving is not applied, as in that case the mill will certainly choke.

The mill settings generally are not calculated, but adopted by comparison with some existing milling plant. The author has collected considerable data covering this aspect which reveal wide differences, and there is no doubt that the hydraulic rams on the top rollers correct many an imperfect mill setting.

A basic calculation is, however, possible. From tests he made, NOËL DEERR found that bagasse compressed at 2000 lbs. per sq. in. pressure weighs about 79 lbs. per cub. ft. and even higher compression will not cause any appreciable increase in specific weight, as more juice is extracted. The minimum mill setting, therefore, has to be based on this figure, as the author has done many times in practice with good results. From a dozen mills of very good grinding performance in Cuba, the settings of the empty mills gave a specific bagasse weight from 72 to 1300 lbs. per cubic foot, thus far in excess of the safety limit of 79 lbs. The setting "iron to iron" in these cases has been taken as $\frac{1}{16}$ in. so as to produce a volume.

The produced volume is the product of the *area of the mill opening* multiplied by the peripheral speed of the rollers. The area is the product of the mill opening and the roller length for smooth rollers. For grooved rollers, according to *Fig.* 172, it is obvious that for equal grooving on both co-acting rollers (top part of the Fig.) the area will amount to :—

$$A = d \times L \quad \ldots\ldots \quad (67a)$$

where d is the *vertical* mill opening. The effective roller length is increased through the grooving as explained in formula (56) in Chapter V and it should be noted that not the pitch, but the top angle β of the grooving is the deciding factor. The larger pitch causes a better crushing and has a better juice drainage.

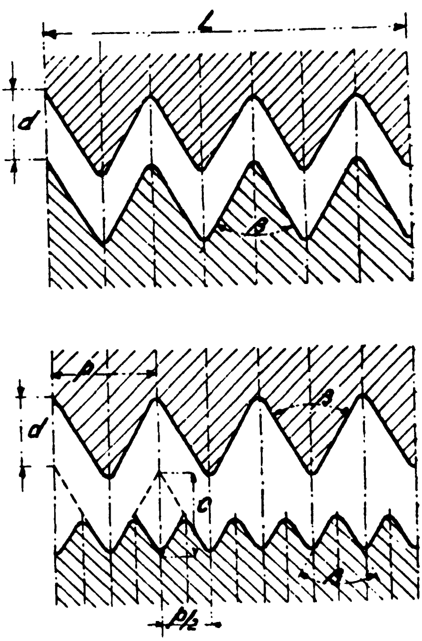


Fig. 172.—Area of Mill Openings.

When one roller has twice the number of grooves as the other (as shown in the lower part of *Fig.* 172) thus the top roller has a pitch p, whereas the lower roller has a pitch $p/2$, then :—

$$A = d \times L + \left(\frac{L}{2} \times \frac{C}{2}\right) \quad \text{or :}$$

$$A = L\left(d + \frac{C}{4}\right) \quad \ldots\ldots\ldots\ldots \quad (67b)$$

For a 45° grooving this will be : $A = L\,(d + 0{\cdot}30p)$, whereas for 60° it is : $A = L\,(d + 0{\cdot}217p)$; and it will be seen that in these cases the pitch also is considered.

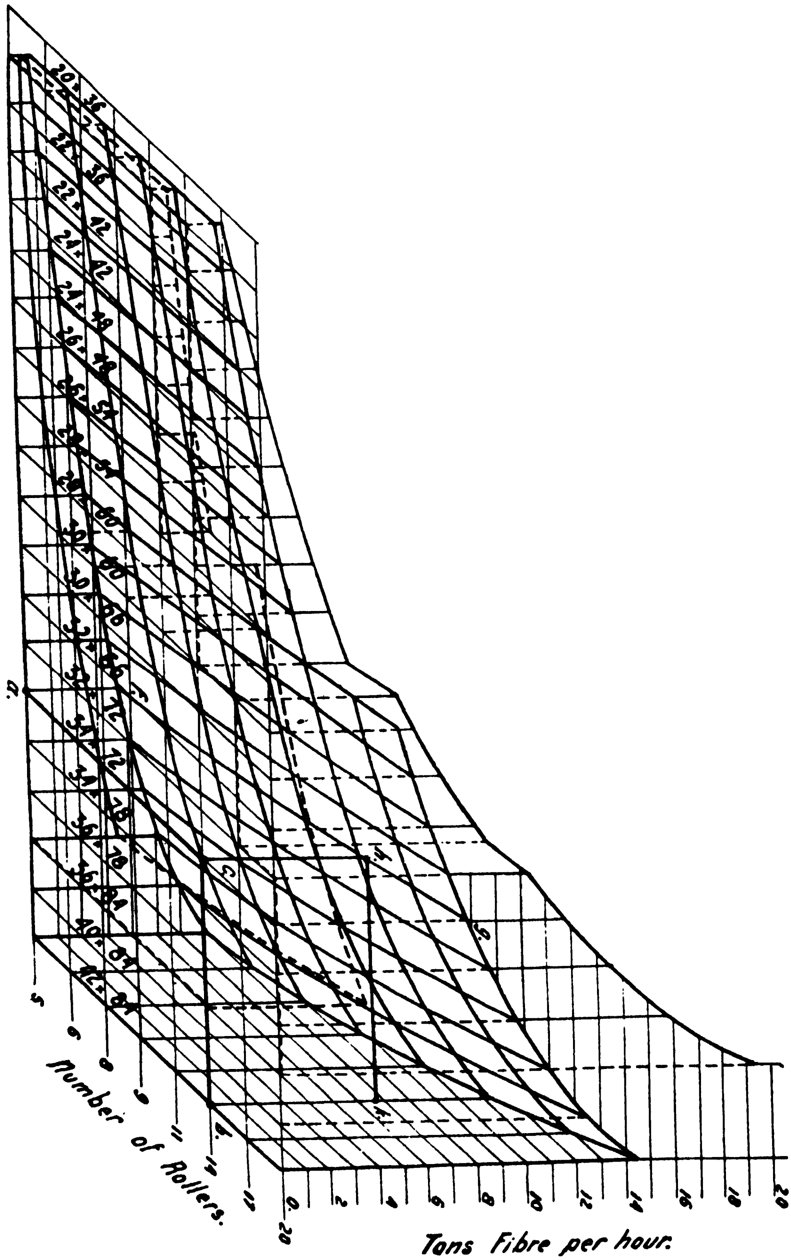


Fig. 173.—Mill Capacity Diagram.

When one roller has thrice the amount of grooves of the other, then the equation will read :—

$$A = L\left(d + \frac{C}{3}\right) \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (67c)$$

It should not be overlooked that with, e.g., 50 per cent. fibre in the bagasse, the weight of bagasse is twice that weight of fibre.

A fineness coefficient also has to be taken into consideration, as otherwise the crushers and first mills will not take the feed ; this coefficient has been reduced by the author in an arbitrary manner from practical observation to the following values, being a multiple of the volume of last mill bagasse.

	Arbitrary Fineness Coefficients.
Uncrushed cane to the first crusher	4
Knifed cane to the first crusher	3—2
Second crusher	3
First mill	2
Shredded cane	2—1
Second mill	1·5
Third and following mills	1

The *grinding capacity* of a given milling train is a very flexible figure, as there are installations known which have ground nearly twice the amount of cane for which they were originally designed. The mills of course have to be of substantial construction in such a case.

The capacity is equal to the actual volume produced by the revolving rollers and the opening between them, and the author has used the following formula for maximum capacities :—

$$C = \frac{V \times L \times d \times 60 \times 24 \times S \times N \times 100 \times K \times M}{2240^* \times N_n \times B} \quad \ldots . (68)$$

where : C = milling capacity per 24 hours in tons.

V = peripheral roller speed in ft. per min., taken as 1·5 times the roller diameter in in. for max. values.

L = roller length in in. = $(L \div 12)$ ft.

d = vertical mill opening in in., assuming $\frac{1}{4}$ in. in operating order, = $\frac{1}{48}$ ft.

S = maximum obtainable specific weight of compressed bagasse = 79 lbs. per cub. ft. (1·275 kg./dm.^{3}).

N = number of compressions, each crusher having one, each mill having two.

K = coefficient of roller diameter = $D \div 30$.

M = Coefficient of shredder or revolving knives = 1·25 to 1·1.

N_n = Normal number of compressions = 9 (as a standard milling plant is taken to be one crusher and four mills).

B = Percentage of bagasse on cane.

*When capacities are desired in other denominations, the divisor 2240 should be changed as follows :—

For metric tons to	2208
„ quintals (Java) to	220·8
„ piculs to	135
„ arrobas (Cuba) to	25
„ short tons to	2000

With these figures, the formula for maximum capacities reads :—

$$C = \frac{1{\cdot}5D \times L \times 1 \times 1440 \times 79 \times N \times 100 \times D \times M}{12 \times 48 \times 2240 \times 9 \times B \times 30}$$

$$C = \frac{D^2 \times L \times N \times M}{B} \times 0{\cdot}05 \quad \ldots\ldots\ldots\ldots \quad (68a)$$

With large milling trains, the rise in capacity due to a shredder or revolving knives is not as pronounced as with a smaller unit.

In *Fig.* 173 is shown a *Mill Capacity Diagram* from data supplied by the manufacturers for the following sizes of mills :—

20 in. and 22 in. × 36 in.
22 in. ,, 24 in. × 42 in.
24 in. ,, 26 in. × 48 in.
26 in. ,, 28 in. × 54 in.
28 in. ,, 30 in. × 60 in.
30 in. ,, 32 in. × 66 in.
32 in. ,, 34 in. × 72 in.
34 in. ,, 36 in. × 78 in.
36 in., 40 in. and 42 in. × 84 in.

and from 5 to 20 rollers per milling train. The grinding capacities are given in tons fibre ground per hour and the data are to be considered as average ones. Assuming, e.g., a milling train having 14 rollers 34 in. × 72 in., the line *a* is taken to represent the size 34 in. × 72 in., the line *b* stands for 14 rollers, these giving the intersection at *c*. The vertical at this point will reach the curve *f-g* at *h* and by transferring *c-h* = *b-k* about 7 tons fibre per hour is the average capacity to be ground, being 1,400 tons of cane per 24 hours with 12 per cent. fibre in cane.

2.—Mill Types.

The type of a mill is generally defined by the design of the mill housings and the arrangement of the trash turner. Originally the cane was ground between two vertical rollers, operated by hand, animal or water power and the rollers were made of hard wood or stone. Mills mechanically driven by steam power did not follow this construction, but used horizontally arranged rollers and in mill units of three, to obtain two compressions. The feeding of the horizontal mill is easier than that of a vertical one, but the three-roller mill has the inconvenience that the crushed cane after having passed the front roller has to be guided over a trash turner plate from this roller to the bagasse roller, and as the bagasse has to be delivered in a compressed state to the bagasse or rear opening of the mill, a considerable friction and consequent power outlay will result.

In *Fig.* 174 is shown a *Cast Steel Housing* for an existing 36 in. × 84 in. mill, as supplied through the author. The housings are made from cast steel, which will stand ultimate tensile stresses from 49,000 to 98,000 lbs. per sq. in., whereas cast iron has only one-third of the tensile strength and about half the ultimate shear, as compared with cast steel. Semi-steel is also used, being cast iron with a small percentage of steel, having about 30,000 lbs. ultimate strength, but this material is not ductile and may develop a sudden rupture under stress. Cast steel castings have to be properly annealed or slowly cooled down, to avoid internal stresses.

It is sometimes assumed that the material of the housings is of secondary importance, because the mill stresses are taken up by the king and side bolts, but it may be useful to mention that these bolts in most instances are unfortunately not arranged in the direction of these mill forces and the housings therefore are particularly subject to bending stresses. Moreover, the king and side bolts are not always tightened as firmly as should be the case, and in other instances the elastic limit of the bolt material is reached and it is therefore advisable to use the strongest material, wherever possible.

The design of *Fig.* 174 is more substantial than the older construction which it had to replace. As the existing king bolts had to be retained, the distance between the lower rollers could not be made as short as is usual in modern design.

The trash bar is supported on footings *b* cast integrally on the inside of the housings. This trash bar is of the rocker type, with easy adjustment in horizontal direction at the lower end. The rocker adjustment is achieved by means of a slotted lever, acting on the trash bar pivot. This slotted lever is

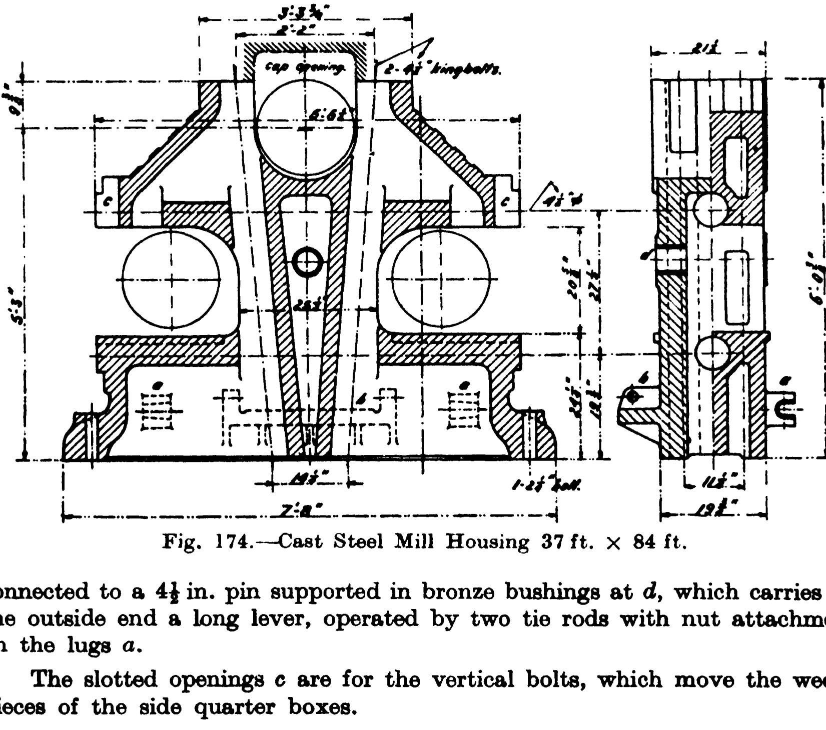


Fig. 174.—Cast Steel Mill Housing 37 ft. × 84 ft.

connected to a 4½ in. pin supported in bronze bushings at *d*, which carries on the outside end a long lever, operated by two tie rods with nut attachment on the lugs *a*.

The slotted openings *c* are for the vertical bolts, which move the wedge pieces of the side quarter boxes.

Owing to the fact that the existing king bolts are short, an opening is made in the top caps as indicated by dotted lines, so that a larger roller diameter can be used. There are four king bolts 4½ in. in diameter and two side bolts 4½ in. in diameter for each housing.

In *Fig.* 175 is shown an existing *Cast Iron Housing* for a 32 in. × 72 in. mill as measured by the author, and having also four king bolts and two side bolts. The king bolts have been flattened in the middle, to allow a smaller top angle between the centre lines of cane and bagasse rollers.

The trash bar adjustment is of interesting design owing to two wedges *a*, which are operated from the top cap side and a split bronze pivot link block *b*. This arrangement is convenient for grooved rollers, where small adjustments of the trash bar have to be made during grinding. The wedge pieces bear against lugs integrally cast with the housing. The pivot bearing on the lower end of the trash bar rests on the mill bedplate.

When choosing between cast iron or cast steel for mill housings, there is a further point of interest that cast steel housings will not corrode so rapidly from the action of the juice. It has, nevertheless, happened that cast steel housings have sand pitted and where this is the case, there will be heavy corrosion. Cast steel should have a flush surface.

The top angle of the triangle formed by the rollers varies between about 72° and 84°, and the smaller this angle the shorter the trash plate, resulting in reduced trash plate friction.

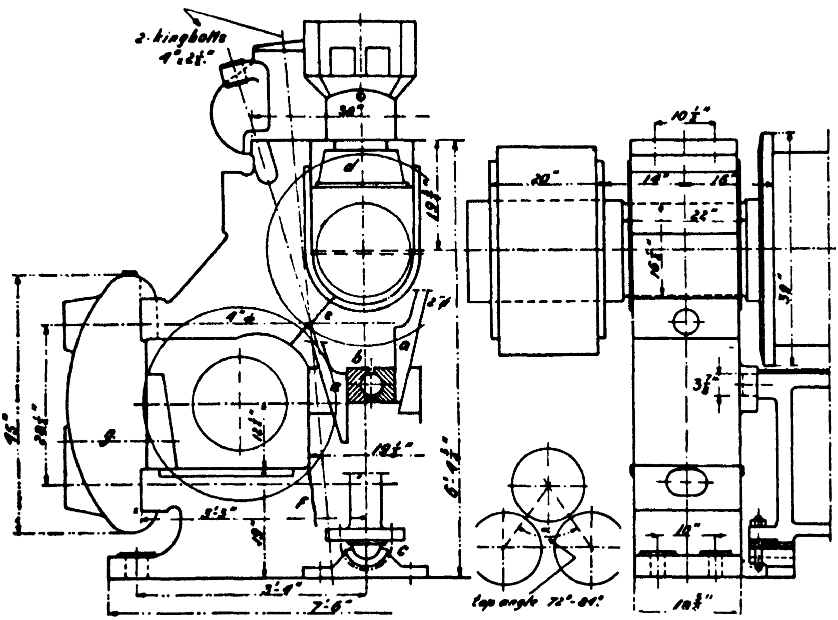


Fig. 175.—Cast Iron Mill Housing 34 ft. × 72 ft.

The design of *Fig.* 175 has a safety device taken from steel mills, in the form of a break pan *d*, made of cast iron of hollow construction. This pan or tub will break, when the pressure on the top roller rises beyond the safety limit of the entire mill construction, as may happen when pieces of tramp iron pass through the mill, and major damage will thus be avoided. A few spare tubs should be kept at hand for the purpose. Such a construction, nevertheless, is not generally adopted, as pieces of tramp iron should be separated at the start of the grinding process, as they will do much harm to the rollers, especially when these are grooved.

The side bearings (quarter boxes) are adjusted by wedges and as the adjustment travel is limited, packing plates have to be used between wedge and bearing for further adjustment or for worn rollers. This has induced manufacturers to provide a heavy bolt, which carries the total thrust of the bearing and has a much larger travel than the wedge piece.

The bolts *g* are attached to the bearing on both sides, to pull it back when the wedge is lowered.

A side view is shown with the arrangement of the trash bar and the rollers. The heavy trash bar pivot should be noted, as these pivots are prone to break when not sufficiently strong.

The mill housings are liable to fracture at the spot indicated by *e* at the feed side (front side) of the mill. The rupture indicated at *f* generally takes place on the discharge side. Tramp iron, loose bolts or caps which do not fit tightly, are amongst the frequent causes. Repairs during crop time are made by means of heavy steel plates, firmly bolted to the ruptured housing on both sides.

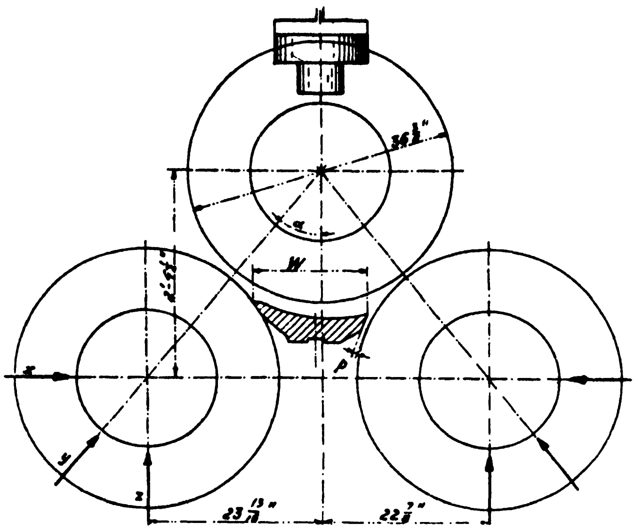


Fig. 176.—Mill Opening Adjustments.

In *Fig.* 176 are shown the three ways of adjusting *Mill Openings*. As is the case in most of our present-day mill designs, the hydraulic pressure rams act on the top roller, and the top roller only rises when under load. Adjustment of the mill openings therefore is done through the lower rollers.

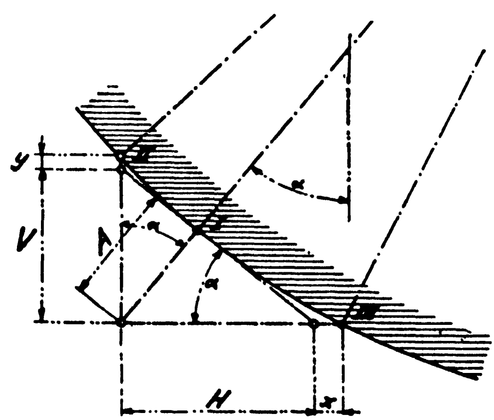


Fig. 177.—Adjusting Travels.

The horizontal adjustment, as indicated by the arrow x, is the one most widely used, this being accomplished by vertical wedges or thrust bolts in the side caps.

From *Fig.* 177 it will be seen that the *travel* for horizontal adjustment is to be derived from :—

$$H = \frac{A}{\sin \alpha}$$

where α is half the top angle.

With $\alpha = 37°$, $\sin \alpha = 0{\cdot}6$ and hence $H = 1{\cdot}67\,A$.

The angle α changes with the adjustment, but this change is of very little importance and consequently the distance x can be neglected.

It will be obvious that the horizontal adjustment is limited by the width W of the trash plate, and although the thrust bolt generally is designed for a long adjustment, the trash plate will not allow this and it has frequently happened that the juice passage p (*Fig.* 176) has been nearly or completely closed. The bagasse roller will indicate this phenomenon by so-called "spitting."

The second method of adjustment is indicated by the arrow z in *Fig.* 176. This design is used by several manufacturers and it reduces the travel to the amount (see *Fig.* 177) :—

$$V = \frac{A}{\cos \alpha}$$

and with the same angle $\alpha = 37°$, $\cos \alpha = 0{\cdot}798$ and $V = 1{\cdot}25H$.

The vertical travel is thus about 75 per cent. of the horizontal one. The distance y of *Fig.* 177, due to changing angularity, can be neglected as in the previous case.

It is apparent that when the trash bar is lifted correspondingly, there will be no effect on the trash plate width W and this is an advantage of vertical adjustment over the horizontal one ; but it should be kept in mind that horizontal adjustment is more easily achieved, as the roller weight does not bear on the wedge pieces.

The third method is to adjust in the direction of the centre line of the rollers, as indicated by the arrow y in *Fig.* 176. This adjustment affords a travel equal only to A (*Fig.* 177) being the shortest one of the three. The distance between the lower rollers, nevertheless, gets closer and the width W of the trash bar will put a limit to this type of adjustment.

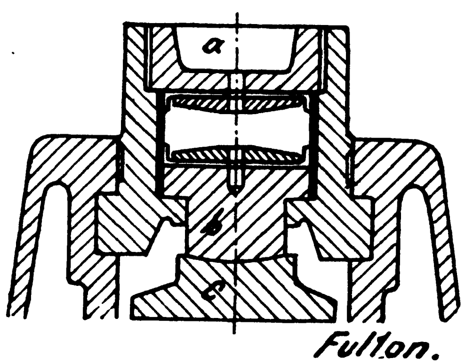


Fig. 178.—Boltless Mill Cap.

The time wasted in the task of removing the king bolt nuts, etc., when a top roller has to be removed or replaced, has induced designers to develop a *Boltless Mill Cap*, as shown in *Fig.* 178. To loosen the cap, it is turned about 90° and can then be easily removed. The hydraulic ram can be made of large diameter, as there is no space to be saved for the king bolts. The ram cylinder is closed by a cap a having interrupted threads, the so-called gun-closing device, with a leather cup attached underneath. The ram b has a leather cup on top, and for good design this ram should have long guides in a vertical direction as short ones might allow some tipping, which would cause the ram to stick. The ram cylinder is provided with a bronze bushing to reduce friction. The inlet of the pressure oil is in the sides of the caps, so there are no hydraulic pipelines to be disconnected, when a leather cup has to be replaced.

The mill to which this boltless cap belongs has, nevertheless, side bolts, and *Boltless Mill Housings* are now constructed as shown in *Fig.* 179. The top cap embraces the mill housing and this detail will give additional strength against any lateral thrust of the top bearing, which tends to open the top bearing gap. The light bolts a are only for securing the cap in its working position. For removal, the caps are turned 90°, similarly as with those of *Fig.* 178.

It should be mentioned that there are also boltless constructions, where the caps have to slide sideways for removal.

The side caps are held by the thrust bars *c* and short bolts *b* having imbedded T-heads in the housing. Since the reaction of the lower roller thrust is below the horizontal centre of the cap, the bolts *b* will carry only a reduced part of the load.

The boltless housings have to carry all the working stresses of the mill, so they necessarily have to be of substantial design, and cast steel should be used throughout.

The lower rollers are adjusted in the direction of the centre line of the rollers by wedge pieces *e* and thrust bolts *d*. Before removing the side caps, the lower rollers have to be supported on wooden beams or jacks, as they otherwise will be pushed outwards.

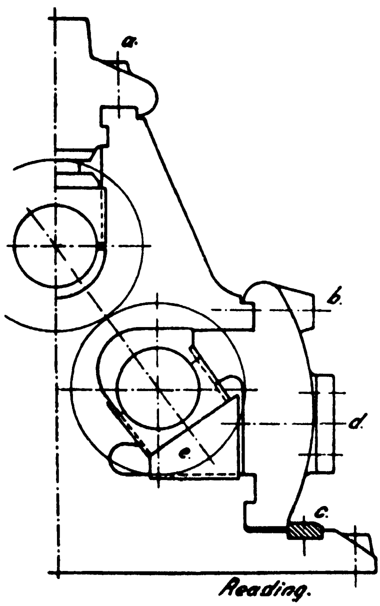


Fig. 179.—Boltless Mill Housing.

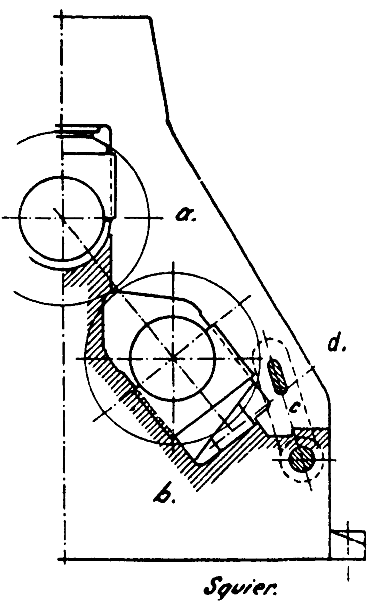


Fig. 180.—Triangular Mill Housing.

In *Fig.* 180 is shown a *Triangular Housing* of recent design. The special feature of this housing is that there are no caps whatsoever, as it is composed of the top part *a* and the bottom *b*. The division line has been shaded in the drawing and the connection of both parts is effected by strong links *c*, which can be swung outwards on heavy pivots, after removal of the tightening wedges. After the top part has been lifted, all three rollers are freely accessible.

The hydraulic rams are integrally built into the upper part, whereas the adjustment of the lower rollers takes place through double wedge pieces, moved by a worm-operated rack at *d*.

In contrast to the construction of *Fig.* 179, the rollers are well supported in their bearings on removal of the top part.

In Java trials have been made with mill housings arranged for *floating top rollers*, a point of special interest, as the top roller will be in such a position that there is no side thrust on the top bearing and thus the pressure between top and feed rollers has to be equal to the pressure between top and discharge rollers. There is one drawback to the feeding of such a mill, as the cane roller takes the feed less readily than does the discharge roller and in our present-day mill settings an allowance has to be made in this direction. A more recent construction, therefore, arranges the hydraulic ram in such a way that it can be placed at any desired inclination, thus providing a heavier pressure between the top and discharge rollers than between top and feed rollers. This inclination of the ram is towards the feed side of the mill and, moreover, the top roller with the hydraulic ram attached can swing on a centre lying below the bedplate of the mill, so the established proportion between front and discharge roller pressures will be maintained, whatever the feed may be. Favourable reports have been issued about the milling performance of this design.[1]

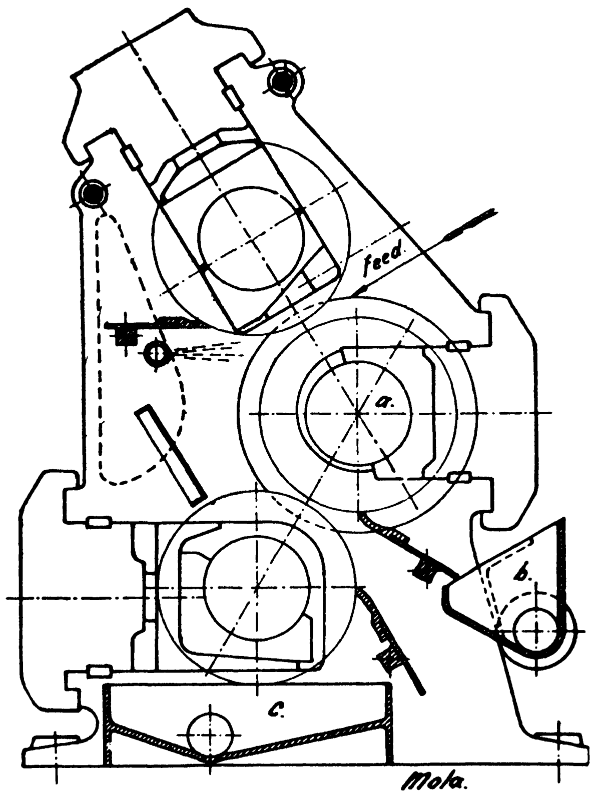


Fig. 181.—Mill without Trash Bar.

This De Bruyn system of floating top rollers can also be applied to existing mills. The top roller bearing must have lateral play in the mill housing gap, and a horizontally arranged hydraulic ram, connected to the top roll bearing flanges, will keep the side thrust within the pre-established limits.

It should be mentioned that that noted designer of sugar mills, the late J. V. HAMILTON, has designed mills with two sets of hydraulic rams under the lower rollers and arranged in the direction of the roller pressures. The advantages of the floating top roller are also obtained by this construction. The

1 See F. MAXWELL, "Modern Milling," pages 140 and 194.

inconvenience is the location of the hydraulic rams and, in addition, the trash bar has to be connected to the front roller bearings to allow it to move up and down with this roller.

A novel construction, recently patented and for which the author made a preliminary design, is shown in *Fig.* 181, being a *Cane Mill without Trash Bar.* The three rollers are placed in a slanting position, each on top of the other, the angle of inclination being about 30° with the vertical. The middle roller *a* is the driven one and has a fixed position, which will allow an easy drive, as this roller does not move like the top rollers in the widely-used Rousselot housings, as shown in *Figs.* 174 and 175.

The mill housings need to be higher, but this does not result in any practical inconvenience, as high level crushers are now in use everywhere.

The lack of a trash plate will reduce the friction and thus the power consumption of the mill, and, what is more, imbibition can be applied between the first and second pressings.

The juice of both these pressings is collected in separate juice trays *b* and *c*, so that each can be kept separate for imbibition purposes. The top and the lower rollers have separate hydraulic rams, which can be made with different diameters or be equipped with separate hydraulic accumulators.

The advantages of the floating top roller are obtained and although the construction has not yet been put into practical operation, it is of sufficient interest to be mentioned here.

In the design of mill housings and the other mill details, the acting forces which cause the working stresses have to be considered at the outset and in *Fig.* 182 a *Mill Forces Diagram* is drawn.

As already mentioned, the pressure between feed and top roller has to be smaller than between top and discharge roller, as otherwise the feeding of the mill will be impaired. As a rule the mill settings are arranged in such a way that the front opening under operating conditions is about twice as large as the back one, which will allow a good feed. The bagasse opening is the calculated one according to the calculation already mentioned in this chapter.

In *Fig.* 182 it is assumed that the total resultant force R is obtained by the front component $C_{\prime} = 180$ tons and the back component $C_{\prime\prime} = 360$ tons. This resultant R slants towards the feed side at an angle α and the construction of the Puunene housing, which originated in Hawaii, is based upon this fact, it having the hydraulic ram acting in a slanting direction of about 15° with the vertical, to avoid any lateral thrust on the mill housings. In the standard Rousselot housings, the side thrust $H_{\prime}$ amounts to about 100 tons in our case and the two housings have to withstand the bending effect of this force.

The feed roller component $C_{\prime}$ is subdivided in a horizontal force $H_{\prime\prime} = 100$ tons and a vertical force $V_{\prime\prime}$, whereas the back roller component $C_{\prime\prime}$ is subdivided in a horizontal force $H_{\prime\prime\prime} = 200$ tons and a vertical force $V_{\prime\prime\prime}$. The side bolts on both housings, therefore, have to withstand a maximum force of 200 tons, but it should not be overlooked that the pressures on both housings are not always equal and an allowance has to be made in this direction. Moreover, there might be a difference in load between the upper and the lower side bolts, as the reactions $H_{\prime\prime}$ and $H_{\prime\prime\prime}$ may act below the mid centre of the side cap and thus impose heavier stresses on the lower side bolts.

The mill is driven by a driving momentum of the value :—

$$M_t = P \times r.$$

As the power input is assumed to be 300 h.p., we easily derive :—

$$N = \frac{P \times V}{550}$$

where N is the input h.p., V the speed in ft./sec. of the force P in pounds and 1 h.p. = 550 ft. lbs./sec.

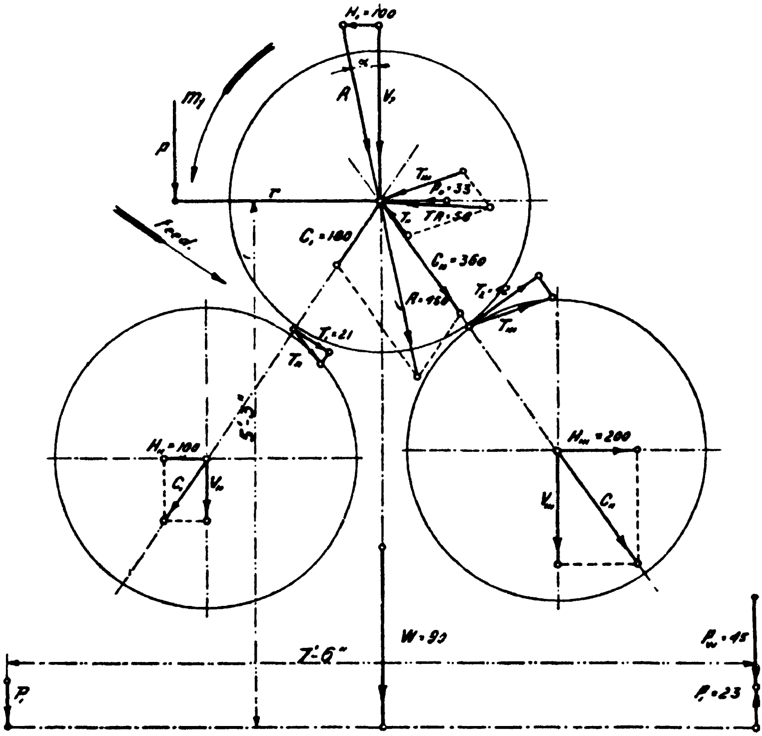


Fig. 182—Mill Forces Diagram.

The speed V obviously is :—

$$V = \frac{r \times 2\pi \times n}{60} = 0{\cdot}1046 \times r \times n \text{ ft./sec.}$$

where n is the number of revolutions per minute of the top roller ; it may be written :—

$$N = \frac{P \times r \times 0{\cdot}1046 \times n}{550} = \frac{P \times r \times n}{5258{\cdot}1} = \frac{M_t \times n}{5258{\cdot}1}$$

and it follows herefrom that :—

$$M_t = 5258{\cdot}1 \times \frac{N}{n} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(69) in ft. lbs.}$$

For the metric system the formula, derived in the same way, reads :

$$M_t = 71620 \times \frac{N}{\text{n}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(69a) in kg. cm.}$$

It being assumed $n = 4$ r.p.m. in our case, there will result :—

$$M_t = 5258{\cdot}1 \times 300 \div 4 = 394{,}357{\cdot}5 \text{ ft. lbs.}$$

The driving momentum M_t drives the three rollers and as in a mill all the mechanical work is transformed into frictional resistance, it may be assumed that the turning moments of the three rollers are in proportion to the roller pressures. In practice the trash plate friction is delivered by the top roller, thus the pressures on the lower rollers will be less. The proportion of the roller pressures is :—

top : front : back = 450 : 180 : 360 = 5 : 2 : 4.

The top roller therefore will consume approximately $\frac{5}{11}$ of the power input, the front roller $\frac{2}{11}$ and the back roller $\frac{4}{11}$, thus :—

M_t top roller	179,253 ft. lbs.
M_t front roller	71,701 ,,
M_t back roller	143,403 ,,
Total	394,357 ,,

With a crown wheel pitch diameter of 1·5 ft., the tooth forces are :—

T_1 front roller	47,800 lbs. ≃ 21 tons.
T_2 back roller	95,602 lbs. ≃ 42 ,,

As the tooth flanks are curved to the involute, the tooth pressure deviates 15 to 18° from the normal, thus increasing up to $T_{,,}$ and $T_{,,,}$.

The reactions of the tooth forces have an opposed direction at the top roller centre and the resultant tooth reaction TR of about 58 tons has a nearly horizontal direction, and it will be obvious that the crown wheels will not cause any appreciable separating force acting on the rollers.

The driving momentum gives rise to a tipping couple of forces $P_o \times 5{\cdot}25$, the force P_o amounting to :—

$$P_o = \frac{394{\cdot}357}{5{\cdot}25} = 75{,}100 \text{ lbs.} \simeq 33 \text{ tons.}$$

This tipping couple is counterbalanced by the couple of forces $P_{,} \times 7{\cdot}5$ of the bedplate bolts and therefore :—

$$P_{,} = \frac{394{\cdot}357}{7{\cdot}5} = 52{,}580 \text{ lbs.} \simeq 23 \text{ tons.}$$

The proper weight of the mill, assumed at 90 tons without the bedplate, causes a momentum : $W \times 3{\cdot}75$, and the reaction at the locus of the bedplate bolts therefore amounts to :—

$$P_w = \frac{90 \times 3{\cdot}75}{7{\cdot}5} = 45 \text{ tons,}$$

thus in excess of and opposed to the force $P_{,}$; and therefore the bedplate bolts are only for holding the housings in place. Keys between the footings of the headstocks and the bedplate ridges are for lateral tightening.

It should be borne in mind that the top roll thrust and the tooth force reactions do not cause a tipping momentum on the bedplate but have to be taken up by the mill housings proper.

3.—Mill Details.

The mill details so far as they were not dealt with in the previous chapter will be treated here consecutively.

The mill housings or headstocks are bolted to the bedplate, and in *Fig.* 183 is shown a *Combined Bedplate and Juice Tray* of European construction. The two box girders of heavy section, on which the mill housings are mounted, are interconnected by the integrally cast juice tray. The bottom of the tray slants towards the juice discharge openings and these are arranged bilaterally, so that the juice can be delivered on either side. The bagasse which falls on the juice tray will not be flushed away by the juice, as the slope is not sufficient for this purpose and the operator has to clean the tray at regular intervals. This applies expecially to mills with Messchaert grooving, where quite an amount of bagasse is scraped out of the grooves, to fall on the juice tray.

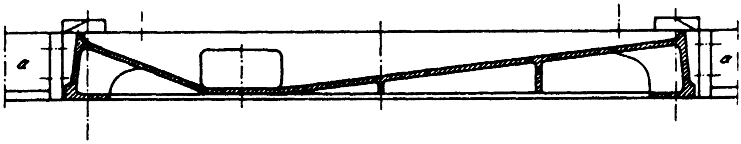


Fig. 183.—Combined Bedplate and Juice Tray.

In Java a perforated brass sheet is sometimes placed over the juice tray, to keep the bagasse out of the juice, but it cannot readily be cleaned underneath and fermentation of the juice sometimes results, hence these perforated plates are best omitted.

The bedplates of consecutive mills of a tandem are connected by cast iron box girders *a*, giving a firm connection to the whole milling train.

An American design of *Continuous Juice Pan* is shown in *Fig.* 184, and this type has also been adopted by several European manufacturers. Instead of the interrupted juice trays under each mill, a continuous one has been arranged with walls well sloped towards the longitudinal centre, so that the bagasse will more easily be flushed down. On the sides of the box girders perforated pipes for steam or hot water can be run for cleaning and disinfecting purposes.

The mill housings are placed on continuous box girders *c*, interconnected at regular intervals by cast I beams *b*, having inverted V-shaped top flanges. A protecting ridge *a* should be provided on the inside of the box girders, to protect the juice pan joint from entering juice. Sometimes the upper part of the juice pan is integrally cast with the box girders.

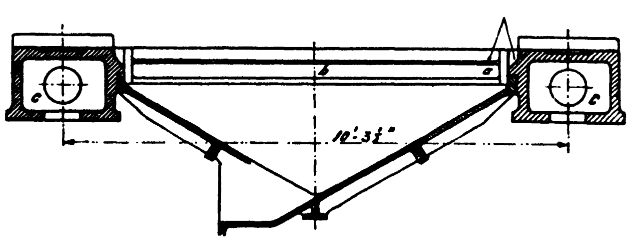


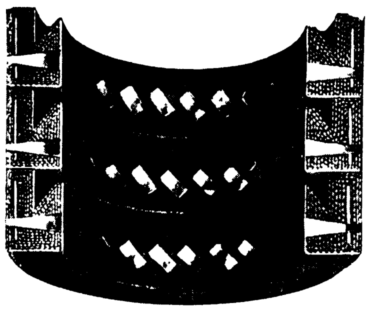
Fig. 184.—Continuous Juice Pan.

The discharge can be effected at any convenient place and the juice pan may be sloped lengthwise towards the discharge spout. Baffles are placed to separate the different mill juices.

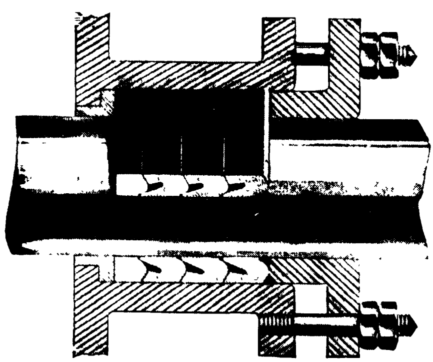
For heavy grinding these continuous juice pans are very convenient, as they will take care of large amounts of juice, and moreover all drippings from the intermediate carriers or falling pieces of bagasse are easily collected.

The bedplates and juice pans are made from cast iron.

Another important detail of a sugar mill is the *Hydraulic Ram or Pressure Regulator.* Several British manufacturers provide spring regulators, of which

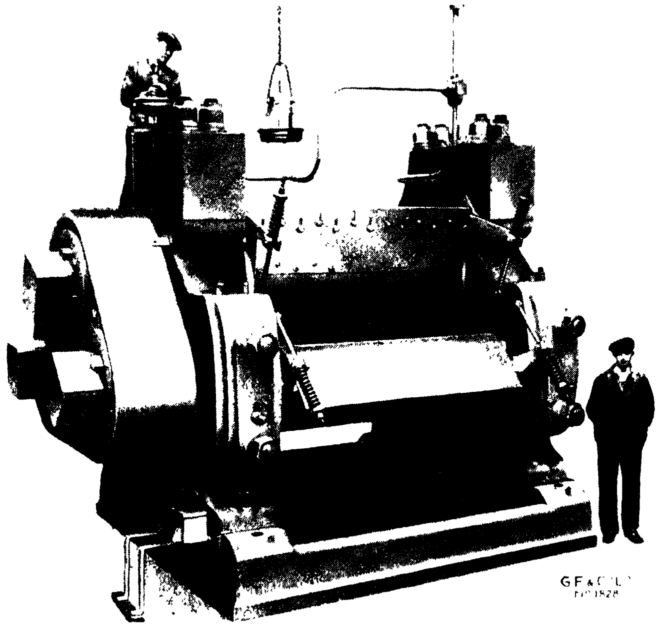


Hydraulic Packing for Mill Caps.
(*James Walker & Co., Ltd.*)

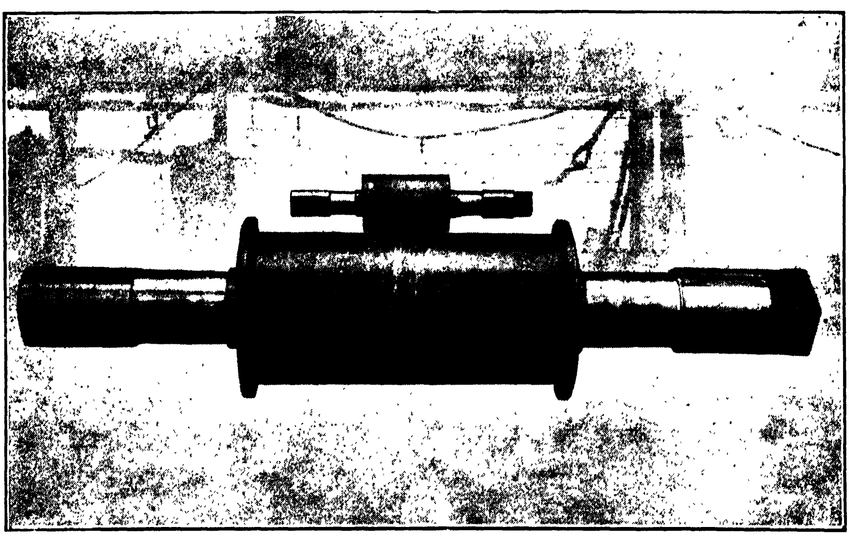


Rod Packing for Hydraulic Pumps.
(*James Walker & Co., Ltd.*)

PLATES 34 & 35.



35 in. x 78 in. CANE MILL WITH SCRAPERS.
(Geo. Fletcher & Co., Ltd.)



40 in. x 84 in. and 16 in. x 24 in. MILL ROLLERS.
(John McNeil & Co., Ltd.)

the "*Toggle*" *type* is well known and has found many applications. In more recent plants hydraulic regulators are predominant, and in countries like Cuba they are used exclusively. These pressure-regulating devices are generally applied on the top rollers, as it will be obvious that one regulator will then control the two mill openings, although there are a few inherent disadvantages, as has been explained above. Horizontal regulators on the lower rollers are now obsolete.

Unpacked hydraulic rams have been tried with labyrinth grooves, but have not proved a practical success under the prevailing working pressures of about 4000 lbs./sq. in. (300 atm.)

Many modern designers use leather cups or U-leathers, whereas some European and British firms employ hydraulic packing ("Lion" packing). In early designs the hydraulic packing got compressed by the hydraulic pressure, and this had to be abandoned; the author knows an instance where the packing was compressed to such an extent, that the rams would not operate in the cylinders and stuck. A modern design with *Adjustable Packing* not having this inconvenience is shown in *Fig.* 185 and favourable operating performance is reported with reduced friction and very little wear on the packing and cylinder walls.

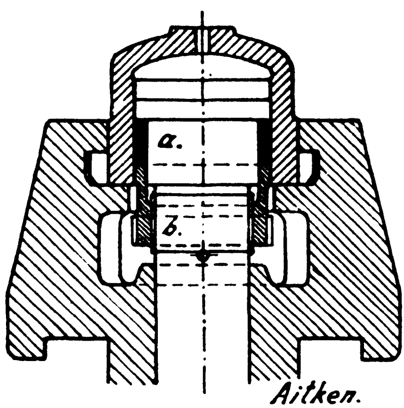


Fig. 185.—Ram with Adjustable Packing.

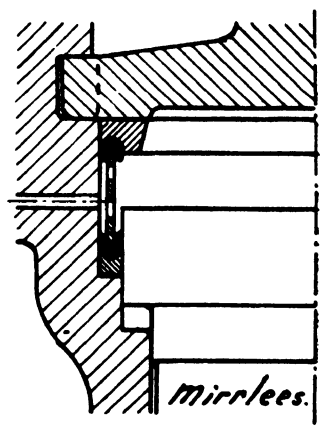


Fig. 186.—Hydraulic Packing.

The hydraulic piston has a space *a* for the packing rings, which can be tightened or compressed by the nut *b* threaded on the lower piston stem. The nut is tightened by a special spanner, which fits into the grooves of the nut periphery. The piston proper has a hole for inserting a pin while tightening the packing, so that it will not turn round with the nut.

The cylinder is of the cap type, having two supporting lugs, so it can be loosened by one-fourth of a turn. The hydraulic pipe connection is at the top part of the cap and for renewing the packing this pipe has to be disconnected. As renewal of the packing is not as frequent as with leather cups, the hydraulic pipe connection is of minor importance. The ram cylinder is made of cast steel and is not lined. The adjustment of the packing can be done while the mills are operating, a point of great advantage.

In *Fig.* 186 the design is shown of a hydraulic pressure regulator with *U-Leather Packings*, and the insertion of these is very easily accomplished. Between the two U-leathers a ring of perforated material with large holes and rounded corners is placed, which keeps the collars open and in place. A light cover is provided with lugs, and for removal only one-twelfth of a turn is necessary, the hydraulic pressure being released.

The hydraulic connection is placed at the side of the top cap casting and need not be removed when replacing the U-leathers. The protecting ring for the upper leather is fitted tightly or welded to the cylinder cover.

Another interesting *Hydraulic Ram Construction* is shown in *Fig.* 186a, this also having the hydraulic pipe connection at the side of the cap. At *a* threaded pins can be inserted, after the hand cover has been removed, so as to turn the breech block *b* one-eighth of a turn to release it. The U-leathers are held in forged steel bolster plates *c*, which give a good bearing to the crowns of the leathers, and bronze guard rings hold the leathers in place. Renewal

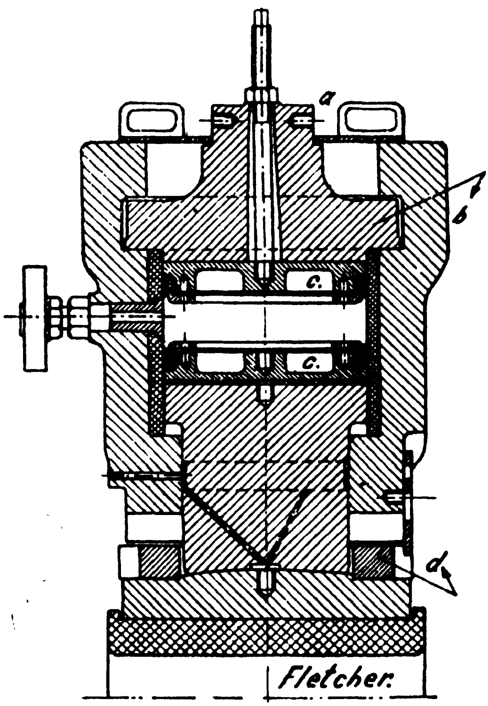


Fig. 186A.—Hydraulic Ram Construction.

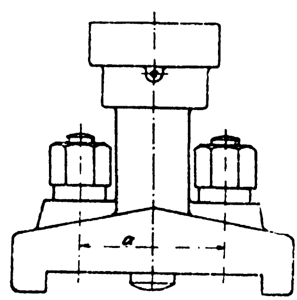


Fig. 187.—Narrow Type Top Cap.

of the U-leathers is feasible, without removing the ram proper and a set of bolster plates, complete with U-leathers and guard plates, can be kept as spares, so they can be inserted at once, as soon as the other set has been removed.

When it has to be replaced the top bolster plate is inserted loose from the breech block, so as not to damage the leather. In case the hydraulic gear is cut out, a steel locking ring *d* is rotated about one-eighth of a turn and fixes the top bearing.

In America leather cups are used as shown in *Fig.* 178, and a *Narrow Type Top Cap* as shown in *Fig.* 187 is provided with this packing. The design is for existing mills, not having hydraulic pressure regulators and where the king bolt distance *a* is short. Here it will be not irrelevant to mention that large hydraulic rams are to be preferred, as the specific pressure of the system will be less, and there will be less wear on the leather packing.

The long piston ensures a good guide within the cap, without the danger of sticking.

Leather packings have caused many a trouble, expecially when the cylinder linings are worn or grooved or the pistons have lost their shape and the leather thus becomes wedged between the piston and the lining. In *Fig.* 188 different ways of *protecting leather cups* are demonstrated. In design I, on top of the piston *p* is laid a disc *c* of plastic material like "vulcan-fibre" and on top of this the standard leather cup is attached. It is obvious that the space *a* is not filled by the cup, when this is just inserted and the hydraulic pressure will force the cup heel into this sharp corner, and may thus rupture the leather. The top of the piston or of the plastic disc may not have convexly rounded corners, but the latter should have concave fillets.

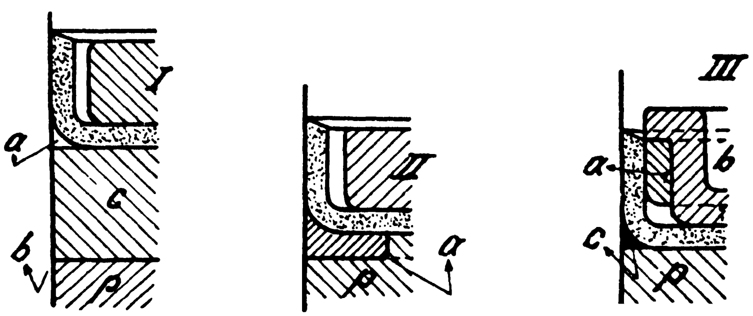


Fig. 188.—Leather Cup Protection.[1]

In design II a ring *a* is laid on top of the piston *p*, which has been turned down locally. Monel metal is here used to advantage, but the author has used a good quality of white metal, having a high ductility, with good results.

Design III has a monel metal fillet ring *c* on top of the piston *p* and a self-expanding ring *a* inside the leather cup, so that it will not curl under pressure. The cover plate *b* holds the leather cup and the self-expanding ring in place.

Leather cups have to be manufactured from first class chrome leather and the friction coefficients of leather cup packing are given as follows :—[2]

1.	Cup 1 in. high and a $1\frac{1}{8}$ in. fibre disc	0·0587
2.	Cup only 1 in. high	0·0488
3.	Cup 1 in. high and monel metal protective ring at heel	0·0300
4.	Cup $\frac{3}{4}$ in. high with protective ring	0·0278

The tests for ascertaining these friction coefficients were made by bolting two mill top caps together with the rams acting against each other and each cap connected to a hydraulic accumulator, the pressures being measured by hydraulic manometers. The top caps were provided with new linings having rams with 0·01 in. clearance.

For a mill having a 600 tons load on the top roller, the frictional resistance of the hydraulic pressure regulators will thus vary between 35 and 17 tons. The author knows an instance where only the disc of plastic material was used, as the leather cups had ruptured. There was no leakage and from the difference between 1 and 2 it may be assumed that a very low friction resistance was ruling.

[1] *I*: Construction of R. MAYO Jr. *III* : Construction of H. SCHARNBERG.
[2] See article by H. J. B. SCHARNBERG in *F.A.S.*, Oct. 1929, page 998.

King bolts for different mills, size 34 in. × 78 in., are shown in *Fig.* 189, these having been measured by the author for replacement, as the existing ones were broken. The wedges at the lower end of the king bolts are mounted in a special cast steel wedge saddle, bolted under the mill bedplate. Nuts are not used for slanting king bolts at this spot, as they occupy too much space.

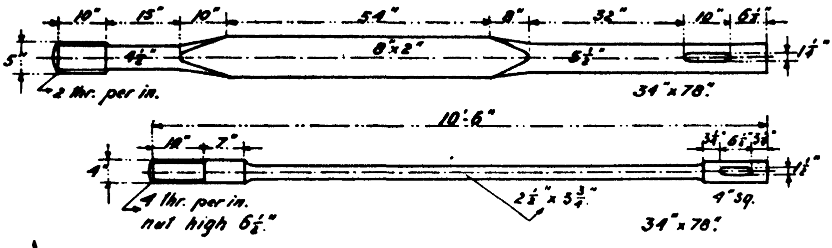


Fig. 189.—King Bolts.

Both bolts have a flattened section in the middle and the top one has the diameter below the thread turned down, to avoid the V notch stresses caused by the thread. The upper bolt is that of a mill of recent design, whereas the other is of an older mill and much shorter. The lower end of the 4 in. bolt is square, to give additional strength around the wedge slot.

Roll Flanges were formerly fitted to the lower rollers of the three-roller mill; as these rollers were the juice draining ones, it was logical to provide them with flanges. The horizontal centre distance of these lower rollers had to be equal to the flange diameter plus the trash bar width plus the necessary clearance, and a very wide trash plate resulted. This construction, therefore, has been abandoned and nowadays the top rollers have the flanges.

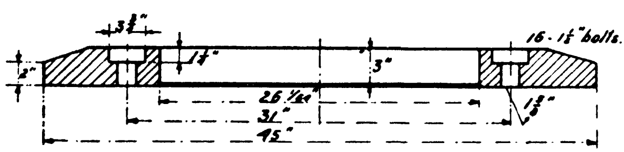


Fig. 190.—Roll Flange 36 in. × 84 in.

The flange of *Fig.* 190 is made of cast steel for a 36 in. × 84 in. mill. The diameter is large because the flange is used on a first mill, having large mill openings. For the subsequent mills smaller flange diameters are used, as the trash bar body will not allow larger ones. There is a considerable wear on these flanges on the inside, so the original thickness has to be as great as possible to make facing feasible. Cast steel flanges seldom break, and then only when they are too thin.

The flange slants to the outer periphery, but this is not a necessity. Sometimes a groove is provided on the outside diameter for juice drainage. The bolt holes are cottered to accommodate the bolt heads. (See also *Fig.* 164).

Flanges are sometimes cast integrally with the roller, but any breakage of such a flange will result in a heavy job to turn the material down and make a seating for a loose flange.

On the crown wheel side split flanges are used, so that they may be attached or removed without removing the crown wheels.

Roll diameters of existing mills are sometimes increased for better grinding or heavier capacities, and it has happened that the crusher or mill housings would not allow of this increase in diameter and *Top Cap Fillers*, or distance pieces, as shown in *Fig.* 191, had to be inserted between the housings and the top caps. The fillers shown were supplied for crushers, but the system can be applied to mills as well with the eventual use of new king bolts.

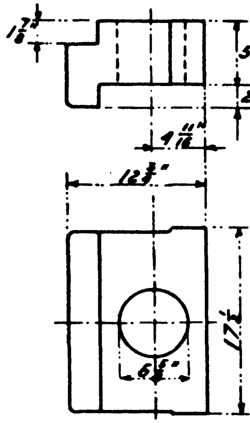


Fig. 191.—Top Cap Filler.

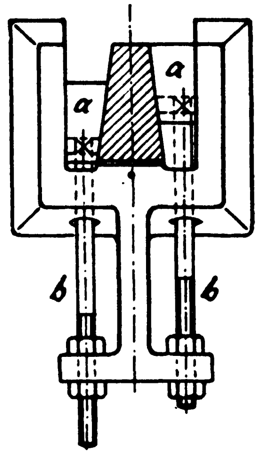


Fig. 192.—Trash Bar Adjustment.

The *Trash Bar* belongs also to the important features of a cane sugar mill, as it is subject to heavy stresses as well as to heavy corrosion. The author has supplied many a trash bar of improved construction for existing mills, and when breakages occur, it should be considered whether the construction cannot be improved upon, before ordering the new material.

Trash Bar Adjustment is arranged in different ways, and *Figs.* 174 and 175 show the usual constructions adopted in America. In Java, where mill grooving is not practised as in other countries, the adjustment as shown in *Fig.* 192 has given very good operating performance. The ends of the trash bar have inclined surfaces on which two wedge pieces *a* co-act. The wedges are operated by brass bolts *b*, having T-heads. The trash bar and the wedge pieces are firmly held in a housing of ample dimensions and integrally cast with the cast steel mill headstock.

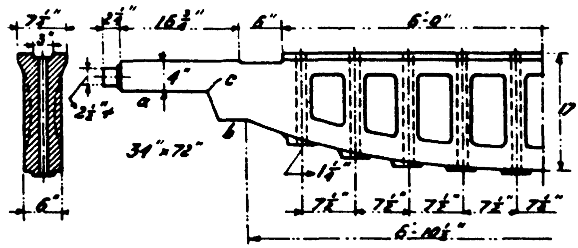


Fig. 193.—Sliding Trash Bar.

The mill headstock and trash bar will thus form a solid unit, but adjustment can only be effected from under the mill rollers, a not very accessible place during grinding. For this reason, with radially grooved rollers, where trash bar adjustments are frequent, this construction has not been used to any great extent.

In *Fig.* 193 is shown a *Sliding Type Trash Bar* for a 34 in. × 72 in. mill, which was supplied to some existing mills. The mill housings have openings for accommodating this trash bar and it is adjusted by levers on the outside of the mill housings, which engage on the 2½ in. diameter trunnions. The bar is supported at a and b, which should be adjusted properly, as otherwise the bar will break at c when under load. The trash plate is connected by through-going bolts and firmly fitted in a groove at the top of the bar. With this construction the trash plate can be removed without taking out the lower rollers.

The *Rocking Type of Trash Bar* (*Fig.* 194) is a construction supplied by the author for existing 36 in. × 84 in. mills. The trunnions for the adjustment, like that shown in *Fig.* 175, are 4½ in. in diameter. The trash plate is of the side flange type and horizontal bolts connect it to the trash bar.

A long rocking radius of the trash bar is to be preferred, as the back opening between trash plate and top roller will then not be narrowed too much by adjustment. Moreover, the lower end of the trash bar can be arranged to slide along the bedplate to restore the vertical position of the bar.

These trash bars are made of cast steel, as they are subject to heavy stresses. Mild steel has been used in the past, and this practice might come into vogue again for welded trash bar constructions of heavy plate section; this would reduce the distance between the lower rollers.

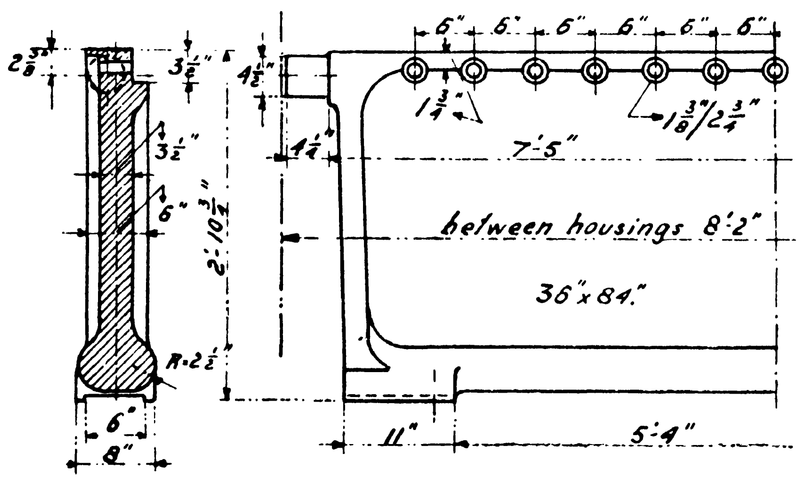


Fig. 194.—Rocking Trash Bar.

The dimensions of a trash bar are generally taken from previous constructions which have been fully tested and proved satisfactory, but it should be of sufficient strength to withstand the hydraulic load plus the top roller weight uniformly distributed. In case of tramp iron going through the mill, it can easily happen that this load is applied at one point only and a breakage might then occur.

The *Setting of the Trash Bar Plate* or knife is accomplished in many ways. BERGMANNS of the Java Experimental station found in 1889 by mathematical computation that the plate curve had to be a logarithmic spiral, but one function of this spiral construction had to be estimated arbitrarily.

It is obvious that the purpose of the trash plate is to conduct the bagasse from the front to the rear mill opening in a continuous flow with as little friction as possible. In *Fig.* 195 the front opening a of the trash plate is taken to be about twice the front mill opening and $b > a$ and $c > b$. The sloping towards c is taken as about 4 to 5 per cent. of the plate width, or in general terms $b = a + \frac{1}{8}$ in. and $c = b + \frac{1}{8}$ in. for 7 ft. mills.

The starting point of the trash plate curve should, according to MULLER VON CZERNIKY (Java), be located at an angle $\beta = \frac{1}{8}\ \alpha$ which will amount to about 13°. In practice in several Cuban mills a is measured on the line no, where n is located at one-third of the roller radius from the outside periphery. The distances ab and bc are then divided perpendicularly into halves by the lines xx and yy, the intersection of these lines being the centre point of the trash plate radius R.

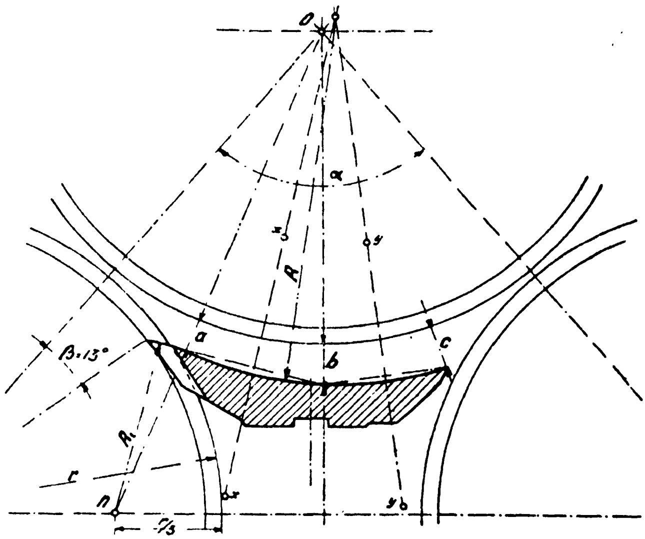


Fig. 195.—Setting of Trash Bar Plate.

The passage for juice between plate and back roller should be from ½ in. to ¾ in., depending upon the roller adjustment. This passage increases when the trash plate lip wears down. In case the bagasse roller has heavy grooving, the plate end at c should have corresponding grooves, so that the bagasse may not fall through at this spot.

In general too low a trash plate setting will result in the bagasse not being delivered in a continuous flow and the mill will slip, which fact is disclosed by the "grumbling" sounds coming from it.

Too high a trash plate setting will impair good milling as the hydraulic load is taken up by the trash knife and not by the rollers, thus increasing the power consumption. Moreover, the plate is subject to heavy wear.

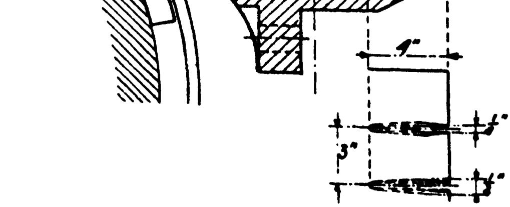


Fig. 196.—Flange Type Trash Plate.

Fig. 196 shows a *Flange Type Trash Plate*, which gives a firm connection between plate and bar. On the rear end at b are grooves cast in the plate of a self-draining type with a pitch of 3 in., this being a modification of FISHER'S patent. These grooves serve the purpose of draining the juice that cannot escape readily through the juice passage. There is also a patented construction providing juice drainage grooves cut longitudinally in the top roller.

The author knows an instance where the width of the trash plate became too small through wear, etc., and the back lash increased beyond practical limits. The factory engineer for this reason bolted a flat iron with countersunk bolts to the trash plate, but unfortunately this flat iron was ripped off by the bagasse, causing great damage to the groovings, when passing between the rollers.

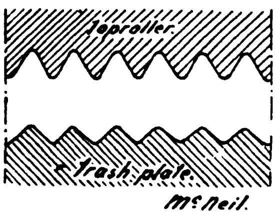


Fig. 197—Grooved Trash Plate.

The scraper for the Messchaert grooves *a* will be dealt with later when referring to *Fig.* 203.

Trash plates are made of cast iron, semi-steel or special cast alloys, such as the British "Wearever" material and the like. Grooved cane rollers need to have the plate tip grooved, and firmly pressed against the roller.

Ungrooved rollers sometimes have a small clearance between the feed roller and the trash plate tip, but bagasse might then adhere firmly to the roller surface, and subject the trash bar to vibrations when it scrapes the bagasse off.

A trash plate co-acting with a grooved cane roller will become grooved on top through wear by the bagasse and these grooves generally spread towards the ends of the trash plate, so are not exactly parallel to the roller grooves.

A British patent has been obtained recently for a *Grooved Trash Plate* as indicated in *Fig.* 197, being the logical sequence of the point just mentioned.

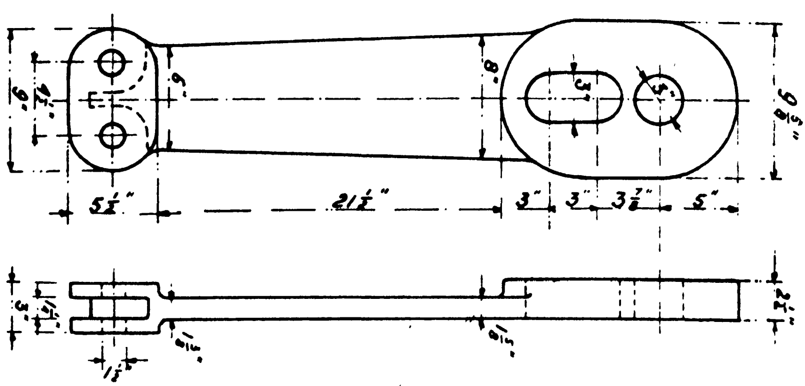


Fig. 198.—Trash Bar Levers.

The sliding type trash bar of *Fig.* 193 is operated by two cast-steel *Trash Bar Levers* as shown in *Fig.* 198, which are pivoted to two 1½ in. tie bolts on each side of the mill headstocks. Heavy forces act on these levers and they may break when tramp iron goes through the mill or under too heavy adjustment.

As the position of the trash plate cannot be ascertained when the top roller is in place, there is now an interesting implement on the market, the

Trash Bar Setter,[1] which will produce an exact copy of the trash plate setting of the empty mill on a piece of paper attached to the top roll periphery, the top roller being in place.

To have a good record of the exact setting of the trash knife, a template should be made of each knife after every crop, so that they can be compared with the previous ones. This will give valuable information as to shape and wear of the trash knives, and faulty settings can thus be remedied.

The number of bolts for trash plates of the flanged type is normally 14 of $1\frac{1}{8}$ in. diameter; flush plates have from 7 bolts of $1\frac{1}{4}$ in. to 20 bolts of $\frac{7}{8}$ in., all for 7-feet mills. The author has used steel of high tensile strength for these bolts, but breakages occurred through the brittleness of the material, and common wrought iron is to be preferred, it having better lasting qualities.

Mill Roller Shafts have gradually increased in size, as may be noted from *Fig.* 199. These shafts were supplied by the author for mills of the same manufacturer and of the same roller length.

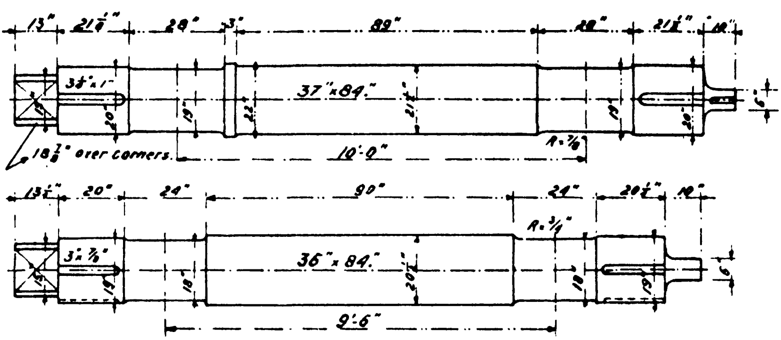


Fig. 199.—Mill Roller Shafts.

The mill shells are attached the same way as mentioned in *Chapter VI* for crusher rollers. Single and double crown wheel arrangements are in use and the shafts should be dimensioned according to the combined bending and torsional moments. The cast iron shell as a matter of fact adds to the strength of the roller, but it should not be considered as a dependable factor, as cast iron has a very varying tensile strength and when grooved rollers are used, the bending stress resisting capacity is greatly reduced.

Steels of special alloy, like nickel steel or similar material, have been advocated for mill roller shafts, and although they behave better against journal wear, the resistance against shaft fatigue in some instances has been less and a ductile material is to be preferred, even if the ultimate tensile strength may be considerably less. In Java the shaft material has from 50-55 kg./mm.^{2} ultimate strength (from 71,000 to 78,000 lbs./sq. in.) and an elongation of 25-20 per cent., *measured on a length of* 200 *mm.* (8 inches). In Great Britain and U.S.A. the elongation is generally measured over a length of the trial rod of 2 in. and this is not comparable with the metric standard of 200 mm. A 25 per cent. elongation on two inches might be less than 10 per cent. on 200 mm. There is no proportion between these values.

The products $50 \times 25 = 1250$ and $55 \times 20 = 1100$ are called the quality limits.

[1] See *Int. Sugar Jl.*, 1929, p. 189.

Mill shafts should not be turned on the lathe with too heavy an advance of the cutting tool, as it will cause microscopic surface cracks. As already mentioned in the previous Chapter, welding by the electrical or acetylene processes is only allowable when the object is properly annealed afterwards.

The specific journal pressure may run to about 1500 lbs./sq. in. when proper lubrication is at hand; thus, for a 19 in. × 28 in. journal, up to 350 tons per journal (700 tons per shaft) is the maximum.

In *Fig.* 200 is shown a *Roller without Crown Wheels or Pinions*, as supplied by the author. The mill in question is driven by three coupling bars and the crown wheels are arranged integrally with the mill gear at fixed centre distances. The coupling bars, therefore, only have to transmit the corresponding torque for each roller.

As the juice drainage takes place over the lower rollers, which have no flanges, the juice will probably spill over on the roller sides, thus entering the side bearings. To prevent this inconvenience or to reduce its effect, ***Juice Rings*** are integrally cast on the lower rollers, as indicated in *Fig.* 201. Formerly mild steel juice rings were shrunk on the shafts, to maintain the keys and also the roller at the predetermined locus on the shaft, but this practice has now been discontinued. A hermetically sealed joint should exist between shaft and shell, so that the juice cannot penetrate there under any circumstances, as heavy corrosion would develop corrosion fatigue.

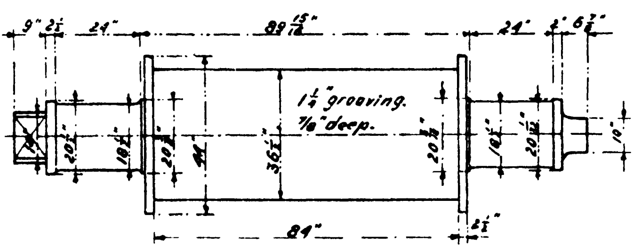


Fig. 200.—Roller without Crown Wheels or Pinions.

Sometimes a sheet iron or brass disc x of $\frac{1}{8}$ in. material is bolted to the juice ring by countersunk bolts, to reduce the danger of juice splashing into the side bearings.

The shell material has to be rough and hard, as roughness will be necessary for good feeding of the mills and hardness will reduce wear. The following analyses are considered good shell material in Java:—[1]

Combined carbon	1·94	2·85
Free carbon	1·74	1·06
Manganese	0·95	4·61
Silica	1·90	0·87
Phosphorus	0·79	0·32
Sulphur	0·01	0·01

Microscopic photographs of the polished and etched shell material are of great assistance for the qualitative determination, and leading manufacturers have laboratories equipped for microscopic and chemical analysis.

The roll wear of each roller should be measured on the periphery after each crop by a steel measuring tape. The wear for good material amounts to $\frac{1}{8}$ in.—$\frac{3}{16}$ in. on the diameter for ungrooved and $\frac{1}{4}$ in.—$\frac{1}{2}$ in. for grooved rollers per 100,000 tons of cane ground, and the life time of ungrooved rollers will be 7 to 8 years for small capacities, whereas grooved rollers under heavy grinding will be worn down to the least allowable diameters within about three

[1] See Q. A. D. EMMEN, "Rietsuikerfabrieken op Java." part I, page 153.

years. A good roller of the larger sizes of mills should grind between 500,000 and 1,000,000 tons of cane; the higher value for large grinding plants.

Every factory should keep a record of all rollers on hand in the milling train, physical qualities such as roughness, and the wear per 100,000 tons of cane ground.

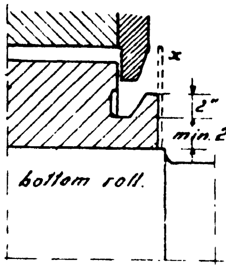


Fig. 201.—Juice Rings.

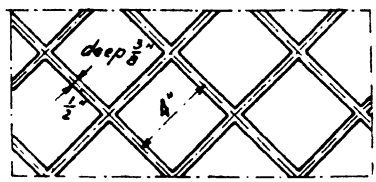


Fig. 202.—Van Raalte Grooves.

Where there are no crushers, the first mill has a difficult task in respect to the feed, and special top rollers, such as the Caterpillar, Paleaz, and Diamond rolls, etc., having a coarse grooving, are then an advantage.

For ungrooved rollers it is sometimes necessary to increase the roughness of the roller periphery by cutting crosswise *Van Raalte Grooves*, ½ in. wide and ⅜ in. deep, as shown in *Fig.* 202. These grooves will soon get filled with compressed bagasse and thus roughen the roller surface. For grooved rollers the Kay patent, composed of longitudinal grooves, may be used.

In Hawaii deep radial grooving of Hind-Renton design has been introduced, having a top angle of 30° and flattened on the outer periphery to ¼ in. by 1 in. pitch. As these grooves do not fill up entirely with bagasse, good drainage of the rollers will take place along the bottom of the grooves. The author has used this grooving in cases where the mill feed has been extremely difficult

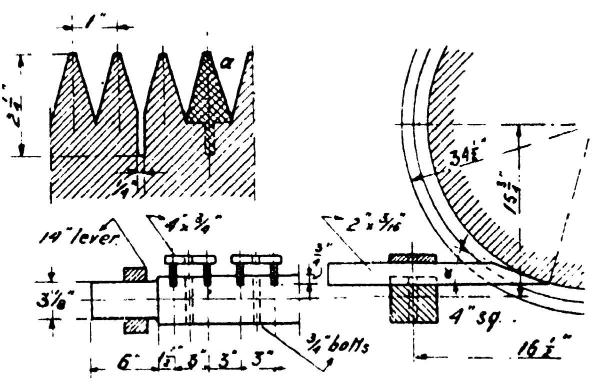


Fig. 203.—Messchaert Grooving.

and good results have been obtained; but the wear of the rollers is considerable and tramp iron will do serious damage. Top angles of 45° showed an improvement in respect to roller wear.

Another type of grooving, originating in Hawaii, is the *Messchaert Grooving*, which has been adopted nearly all over the world. A good arrangement, of which the author has supplied several, is shown in *Fig.* 203. The rollers of this special installation had Hind-Renton grooves, but Messchaert grooves, 2¼ in. deep and ¼ in. wide, were cut at intervals of 3 in. Some operating engineers prefer the Messchaert grooves to have the shaded part at *a* omitted, thus leaving out a full tooth every 3 in. Although it might improve the drainage of the roller, the grooving is weakened and unequal pressure by the bagasse against the ridges adjacent to *a* has caused breakages of these.

The Messchaert grooves ought to have well-guided scrapers, mounted on a shaft without lateral play and supported in bearings of simple construction against the side caps or mill housings. The shaft should be of wrought iron or mild steel, of 4 in. square section with trunnions on both ends and operated by two 14 in. levers. The scraper blades are made of steel 2 in. × $\frac{3}{16}$ in., and the pointing angle α should be about 30°. The blades are firmly held in grooves cut into the shaft at the corresponding loca of the roller grooves. Every two blades are tightened by means of a cover plate 4 in. × ⅜ in. and a ⅜ in. bolt. With a hammer each blade can be adjusted when the mill is at work, so the grooves can be kept completely clear of bagasse. The scraper points are sometimes hardened to reduce the wear.

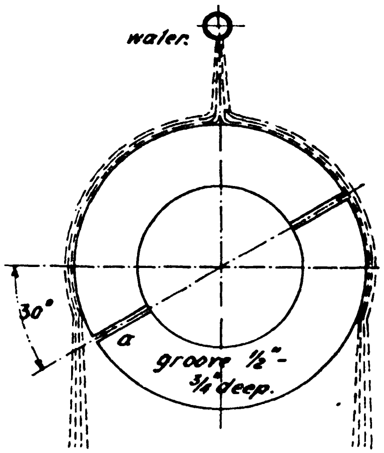


Fig. 204.—Novel Way for removing Worn Shells.

Attempts have been made to get the scraper blades supported by the trash knife as indicated by *a* in *Fig.* 196. There is a heavy wear on these blades and the knife adjustment is interfered with, but the greatest inconvenience is that the grooves are choked by the blades and so the juice cannot drain on the knife-side of the roller, thus reducing the efficiency by 50 per cent., as the Messchaert grooves should drain on both sides of the roller.

On the discharge rollers this arrangement can be used, as the juice is drained off on one side of the roller only, and the scraper blades are laid on the tip of the roller scraper.

Messchaert grooves on the discharge rollers are nevertheless not to be considered as general practice, as is the case on the feed rollers, since the bagasse is not always sufficiently squeezed and dried at the locus of these grooves.

When the scraper blades are not properly guided, there will be heavy lateral wear in the grooves and these are soon widened, making the roller useless in a short time.

To save additional freight, worn shells frequently have to be removed at the sugar factory for shipment of the shafts overseas for reshelling. This shell removal has been done by drilling the worn shell on diametrically opposed lines and inserting wedges with sledge hammers, so as to split the shell. This, nevertheless, is quite a costly job, so some manufacturers have recommended splitting the shell by means of dynamite, inserting cartridges in holes drilled for that purpose. But the explosion has not always caused the rupture desired and damage has been done to the shaft proper.

A novel way for removing *Worn Shells* is shown in *Fig.* 204.[1] A groove $\frac{1}{2}$–$\frac{3}{4}$ in. deep is cut longitudinally round the shell and the roller supported off the ground in such a way, that the groove will make an angle of about 30° with the horizontal. The shell now is heated for about three hours and then rapidly cooled off by means of a stream of cold water from a perforated pipe, about 12 in. above the shell. The cracking will start at *a* and wedge pieces should be inserted to complete the rupture.

When a hydraulic shell press is close at hand, the shaft generally can be pushed out ; in difficult cases, heat should be applied to the shell before the pressure is started.

A *Top Roller Bearing or Bush,* as measured by the author, is shown in *Fig.* 205. These bushes are cast from a good quality of bronze, e.g., phosphor bronze, having a Brinell hardness of about 100°. Both bushes have flanges which embrace the mill headstock, having about $\frac{1}{16}$ in. lateral play. The lower bush is provided with projections *b*, which enter into corresponding niches in the mill cheek, so as to prevent any turning round in the housing. The wear on the lower bush is small, as it has only to support the roller weight, when the mill is running idle. The distance *x* is generally from $\frac{1}{2}$ in. to 1 in.

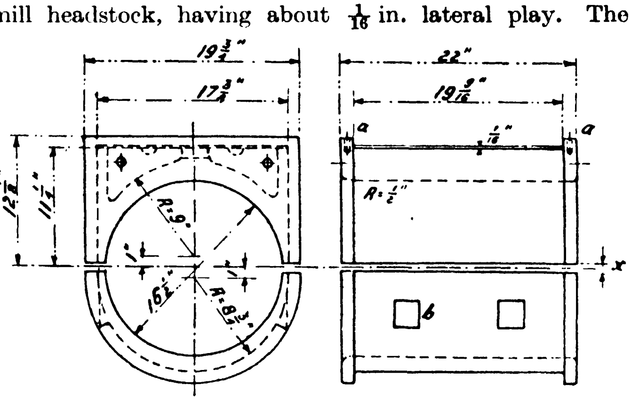


Fig. 205.—Top Roller Bearing or Bush.

The top bush has to withstand the total hydraulic pressure, and a steel pressure plate of sufficient thickness (3 to 4 in.) is laid on top for equal load distribution. Moreover, the side thrust against the side of the mill cheek gap has to be taken up by this bush, and oil grooves should be provided at this side of the bush for lubrication.

Sometimes special steel plates are arranged at this locus and even case-hardened rollers have been used in some instances to reduce the side thrust friction of the top bearing.

For lifting out the bush, tap holes are provided at *a* for inserting eye-bolts.

The thickness of the bearing is augmented on the bearing sides and as the friction heat is excessive, water cooling of the top bush is necessary. The cooling compartments consist of integrally cast ridges, which also reinforce the bearing side and give a zig-zag flow to the cooling water. The cover plate of the cooling compartments is only $\frac{1}{16}$ in. thick, so pressure cannot be applied. This cover plate can be made up to $\frac{5}{8}$ in. thick, as is done by several manufacturers, well bolted by countersunk bolts to the bush, and a reasonable pressure for the cooling water is thus feasible, a point of advantage in case of a hot bearing.

Other manufacturers prefer to have a completely closed water cavity cast in, and making the core for this kind of bearing is more difficult and costly.

[1] See the article of D. BURNS CAMPBELL, *Int. Sugar Jl.*, July, 1932, p. 264.

The amount of cooling water needed can be derived as follows. Considering the journal load to be 225 tons as in previous calculations, and a friction coefficient of 0·12 under good operating conditions, then the friction resistance will amount to :—

$$225 \times 2240 \times 0{\cdot}12 = 60{,}480 \text{ lbs.}$$

This friction resistance acts on the radius of the journal and at 4 r.p.m. the speed is : $\frac{16{\cdot}5 \times \pi \times 4}{12 \times 60} = 0{\cdot}288$ ft./sec. The power input to overcome this friction resistance obviously amounts to : $60{,}480 \times 0{\cdot}288 = 17{\cdot}418$ ft. lbs./sec. = 31·6 h.p.

As 1 B.Th.U. = 777·64 ft. lbs., being the mechanical heat equivalent, 17,418 ft. lbs./sec. is equivalent to 23 B.Th.U./sec. Under tropical conditions the cooling water will enter at about 85°F. and leave at about 125°F. and thus each lb. of water will absorb 40 B.Th.U., or one Imp. gal. 400 B.Th.U. For the total heat absorption of the bearing, being 82,800 B.Th.U.'s per hour, 207 Imp. gals. of cooling water are required. It should be recollected that a part of the heat is radiated to the surrounding air.

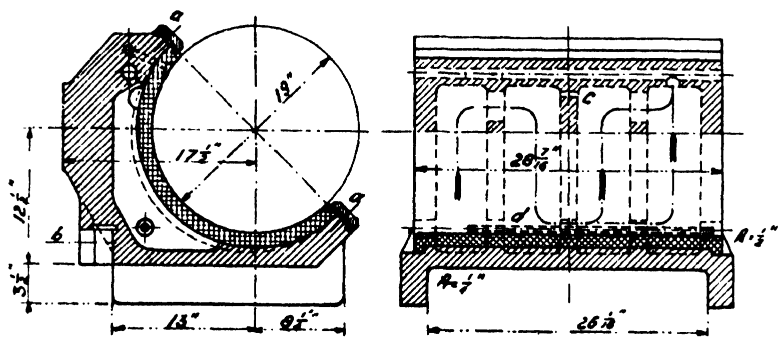


Fig. 206.—Sectional Views of Cast Steel Bearings.

The connections for cooling water to movable bearings have to be made by flexible hose. The entrance has a regulating valve and the outlet should be above an open funnel of the discharge main, generally laid below the mill bedplate, so that the water current can be observed and the temperature of the discharge ascertained by touching. It sometimes happens that the cooling water connections and/or the bearings leak and juice dilution results, which is inconvenient, as it gives the wrong impression of the maceration effect and is a useless surcharge on the evaporators. Cooling water connections, therefore, should be tight.

Fig. 206 gives sectional views of a *Cast Steel Side Bearing* or Quarter Box with integrally cast pressure plate, designed and supplied by the author for existing mills. The bearing has a loose bronze bush, firmly held by the clamps *a* in the finished housing bore. The space between the bronze bush and the steel casting forms the space for the water circulation. A $\frac{3}{4}$ in. tube *d* is inserted for introducing the cooling water, whereas the hot water is discharged at the upper part of the bearing. To avoid air pockets an air vent *C* is provided. At *b* a T-head bolt is inserted for pulling back the bearing.

The side bearing has high flanges on the lower part, which embrace the mill cheek, so that the roller centre may be raised by means of filler plates in case the rollers are worn.

Side bearings completely of bronze with the cooling water compartments cast in, are also in general use. In case the adjustment is made by a thrust bolt in the side cap, a heavy steel pressure plate has to be provided.

Cast steel bearings with white metal linings are used in Java, where the machinery is kept under excellent mechanical supervision. In other countries, this practice has not been copied and bronze is preferred, as the high melting point of the latter will prevent the lining pouring out in case of a hot bearing. The friction coefficient of white metal is lower than that of bronze.

Lubrication is of paramount importance, in respect to power input and wear, but sometimes the entrance of juice or leaking water from the bearings makes lubrication a difficult problem. In early days and in desperate cases, castor oil was used for hot bearings, but nowadays this expensive lubricant is not to be found in many sugar factories.

It should be mentioned that when cooling off a hot bearing, the bronze bushing may shrink more rapidly than the journal and the bush thus may contract on the shaft.

Side Caps are generally made of the same material as the mill cheeks and here cast steel has found wide application. In *Fig.* 207 such a side cap is illustrated for a 34 in. × 84 in. mill, having a heavy 5 in. thrust bolt with square thread, the brass hexagonal nut being embedded in the steel casting. The thrust bolt is secured by four ¾ in. set pins, so that it will not become loose during operation. A tap hole for a heavy eyebolt is provided on the upper part of the cap for lifting purposes.

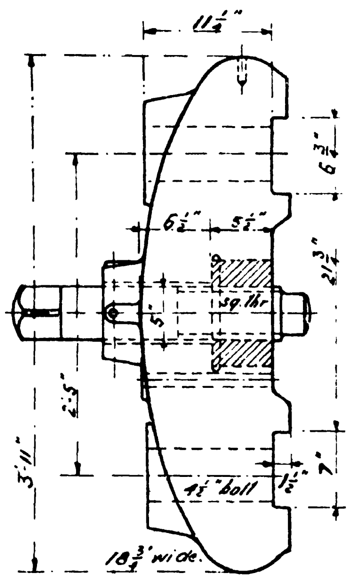


Fig. 207.—Side Cap.

Crown Wheels or Mill Pinions, as in *Fig.* 208, are made of cast steel and the teeth are unfinished, but accurately machine moulded, so that the hard foundry skin will the better resist wear. Formerly crown wheels had 21 teeth and although smooth running was assured, the strength of each individual tooth was low and breakages occurred. Moreover, the height of the tooth profile was less than that obtained by a larger tooth pitch, so the adjustment of roll centres was more limited. Nowadays crown wheels having 17 teeth are to be considered as standard and even wheels with 14 machined teeth have been used, but the advantage of smooth running is somewhat impaired with the latter.

The width of the crown wheels varies. For double crown wheels a width of 14 to 16 in. for 7-foot mills is employed, whereas single crowns have been supplied in some instances up to 22 in. for these mills.

The crown wheel teeth should be calculated for maximum forces as indicated in *Fig.* 182.

The bore does not bear over the whole width, but has a centre core, reducing the weight and cost of machining. The keyways are arranged in different ways, being ⅛ in. taper per foot. At *a* are drawn the keyways for double crown wheels, which are staggered half the tooth pitch and as the shaft keyways on both ends run in line, the crowns on both sides do not intermesh synchronically, thus ensuring smooth running.

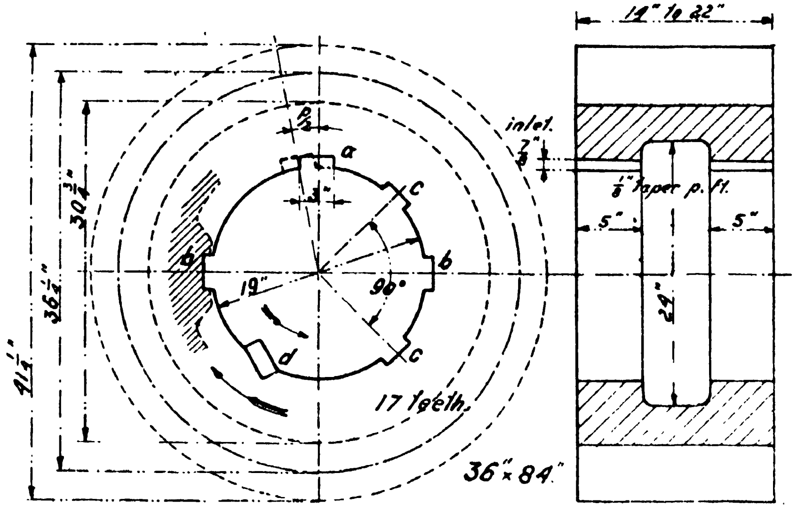


Fig. 208.—Crown Wheels.

The bore is about 0·01 in. larger than the shaft and as one key may pull the crowns a trifle eccentrically, two opposed keys are used by some designers, as indicated at *b*. In this case a greater clearance of the bore can be allowed.

Other manufacturers use two keyways at 90° as indicated at *c*. Tangential keys as shown at *d* are now obsolete, as they are only for one direction of rotation and a second key for the opposite direction is provided, this being necessary for fitting the keys.

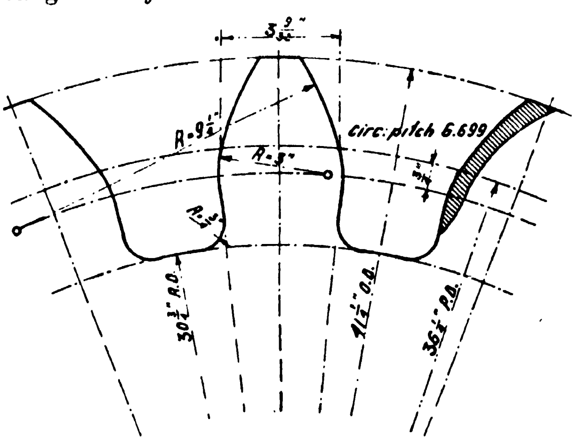
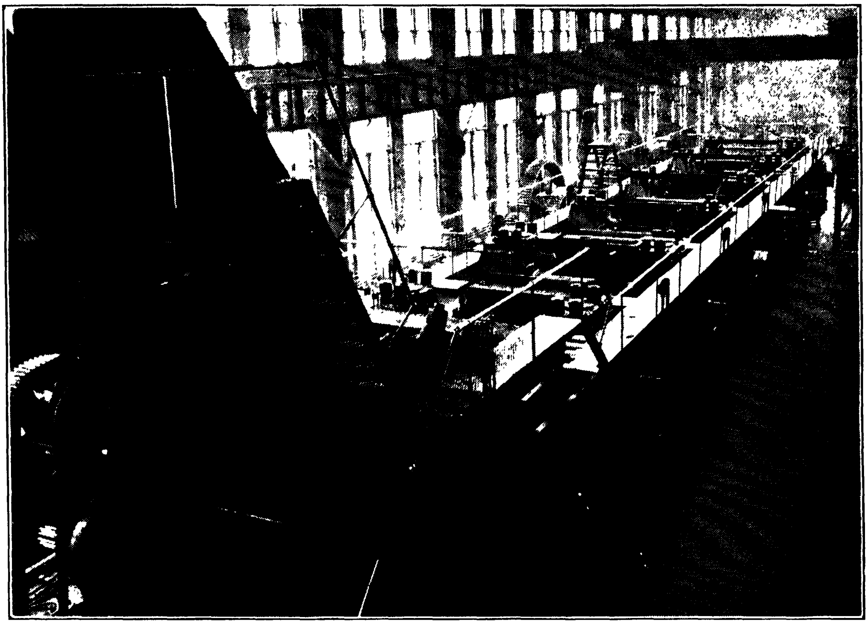


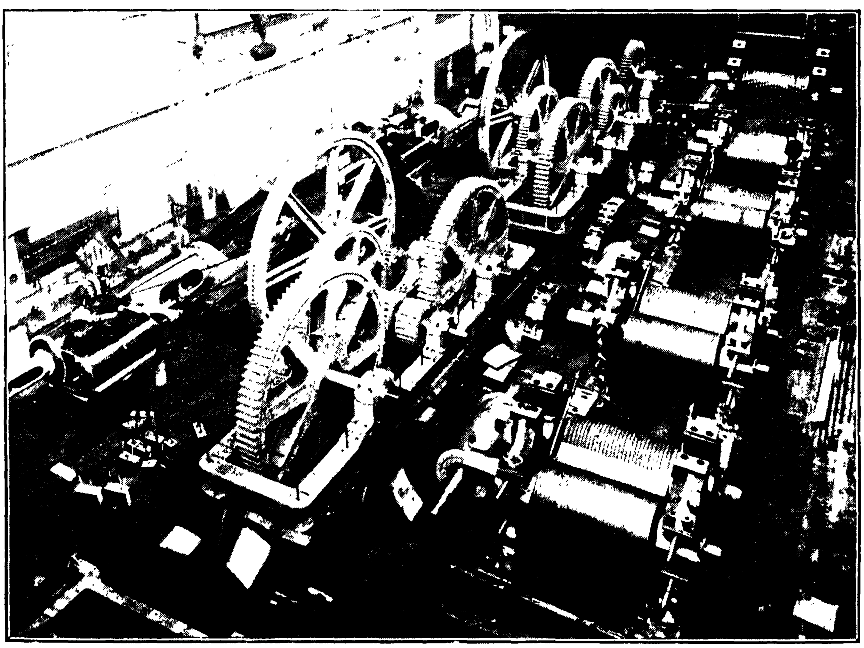
Fig. 209.—Involute Tooth Form.

The single key is widely used and is the most economical, and it can easily be made of sufficient strength.

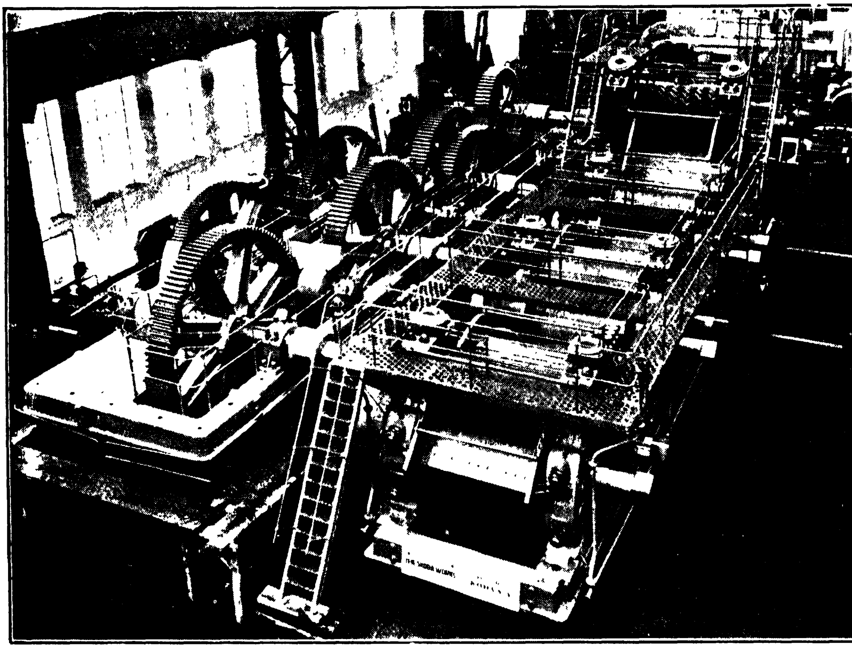
The centre core is filled at the locus of the keyway as indicated at *b*, so the key will be seated over the whole width of the crown wheel.



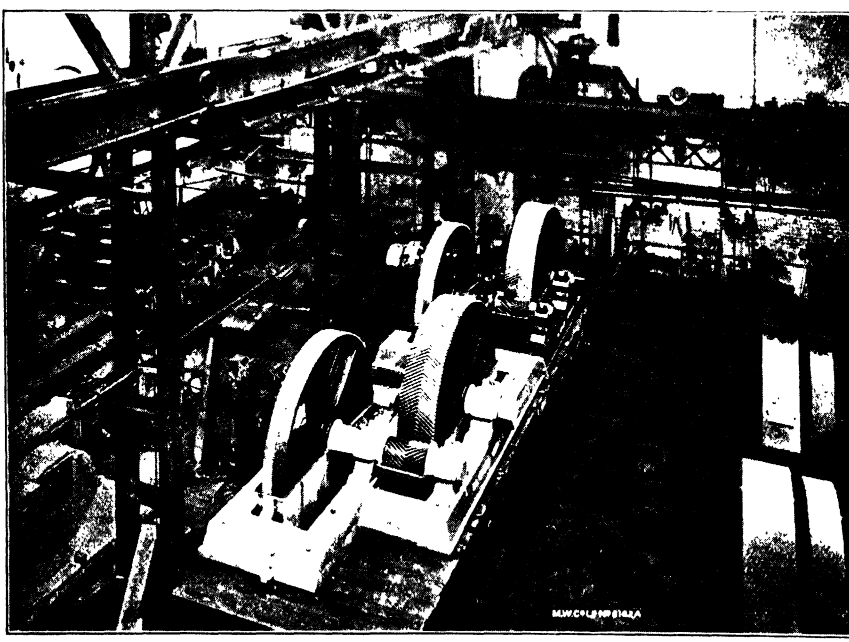
36 in. x 84 in. EIGHTEEN-ROLLER SUGAR MILL AND 42 in. x 84 in. SINGLE CRUSHER (PUNTA ALEGRE).
(*Farrel-Birmingham Co., Inc.*)



14-ROLLER CRUSHING PLANT, 33 in. x 66 in.
(*A. & W. Smith & Co., Ltd.*)



14-Roller Cane Mill with Compound Gearing.
(*Skoda Works, Ltd.*)



Totally Enclosed High Ratio Gears for Cane Mills.
(*The Mirrlees Watson Co., Ltd.*)

An *Involute Tooth Form* is used for crown wheels, as it is not bound to fixed centre distances. In *Fig.* 209 is shown the tooth form for the crown wheels of 36 in. × 84 in. mills, of which the author has supplied many. It is an 18° involute and it produces strong teeth. A variation of 2 in. in centre distance is allowable and in cases where greater variation is required, a special tooth form should be designed by changing the involute angle, but the longer the tooth, the thinner the width on the top.

On the right hand side of the Fig. is shown the approximate wear of the teeth after several years of service. As the centre distance varies, any cyclic form of wear does not take place. With worn teeth, the crowns do not co-act well and an irregular rotation of the driven roller is the result.

In times of depression, worn crown teeth have been restored to their original shape by electric welding. This practice will cause heat stresses and the work has to be done by an expert welder, as otherwise the welded part will soon fall off. Hence, tooth welding should be considered only as an emergency repair.

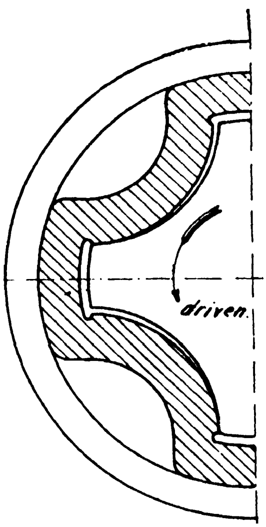


Fig. 210.—Cross Type Coupling.

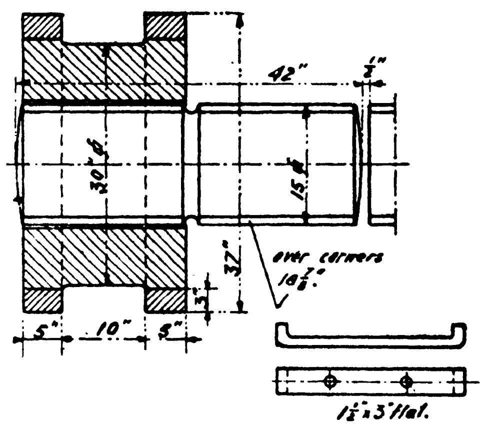


Fig. 211.—Cast Iron Coupling.

Crown wheels with shrouds are still in use. The original design with a small pitch received additional strength as the tooth bases were connected at both ends. Moulding of these teeth is rather difficult and an exact tooth form hard to produce in cast steel.

The top roller of the mill receives its revolving motion from the main gear shaft by means of two couplings and a coupling or tail bar. These couplings are built to allow the lateral movement of one of the two connected shafts so that the top roller may lift under operation of the mill.

A *Cross Type Coupling* is shown in *Fig.* 210 ; this coupling has large bearing surfaces for the coupling bar, the material used being cast steel and the bearing surfaces being machined.

The top roller and the main gear shaft have to be exactly aligned in both the vertical and the horizontal planes in such a way, that under normal milling conditions the shaft centres will be in line. Most of the couplings and tail bars are damaged by excessive wear through improper alignment.

Cast Iron Couplings are generally reinforced by wrought iron rings of 3 in. × 5 in., as shown in *Fig.* 211, shrunk on the outside diameter. Couplings should be made of tough cast iron with a high Brinell hardness, so as to reduce the wear.

The *Square Coupling* as drawn in *Fig.* 212 is widely used and holds a predominant place on account of its easy construction and shop work. The coupling is of cast steel and the clearance generally $\frac{1}{8}$ in. to $\frac{1}{4}$ in. around the coupling bar. During operation the coupling will bear in the corners. Although the bearing surface is less than with the cross type coupling, these square couplings have given good operating performance owing to good alignment of the coupled shafts.

It is sometimes assumed that the couplings are a safety device for the mill, but this does not always apply, as the author has seen broken couplings caused by disalignment of the corresponding shafts, and twisted coupling bars caused by the choking of the mill or crusher owing to foreign bodies passing through.

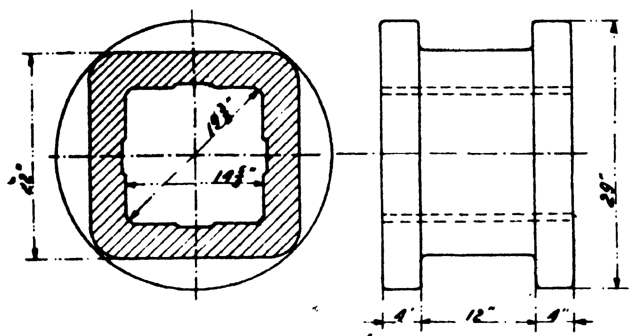


Fig. 212.—Square Coupling.

The coupling or tail bar should have about $\frac{1}{2}$ in. play between mill and gear shaft ends for easy mounting or dismantling of the mills. The ends of the tail-bar are conveniently of spherical shape.

When the couplings are in place a flat iron with bent ends, 3 in. × $1\frac{1}{2}$ in., is bolted to the bar between the couplings, to prevent the couplings slipping off the shaft squares.

The minimum length of the tail bar has to be twice the length of the coupling. In *Fig.* 211 a groove about two inches wide is turned midway for attaching the hoisting cable or chain, and the tail bar is made correspondingly longer.

There are different new constructions of tail bars and couplings designed to achieve better bearing of the bar against the inside surfaces of the couplings, but so far they have not received general introduction.

The *Friction Resistance Diagram* of a cane sugar mill is shown in *Fig.* 213 and apparently the power input of a given mill is no more than its frictional resistance. This friction is caused by the compressing action of the mill rollers on the cane or bagasse, and is composed of the following items :

A.—Top roller bearing friction.
B.—Feed roller bearing friction.
C.—Discharge roller bearing friction.
D.—Turn plate scraping friction.
E.—Bagasse friction over the turn plate.
F.—Top roller scraper friction.
G.—Discharge roller scraper friction,

the rolling friction of the bagasse between the rollers being neglected.

The frictional resistances A, B and C may be considered as caused directly by the milling performance, whereas the other resistances, although they are unavoidable, are useless to that performance proper.

Considering *Fig.* 213, the total *Power Input of a Mill* obviously has to be:

$$T \times v + F \times v + B \times v + S_{,} \times V + S_{,,} \times V + S_{,,,} \times V + P \times V_{,}$$

or: $(T + F + B) \times v + (S_{,} + S_{,,} + S_{,,,}) \times V + P \times V_{,}$ .... (70)

where T, F and B are the total pressures on the rollers derived as in *Fig.* 182, multiplied by the friction coefficient, which for good bronze bearings and good lubrication will amount to about 0·15 for average running conditions, thus: $T = H \times \mu$, where H is the hydraulic load on the top roller and μ the above mentioned friction coefficient.

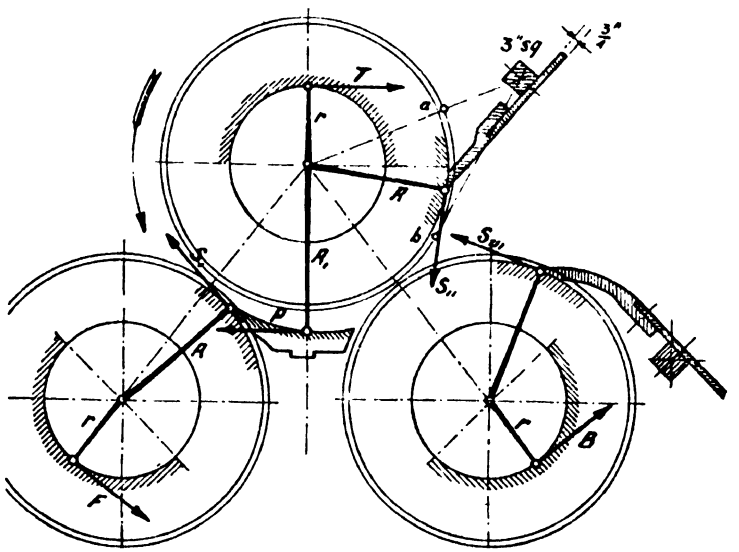


Fig. 213.—Friction Resistance Diagram.

v is the speed in feet per second, these frictional resistances acting against the direction of rotation, thus:

$$v = \frac{\pi \times 2r \times n}{12 \times 60}$$

r being the journal radius in in. and n the number of revolutions of the mill per minute.

The forces $S_{,}$, $S_{,,}$ and $S_{,,,}$ are the forces with which the trash knife and scrapers are pushed against the rollers, multiplied by the dry friction coefficient μ, of about 0·33. These forces are unknown beforehand, but the acting speed in ft./sec. can be derived from:

$$V = \frac{\pi \times 2R \times n}{12 \times 60}$$

R being the roller radius in in.

The frictional force P, being the product of the bagasse pressure on the trash plate, multiplied by an unknown friction coefficient, acts with a speed V_1 derived in the same way as above mentioned for v and V, by inserting the radius R_1. The pressure on the bagasse is not known either, but it is a part of the hydraulic top roller load and will diminish the roller pressures correspondingly.

From a mill of known dimensions, the author has checked the power input of the driving engine, and the data obtained may be taken for overall calculations. The power consumption of the engine and gear under empty load amounted to 0·35 of the power consumption of the total empty load of mill and gear with scrapers attached, and the mill power input proper has been taken as 65 per cent. of the indicated horse power, although there might be a difference when grinding.

Rollers : 34 in. × 72 in. Journals 16 in. × 19 in.
Hydraulic rams, 13 in. dia.
Hydraulic pressure, 4000 lbs./sq. in.
Hydraulic load on top roller, 1,000,000 lbs. after deduction for leather cup friction.
Cane roller pressure, 415,000 lbs.
Bagasse roller pressure, 830,000 lbs.
Engine, 26 in. × 52 in. Power input under load 256 mean i.h.p. (last mill).
No. of revolutions of mill, $n = 2{\cdot}3$ r.p.m.

Calculated data :

$v = 0{\cdot}16$ ft. per sec.
$V = 0{\cdot}341$ ft. per sec.
$V_1 = 0{\cdot}401$ ft. per sec.
Mill power input $0{\cdot}65 \times 256 = 166{\cdot}4$ h.p. $= 91{,}520$ ft. lbs./sec.

The total roller bearing load, therefore, amounts to 2,245,000 lbs., which needs a power input of :

$$2{,}245{,}000 \times 0{\cdot}15 \times 0{\cdot}16 = 53{,}880 \text{ ft. lbs./sec.}$$

The scraper friction is not known when under milling performance as, in addition to the metal friction, the adhering bagasse has to be scraped off, and even assuming that the total scraping resistance amounts to 100 lbs. per in. roller length, the total power input for the three rollers only amounts to :

$$3 \times 7200 \times 0{\cdot}341 = 7{,}365 \text{ ft. lbs./sec.}$$

This may indicate that the power consumption for the scraper friction is of no great importance. By simple deduction there remains 30,275 ft. lbs./sec for the trash knife friction, and this friction force will amount to $30{,}275 \div 0{\cdot}401 = 75{,}500$ lbs.

With a high friction coefficient of 0·50, the pressure lost on the trash knife amounts to 151,000 lbs., being 15·1 per cent. of the total hydraulic load on the top roller. The power consumption for actual milling is 59 per cent. of the power input at the mill, or only about 40 per cent. of the indicated power input at the driving engine.

This calculation clearly indicates that the trash knife is consuming a considerable friction load and the power input of a mill can in some instances be reduced by lowering the trash bar, as the author has done on several occasions. The mill without trash bar, as shown in *Fig.* 181, ought to have a reduced power consumption.

From indicator cards the author has reached the conclusion that the power input is approximately proportionate to the mill speed.

The top scraper (*Fig.* 213) is attached to a 3 in. square shaft, on which a ¾ in. plate is mounted, having a *Cast Iron Scraper Tip*, as indicated in *Fig.* 214. The top scraper is thus arranged to touch the top roller along the horizontal centre line, being midway between the shaft centre tangent at *b* and the shaft centre line at *a* (*Fig.* 213).

The bottom scraper is curved and of the same construction as the top roller scraper. The slanting of the discharge plate is about 45° with the horizontal.

The scrapers have adjusting levers, which are spring-loaded, so the scraper tips press against the rollers with about 5–15 lbs. per inch of roller length. In some instances the springs are removed and a piece of tube is inserted. For the discharge roller this might be allowable, but for the floating top roller springs are to be preferred.

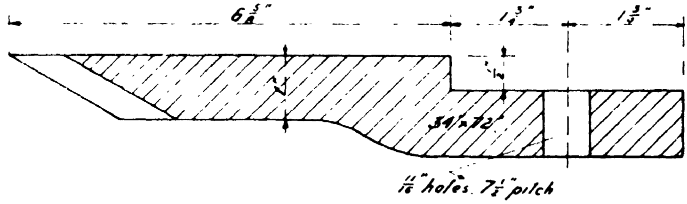


Fig. 214.—Cast Iron Scraper Tip.

The scraper tip of *Fig.* 214 has a thickness of 1 in. and the grooving is cut beforehand for intermeshing with the roller grooves.

Steel scraper tips are not used for grooved rollers, on account of wire edging and roller polishing. The wear on the scraper tips is heavy and generally they are renewed after two crops, depending on the grooving, the peripheral speed and the roughness of the roller. Special alloys are also used for these scraper tips, assuring longer wear.

The factory engineer has to get his information about the milling performance from the laboratory report, and the chemists' staff will supply him with figures for juice and sucrose extraction, sucrose and moisture in bagasse and/or lost juice in final bagasse. There is nevertheless the inconvenience

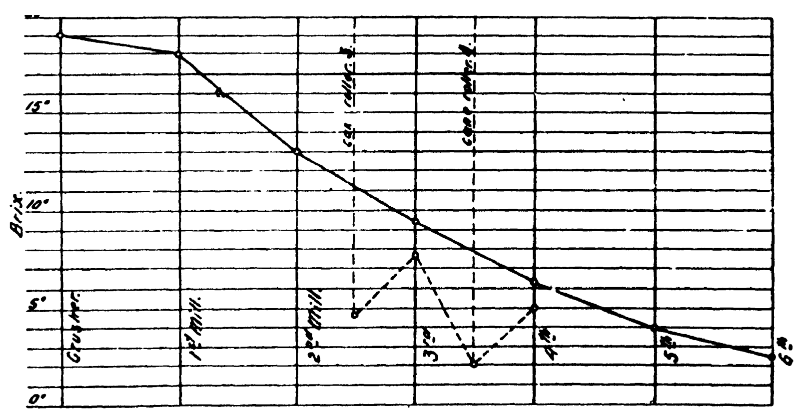


Fig. 215.—Juice Brix Diagram.

that only final results of the milling train are given, which are doubtless of paramount importance for the following manufacturing process, but do not give information as to the performance of each individual mill.

For individual mill data the *Juice Brix Diagram* (*Fig.* 215) is very useful and the author has applied it in many instances to improve the milling performance.[1]

[1] See the article by B. P. LUCE, *Proceedings Cuban Sug. Tech. Ass.*, 1928.

The Brix of the crusher juice and the juice of the discharge rollers of the following mills should be plotted in the diagram and a downward sloping line towards the last mills be obtained. When a zig-zag line is produced, the corresponding mills should be checked as to setting and speed and the maceration or imbibition be then improved. A fair sample of these juices has to be taken in the customary way and in case information is desired of each pressing, the Brix of the feed roller juice as well as that of the discharge roller should be analysed. The dotted lines in *Fig.* 215 indicate the values obtained from the last mills of a 14-roller milling train ; and it is interesting to note the large difference in Brix between the two pressings of each mill and it is obvious that the maceration effect is very pronouncedly shown in the cane roller pressing, whereas the discharge roller pressing has a much higner Brix, due to additional rupture of juice-containing cells.

4.—General Mill Data.

The power consumption of mills in actual operation varies greatly, and the author can cite a case where the driving engines were indicated by him. The plant was composed of 14 rollers, 34 in. × 72 in., and ground 1600 tons of cane per 24 hours. The crusher and the first three mills were driven by a 32 in. × 60 in. Corliss engine, whereas the fourth mill was driven independently by a piston valve engine, 26 in. × 52 in. The indicated power input of the first engine was 560 i.h.p. whereas the second engine developed 256 i.h.p. The milling performance was very good and nothing abnormal could be detected ; the last mill, though of older construction, was nevertheless in first class condition.

The Java Experimental Station has published very conscientious data of the *Mean Power Consumption of Mill Engines* and they are reproduced here.[1]

Crushers	7 — 31	mean	15 i.h.p.	Per short ton of fibre ground per hour.
First mills	13 — 46	,,	25 ,,	
Second ,,	12 — 34	,,	21 ,,	
Third ,,	10 — 28	,,	17 ,,	
Fourth ,,	9 — 25	,,	16 ,,	
Fifth ,,	8 — 21	,,	14 ,,	

The last mills have a smaller mean power consumption than the earlier ones in the train whereas, in Cuba and Hawaii, the power input of the last mills is generally larger.

The prevailing practice in Java is to have each mill driven by a separate steam engine. The regulation of the individual mill speeds is thus quite feasible, but it should not be overlooked that mill speed is only one factor of grinding cane, and in other countries compound mill drives are very frequent, these having been introduced by British designers.

In 1931 there were in Java	8	engines driving	3	mills.
	76	,, ,,	2	,,
	641	,, ,,	1	,,

Independent crusher drive is to be preferred, as it will be very convenient for varying capacities or varying fibre contents of the cane.

Sizes of Mill Engines for normal steam pressures from 90 to 125 lbs. in actual operation may be tabulated as follows :—[2]

1 See *I.S.J.*, 1932, p. 392.

2 See also *Het Archief*, 1925, No. 10 ; and A. B. GILMORE, "The Cuba Sugar Manual," 1928.

		Inches			
3	mills	26 × 48	....	engine	26 in. × 48 in.
2	,,	26 × 48	....	,,	24 in. × 44 in.
1	,,	30 × 54	....	,,	18 in. × 42 in.
1	,,	30 × 60	....	,,	24 in. × 36 in.
1	,,	30 × 60	....	,,	550 mm. × 1100 mm. (22 in. × 44 in.)
1	,,	32 × 72	....	,,	650 mm. × 1100 mm. (26 in. × 44 in.)
2	,,	32 × 72	....	,,	800 mm. × 1200 mm. (32 in. × 48 in.)
Cr. and 2	,,	32 × 72	....	,,	850 mm. × 1200 mm. (34 in. × 48 in.)
Cr. and 3	,,	34 × 72	....	,,	32 in. × 60 in.
2	,,	34 × 78	....	,,	850 mm. × 1200 mm. (34 in. × 48 in.)
1	,,	34 × 78	....	,,	775 mm. × 1100 mm. (31 in. × 44 in.)
2	,,	34 × 78	....	,,	600 mm. × 900 mm. (triple gear) (24 in. × 36 in.)
Cr. and 1	,,	36 × 78	....	,,	32 in. × 48 in.
Cr. and 1	,,	36 × 84	....	,,	30 in. × 54 in.
3	,,	36 × 84	....	,,	40/42 in. × 60 in.
Cr. and 4	,,	36 × 84	....	,,	50 in. × 60 in.
1	,,	36 × 84	....	,,	750 mm. × 1100 mm. (30 in. × 44 in.)
2	,,	36 × 84	....	,,	30 in. × 60 in.
1	,,	36 × 84	....	200-350 h.p. electric motor	
2	,,	36 × 84	....	engine	750 mm. × 800 mm. (triple gear) (30 in. × 32 in.)
3	,,	37 × 84	....	,,	42 in. × 60 in.

Journals on Mill Roller Shafts are to be found of the following dimensions:

30 in. × 60 in.	mills	from	15 in. × 19 in.	to	12 in. × 14 in.
32 in. × 72 in.	,,	,,	16 in. × 22 in.	,,	15 in. × 18 in.
34 in. × 78 in.	,,	,,	18 in. × 24 in.	,,	17 in. × 20 in.
36 in. × 84 in.	,,	,,	19 in. × 28 in.	,,	18 in. × 24 in.

Maximum Hydraulic Load on the top rollers ranges as follows:—

26 in. × 48 in.	mills	200 tons.	Rams	7 in.	diameter
30 in. × 60 in.	,,	415 ,,	,,	11 in.	,,
32 in. × 72 in.	,,	560 ,,	,,	13 in.	,,
34 in. × 78 in.	,,	530 ,,	,,	15 in.	,,
36 in. × 84 in.	,,	670 ,,	,,	12-16 in.	,,

Hydraulic Accumulators have leather cup packings as a rule and they are not a source of trouble, save when the leather packing bursts, due to unclean or grooved rods. The rods therefore should be well polished.

There are two types of hydraulic accumulators, the upright and the suspended. The upright type has the hydraulic cylinder on the bedplate and the ram is subject to compression. The other type has the hydraulic cylinder mounted on a platform and the differential ram is mounted with the smallest diameter downwards. The ram is in tension, when under load and the only thing to be ensured is sufficient rigidity of the platform. The upright type has one set of leather packings, whereas the suspended type has two. Both types are widely used.

Another type is the air chamber accumulator, where the weight of cast iron plates is replaced by a cylinder containing compressed air. The space occupied by these accumulators is small. Air leakage has to be avoided by an oil seal.

Hydraulic accumulators should have a safety device, so that the oil can escape when the highest position is reached, as otherwise ruptures will occur.

The capacity of the accumulator, of course, has to be in accord with the hydraulic ram cylinders it has to serve.

Hand or steam hydraulic pumps are used for filling the system with heavy oil. Hydraulic manometers should indicate for each mill the pressure prevailing; and the load in tons for the hydraulic rams of a given diameter should also be stated on the manometer dial.

Running Platforms are essential for easy supervision and inspection, and are laid alongside the mill housings at a level flush with the top caps. Crosswise connexions should be made after each mill, but sometimes they are laid between the top caps of each mill which is less convenient. Stairs should give easy access to these running platforms and railings all around should be provided. For dismantling or removal of the rollers, the platforms have to be so arranged that sections can be taken out, the rest remaining intact.

The crown wheels are covered by *guards* of sheet iron of sufficient thickness so as not easily to buckle. An oil bath should be arranged underneath in which the teeth of the lower crown wheels dip.

Roller Bearings of the conical roller type, as in *Fig.* 41, are used for steel rolling mills and are recommended for cane mills as well. The author has no practical operating data available so far, but developments should be watched with interest, as great reduction in power input for milling might be achieved.

The *Peripheral Mill Speed* is kept between 12–24 feet per minute in Java, but in Cuba speeds up to 50 feet for the last mills are now advantageously employed, so as to reduce the blanket thickness.

Weights of Complete Mills without gears or engines are approximately as in the following table :—

30 in. × 60 in.	100,000 lbs.
32 in. × 72 in.	140,000 lbs.
34 in. × 78 in.	180,000 lbs.
36 in. × 84 in.	215,000 lbs.

Milling Results are variously recorded in different countries, although there is now internationally adopted a "Lost Juice in Final Bagasse per 100 Fibre" formula, and the following tabulation[1] will show its comparison with the former sucrose and normal extraction :—

	Factory *A*.	Factory *B*.
a. Fibre in cane	10%	14%
b. Undiluted juice........................	90%	86%
c. Sucrose extraction	93%	93%
d. Normal extraction ($b \times c$)	83·7%	80%
e. Undiluted juice per 100 fibre ($d \div a$)	837	571
f. Undiluted juice lost per 100 cane ($b - d$)..	6·3%	6%
g. Undiluted juice lost per 100 fibre ($f \div a$) ..	63	43

This shows clearly that, with equal sucrose extraction, the milling performance of Factory *B* is superior to that of Factory *A*.

1 From "Reference Book of the Sugar Industry," 1928.

The average results obtained with different milling plants mentioned below are taken from the "Eindstaat 1931" of the Java Experimental Station in Passeroean, being the most complete data compiled :

Plant of 3 mills........	59	lost	juice	per	100	fibre.
Crusher and 3 mills	50	,,	,,	,,	,,	
4 mills	46	,,	,,	,,	,,	
Crusher and 4 mills	40	,,	,,	,,	,,	
5 mills	37	,,	,,	,,	,,	
Crusher and 5 mills	32	,,	,,	,,	,,	
6 mills	24	,,	,,	,,	,,	

The average imbibition amounted to about 140 parts water per 100 parts fibre. It is apparent from these figures that an extra mill represents a higher gain in milling performance than a crusher.

The *Moisture in Bagasse* amounts to about 50 per cent. and can be as low as 41 per cent.; sucrose in bagasse between 5 and 2 per cent.

The percentage of fibre on cane varies widely for different countries, ranging between 17 and 10 per cent. or even lower.

CHAPTER VIII.

GEARS FOR MILLS AND CRUSHERS.

GEAR ARRANGEMENTS—GEAR DETAILS—GENERAL GEAR DATA.

The prime movers of mills and crushers operate at a much higher speed than the latter, so a reduction gear is required, to step down the speed. Electric motors for mill drives run at a speed of about 450 r.p.m., while steam engines range between 45 and 150 r.p.m.

The gear ratio of large single gears requires a limit for good operation of about 1:7, as otherwise the pinions would be too small or the spur wheels too large, the latter proving awkward in transportation or erection. Machine-generated gears are now made to the highest precision and for ratios up to 1 : 25, as used for ship propulsion, but the low speed of the mill roller shafts requires considerable strength and consequently a large circular pitch of the teeth, thus rendering these high ratio gears as yet unsuitable for operating sugar mills. The highest ratio known to the author for intermediate gears for sugar mill drives is 1 : 16·6 and for main gears 1 : 8.

Worm drives are not used for sugar mills, as the efficiency for large ratios is low compared with spur gears and the wear is considerable with the heavy forces employed.

For low speed steam engines a total ratio of 1 : 25 or less is required, and double gears are used, whereas for electric motors a total ratio of about 1 : 125 or less is necessary, requiring a triple or in a few instances a high ratio double gear.

On account of the low speed of mills and engines, the mill gears used formerly were of an unfinished type. These had good wearing resistance thanks to their hard foundry skin, but the higher speeds of to-day call for machine-cut tooth forms, which have less friction and thus yield a higher efficiency.

The working parts of the gears, thus the toothed rims of the built-up spur gears, are made of cast steel. Cast iron is obsolete now for these important parts of the gear.

1.—Gear Arrangements.

In Java the individual mill drive is favoured, but driving several mills by one engine will reduce the space occupied and needs less material, thus reducing the total cost of the equipment. A large sized steam engine will have a more economical steam consumption than several small ones, and supervision, maintenance and repairs will be less.

The inherent disadvantage of several mills being driven by one engine is that the individual mill speed cannot be changed without interfering with the others in the same train. It may be offset by the fact that mill speed is not the only factor to be considered in cane milling and a proper mill setting plays an equally important part, although changes in setting cannot be as easily accomplished as changes in the speed of the driving engine.

The compound gear was originally designed by British engineers and the author has not found any inconvenience in practice with these compound gears, when the mill setting is properly fixed and the individual speed of each mill is accurately determined. Crushers, preferably, should have independent drives, as it will give a more flexible arrangement for higher capacities and the different fibre contents of the cane supplied.

In *Fig.* 216 is shown a *Compound Double Gear* for one crusher and three mills, 36 in. × 78 in., driven by one 46 in. × 60 in. Corliss engine. On the engine shaft are mounted two pinions of 20 teeth each, driving the two intermediate gears, having 96 teeth. The main gears have pinions with respectively 22, 23 and 24 teeth for the 1st, 2nd and 3rd mills, the spur wheels all having 96 teeth. The 3rd mill, therefore, has a higher peripheral speed than the first one, which is common practice in America and has given good milling results.

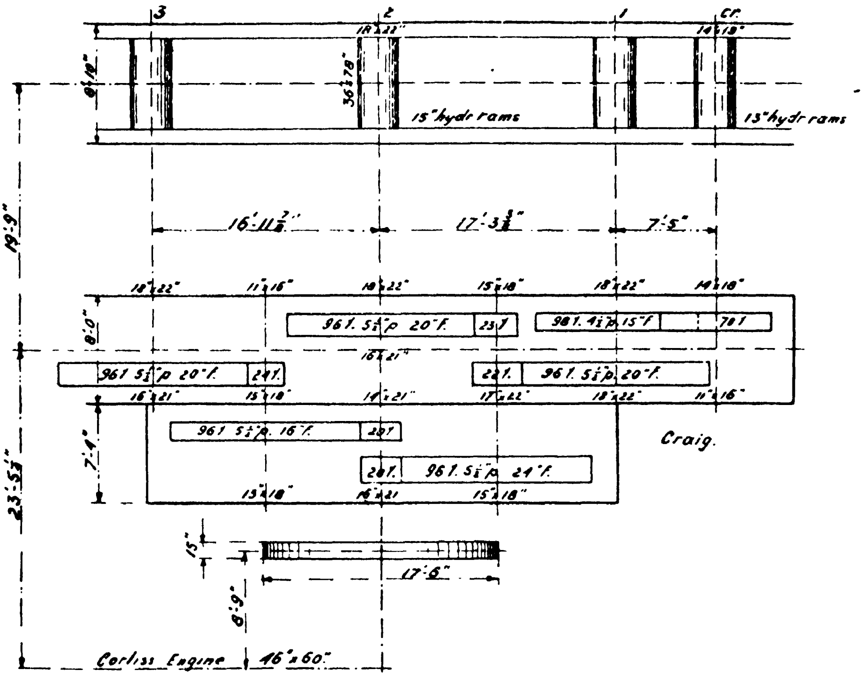


Fig. 216.—Compound Double Gear.

The crusher is driven from the first main gear shaft by a set of spur wheels, having 98 and 70 teeth respectively.

All the intermediate and main gears have 5½ in. pitch—when not otherwise mentioned the circular pitch is meant—four wheels having a face of 20 in. and one having 16 in. face. The crusher gears have 4½ in. pitch and 15 in. face. All the teeth are machine-moulded. The gear ratios are thus :—

Crusher	20 ÷ 96 × 22 ÷ 96 × 98 ÷ 70 = 1 : 14·95
1st Mill	20 ÷ 96 × 22 ÷ 96 = 1 : 20·94
2nd Mill	20 ÷ 96 × 23 ÷ 96 = 1 : 20·00
3rd Mill	20 ÷ 96 × 24 ÷ 96 = 1 : 19·20

The gears are built up from hubs with arms (often called spiders), and separate rims. The shafts are supported in bronze-lined detachable bearings, which are bolted to the bedplate; the last is of riveted plate and angle design.

An interesting detail of this gear layout is that those journal bearing caps having an upward thrust are made of cast steel, whereas those having a downward thrust or reaction are made of cast iron, which indicates thoughtful design. The engine runs at 45 to 60 r.p.m.

Another compound double gear of heavy design is shown in *Fig.* 217, for three mills, 36 in. × 84 in., driven by a poppet or drop valve steam engine, 40½ in. × 60 in. Contrary to the gear of *Fig.* 216, which has involute teeth, this gear has cycloidal tooth forms. The teeth are accurately machine-moulded, and the wear on them has proved very small, as the author observed after more than 20 crops, grinding between 2000 and 3000 tons cane per day. The cycloidal tooth form is not as strong at the base as the involute form, and a fixed and exact distance from centre to centre is obligatory for good operating service. The bearings, therefore, are cast integrally with the cast iron bed plate, making a sound construction, as the bedplate girders are bored in pairs and perfect parallel alignment of the shafts at the exactly required centre distance is easily achieved. The bearings have loose cast iron bushes, lined with white metal, which can be readily replaced.

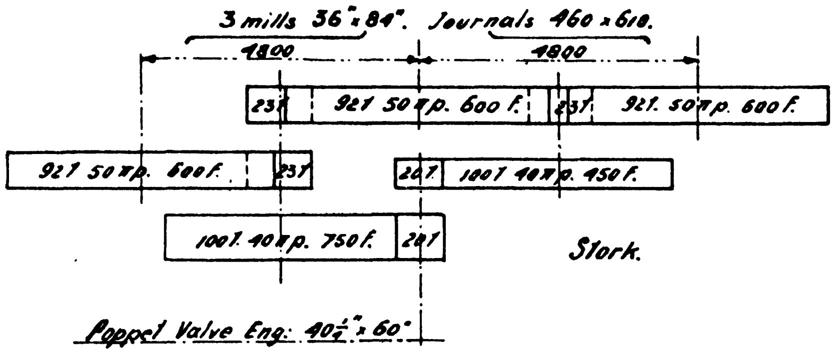


Fig. 217.—Compound Double Gear for Three Mills.

The tooth pitch is taken with these gears as a multiple of the value π, making the centre distance calculations very simple.

Taking $T_{,}$ and $T_{,,}$ as the number of teeth on the pinion and spur wheel, and M the multiple of π, the *pitch modulus*, then it follows that :—

$$Pitch = M \times \pi$$

and the centre distance C of a co-acting pair of pinion and wheel thus equals :

$$\frac{T_{,} + T_{,,}}{2} \times M \quad \dots\dots\dots\dots \quad (71)$$

The modulus system can be applied as well to the metric as to the British system of dimensions.

The gear ratio for all three mills is 1 : 20.

It is sometimes argued that the number of teeth of the spur wheels should not be a multiple of the number of pinion teeth, but an odd number. Present-day practice, nevertheless, is no longer tied to this rule and there are designing engineers who prefer even numbers of teeth, so that the same pinion teeth will always intermesh with the same spur wheel teeth.

There are two methods of gear arrangement, one having the bed plates on the same level as the mill bed plates and the other having the gear bed plates at a higher foundation level. The first is called a *high type* gear and the other a *low type* gear.

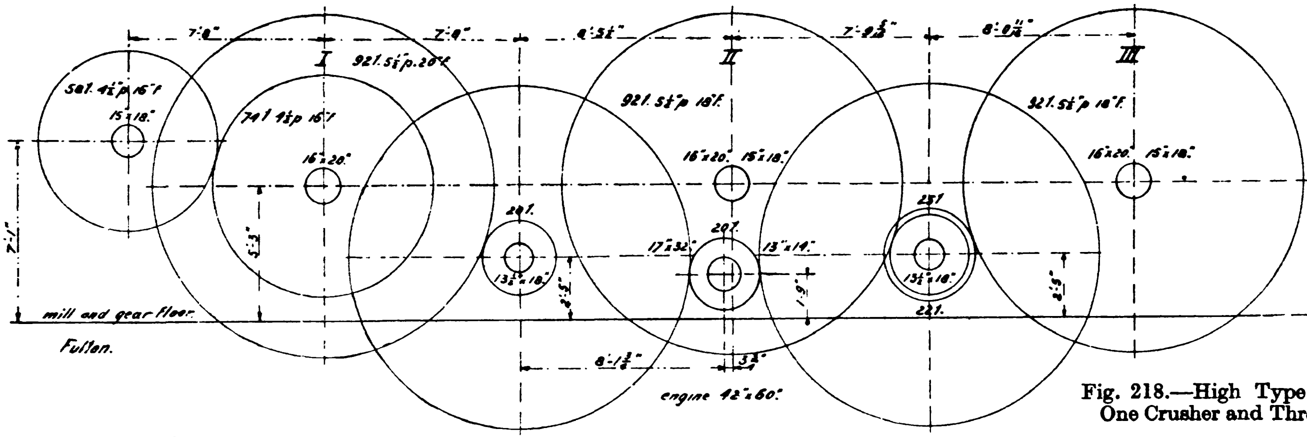
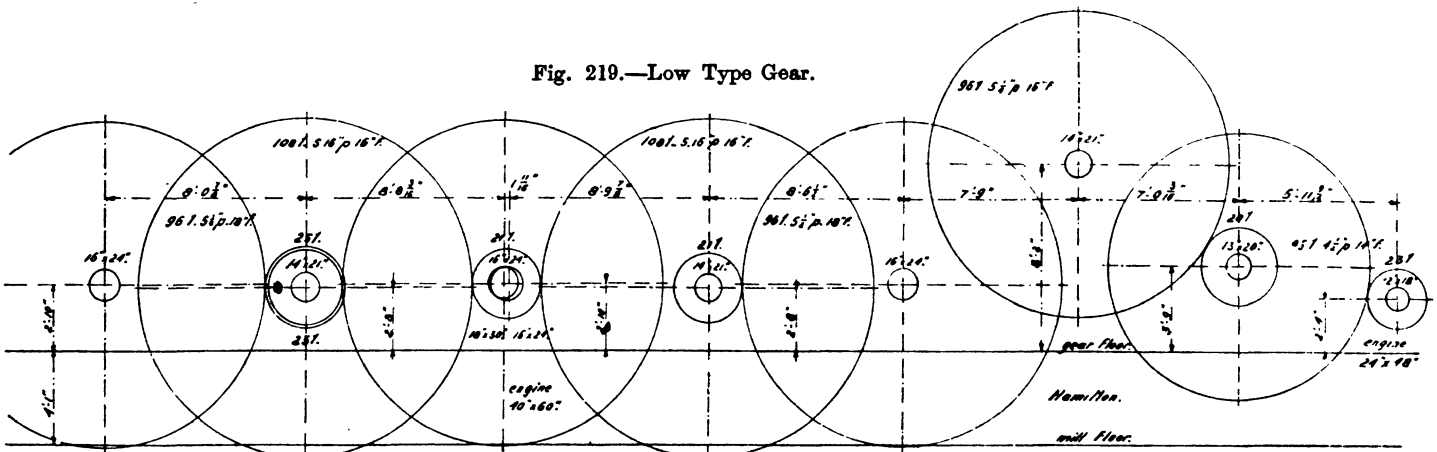


Fig. 218.—High Type Gear for One Crusher and Three Mills.

Fig. 219.—Low Type Gear.



In *Fig.* 218 a *High Type Gear* is shown for one crusher and three mills, 36 in. × 84 in., driven by one Corliss engine, 42 in. × 60 in. The gear has high pedestals for the main shafts, but as the acting forces on the main gear are considerable, vibrations may occur, and the author has replaced several broken pedestals, made of cast iron, by cast steel ones. The cost of the foundation for a high type gear will nevertheless be less.

The gear ratios are as follows :—

Crusher	1 : 16·58
1st Mill	1 : 21·16
2nd Mill	1 : 19·03
3rd Mill	1 : 16·93

All the wheels have 92 teeth of $5\frac{1}{2}$ in. pitch and only one spare need be kept, although a new gear and a worn pinion, or *vice versa*, will not always co-act well.

Sometimes the spur wheels and pinions are reversed, so that the unworn flanks come into action, and this practice is to be recommended when the gears start rattling, but are still of sufficient strength and were originally well designed.

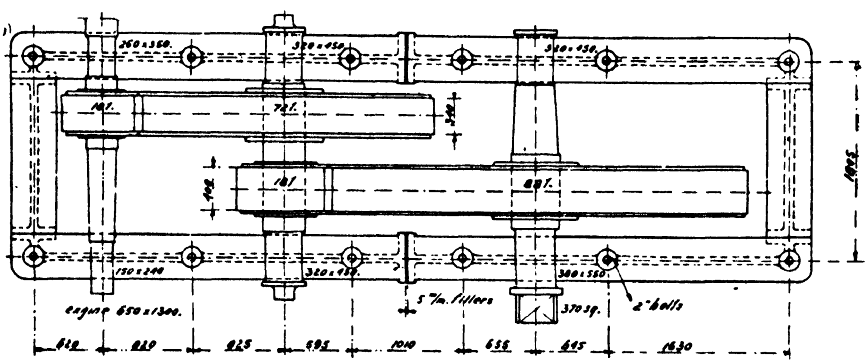


Fig. 220.—Individual Low Type Gear for 34 in. × 72 in. Mill.

The gears of *Fig.* 218 have cast steel rims with machine-moulded teeth and cast iron "spiders."

To avoid high pedestals, the bed plates of the gear of *Fig.* 219 are laid 4 ft. 1 in. higher than the mill foundation. This makes a very sturdy arrangement, as all mill gear shaft centres are not over 2 ft. 10 in. above the foundation.

The gear drives three mills, 36 in. × 84 in., by means of a Corliss engine, 40 in. × 60 in., whereas the crusher is driven independently by another Corliss engine, 24 in. × 48 in.

The intermediate gears, which have about one-fourth of the tooth pressure of those of the main gears, have $4\frac{1}{2}$ in. pitch, whereas the main gears have $5\frac{1}{2}$ in. pitch. On account of this difference some operating engineers prefer to have a smaller pitch for the intermediate gear, which will ensure smoother running. But it should not be overlooked that wear stands also in direct relation to the speed and a large pitch tooth form may have larger wearing surfaces.

All the gears have machine-moulded teeth, cast steel rims and cast iron "spiders."

The bed plate is formed from cast iron box girders with detachable bearings, having bronze bushes and oil cup lubrication.

The gear ratios can be calculated from the tooth numbers given in the drawing. The dimensions of the shaft journals are also given as in the previous figures.

With compound gears, short centre distances between the mills are obtained and the milling train will occupy less space. For 7 ft. mills this centre distance is about 16 ft.

An *Individual Gear* for a 34 in. × 72 in. mill is shown in *Fig.* 220. The previous cast iron bed plate with bolted bearings broke when a bolt fell between the running gears and a *cast steel bed plate* of H-section and having integrally cast bearings was supplied as a replacement. The bed plate girders were bored in pairs, and exact alignment of the shafts was obtained. Being an old installation, built in 1905, 5 mm. packers were provided between the parts of each bed plate girder, so that any previous misalignment between the mill and the engine shaft could be corrected.

The bearings (Fig. 220A) have cast iron removable bushings lined with white metal, and automatic lubrication is achieved by means of the well-known oil ring. The bearings are dustproof and a great reduction in lubrication expenses has been obtained, as the bearings are filled once during the crop with good medium lubricating oil. This oil is drawn off after each crop and the bearing cleaned with kerosene. The operating performance since has been excellent, with greatly reduced friction and smoother working of the gears.

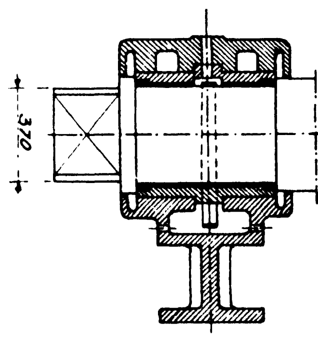


Fig. 220A.—Bearing with Ring Lubrication.

The same foundation and foundation bolts have been used. The caps of the bearings have throughgoing bolts, as many operating engineers have a dislike for studs, which are less easy to replace than a throughgoing bolt having double nuts on each side.

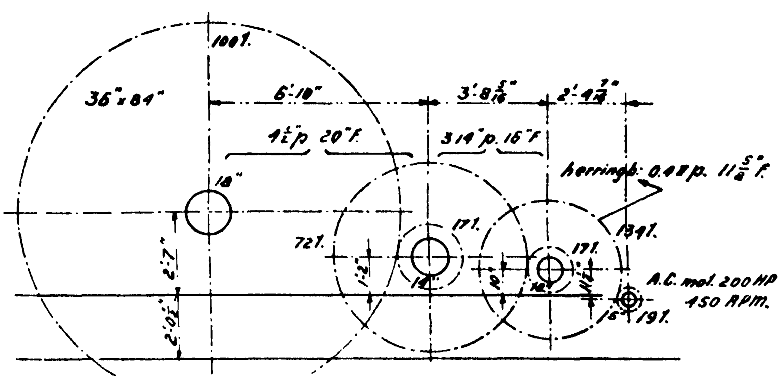


Fig. 221.—Individual Gear for an Electrically Driven Mill.

Another *Individual Gear for an Electrically Driven Mill* is drawn in *Fig.* 221, as measured by the author. The gear is of the low type, for a 36 in. × 84 in. mill, driven by a 200 h.p. alternating current electric motor, which runs at a theoretical speed of 450 r.p.m., equal to about 435 r.p.m. in practice owing to electrical slip. On account of the high ratio necessary, a *triple set of gears* is provided.

These gears are well made, entirely of cast steel and having machine-cut teeth on all gears. The first gear has double helical or V-teeth with sharp intersections at the centre according to the manufacturing method of SYKES. The gears have a total ratio of 1 : 175·7, thus the mills will obtain a peripheral speed of :— $450 \div 175{\cdot}7 \times 3 \times \pi = 24{\cdot}12$ ft. per minute.

As the individual speed of the A.C. motors cannot be varied beyond close limits for continuous service, the fact that the five mills were running at the same peripheral speed proved to be a drawback, and it was decided to make changes in the gears, so that the mill speeds should become 24·12, 26·94, 29·80, 33·56 and 35·70 ft. per minute respectively for the 1st, 2nd, 3rd, 4th and 5th mills.

Larger capacities and better extraction have been obtained through this change.

To produce exact tooth forms there exist different *generating tools*, where the teeth are produced from a cutting tool which revolves with the gear to be cut.

The most common is the worm shape cutter placed in an inclined position, so that the teeth produced will be perpendicular to the blank sides. Another method is to have the cutter reciprocating and taking the shape of a pinion. The gear wheel and the cutter revolve at the same pitched peripheral speed, while the latter is reciprocated for the cutting action.

The latter system has been employed with two cutters, each finishing half the tooth face. By a slanting movement of the gear or the cutter a herring bone or helical type is produced, having a sharp intersection midway. The tools for this purpose are made according to the patents of SYKES. Straight teeth also can be produced with these cutting tools.

Tooth hardening and tooth grinding is not yet in favour for making mill gears, such as is done for high precision gears for other purposes.

The author has seen electrically driven 7 ft. mills having only double reduction gears of the following ratios :—

$$20/24 \div 332 \times 22 \div 176 = 1 : 132{\cdot}8 \text{ to } 1 : 110{\cdot}6.$$

The A.C. motors of 250 h.p. ran at 435 r.p.m. and the gears dipped in oil baths with protective guards. The teeth of this high ratio gear showed pitting, which might be attributed to insufficient or improper lubrication, as an oil film should be maintained between the flanks of the coacting teeth, and in many high speed gears the oil is pumped under pressure through nozzles at the intermeshing point, so that a large quantity of oil covers the teeth. The large gear ratio may also be the cause of any scraping action of the pinion teeth, or too high a specific pressure may have been applied. This pitting is also noted sometimes on the teeth of the intermediate gear of steam-driven mills having machine-cut teeth, and therefore the fullest attention should be given to proper lubrication by means of oil baths, in which the gear dips, covering them with sheet iron guards, so that the gears are totally enclosed.

2.—Gear Details.

Gear details comprise bed plates, bearings, spur wheels, pinions and shafts and are already partly discussed under the previous sub-heading.

Cast iron girders of the box girder type are mostly used for bedplates, but where breakages have occurred cast steel girders might with advantage be used for replacement. The section is generally of the I type with wide flanges for easier foundry work.

Riveted girders also have been used, and in future there might be a predisposition for welded heavy steel plate construction, to which there is no practical impediment.

The bearings have removable linings and bronze is still predominant, but a good class white metal with automatic lubrication has been used by the author with equal or better results, as the friction is less.

Roller bearings are already in use for the high speed gears of electrically-driven mills and they may be used throughout for gear bearings.

Gear wheels formerly were composed of a hub piece, separate arms and a rim made up of several segments. As the bolted or keyed union cannot be made as strong as the integral material, these built-up gears have failed at higher grinding rates and under more intense grinding performance.

Moreover, the built-up rims have the inherent disadvantage that they seldom form a true circle and rattling has been known to take place under severe conditions. The true pitch at the points of coupling was difficult to maintain, and made these gears unsuited for the higher speeds prevailing in the intermediate gears.

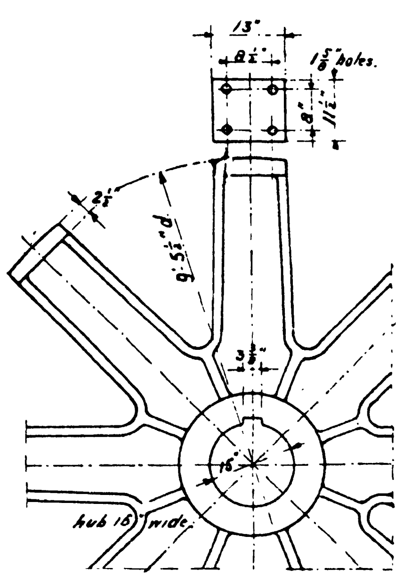


Fig. 222.—Cast Steel Spider.

This has led manufacturers to make the hub and arms in one piece and the toothed rim also. It is argued that with this two-piece arrangement, the rim or the hub and arms (spider) can be more economically replaced on wear or breakage, but it is rather doubtful whether the replacement does not involve extra expense for dismantling and shop work, which in most instances cannot be done at the sugar factory, due to the large dimensions prevailing. The author has supplied new rims with machine-cut teeth, accurately bored and drilled to true dimensions, so that they could be shrunk on the existing arms at the sugar factory; but this needs careful measuring and some skill is required to transfer these large measurements to the manufacturing shops overseas. The one-piece gear wheel is cheaper in first cost and is stronger, as there are no bolted connexions, and it should be preferred where circumstances like railway gauge limits, etc., will permit.

In *Fig.* 222 is shown a *Cast Steel Hub and Arms* for the main gear of a 36 in. × 84 in. mill, supplied by the author, to replace the previous cast iron one, which had cracked. The connexion between the eight arms of the wheel and the rim is obtained by four bolts of 1½ in. diameter for each flange. Sometimes one of these bolts on each flange is reamed to a press fit whereas the other three are loosely fitted. The material of the bolts should be of high fatigue-resisting quality, as these bolts break frequently and the possibility is not to be ignored that they may fall between the intermeshing teeth, thus causing heavy damage.

The rims are well reinforced with inside flanges, integrally cast on, to avoid deformation. This is of especial moment when the machine-cut teeth are produced from a solid blank.

Cast steel parts, as spur wheels, rims or hubs and arms, have to be carefully annealed or slowly cooled off after casting to avoid heat stresses in the material, which may lead to early rupture.

The wear on the teeth is approximately proportional to the gear ratio and it is good practice to have the pinion of a harder material than the rims ; the author has supplied several forged steel pinions, which have a higher Brinell hardness than the co-acting cast steel spur wheels. These forged steel pinions have shown a good operating performance.

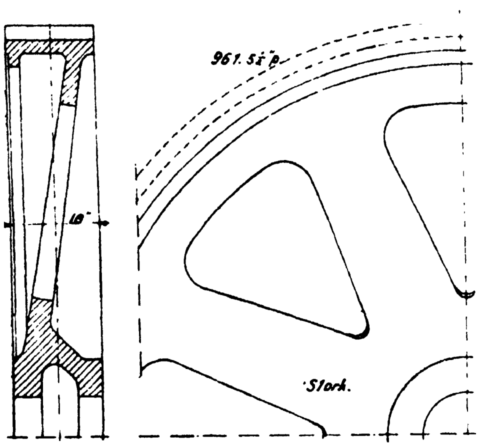


Fig. 223.—Conical Disc Type Wheel.

Of the one-piece spur wheels a novel construction is the *Conical Disc Type Wheel* shown in *Fig.* 223, of which the author has supplied many. Instead of the customary spokes or rims, the connexion between the hub and the rim is achieved by a conical plain disc, having openings for weight reduction and to allow for the contraction of the material. Any heat stresses which may prevail will be easily taken up by lateral bending of the disc.

These wheels are very strong and the author knows of only one wheel having cracked amongst many supplied, the rupture being due to too small a bore, and to applying too much heat to shrink it on the shaft. Spur wheels and pinions should have a clearance of a few thousandths of an inch on the shaft.

Wheels up to 14 ft. diameter have been furnished for actual sugar mill operation, but this large dimension is only allowable in those countries having a large railway loading clearance. Deep-well cars are needed for their transportation, or cars where the deck has been reamed, so that the wheels can be loaded in such a way, that they rest only just clear of the track. In other instances inclined supports of heavy timber have been used, the wheel lying in the diagonal of the clearance frame on top of the car platform. One needs to take care that the wheel cannot slip off during transit, as it is no easy matter to hoist it back on the car en route.

The wheel shown in *Fig.* 223 is a main gear for a 36 in. × 84 in. mill.

The author once encountered a case of *resonant or harmonic vibration* of a disc gear. At a certain speed of the engine, about 66 r.p.m., the wheel showed dangerous vibrations and every possible means of reinforcing the pedestals and bearings failed to give the desired result. By speeding up the engine to about 72 to 74 r.p.m. the vibrations ceased completely and as the driving engine had been arranged for 80 r.p.m. this speed could be maintained. As a curiosity, it should be mentioned that the wheel in question was the slowly moving main gear.

To obtain a variation in the number of resonant vibrations, fish plates have been welded to the disc and have proved a good remedy. This welding, nevertheless, has to be done according to a pre-determined cycle, as otherwise the wheel will warp.

This phenomenon of resonant vibrations occurs from time to time also with spoked gears and due to it many a "spider" has cracked. But a distinction should be drawn between resonant vibrations and those due to improperly formed teeth.

The standard wheel with arms, when cast in one piece, has to have a *Split Hub* to avoid heat stresses, as is shown in *Fig.* 224. The splits are machined, and straight machined fillers *a* or packers are inserted before the heavy shrunk rings are put on, thus restoring a stress-free unit of hub and arms.

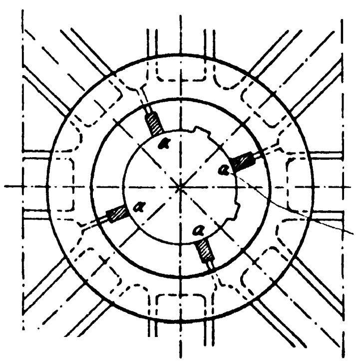


Fig. 224.—Split Hub.

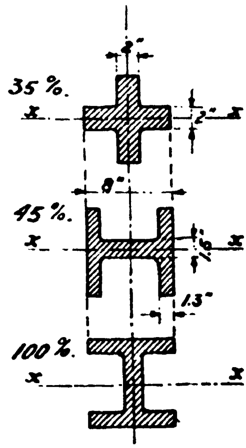


Fig. 225.—Three Types of Arm Sections.

There is a divergence of view amongst manufacturers of gears as to the *arm section*. The ease of moulding or the attachment of the wheel arms to the rims also plays an important part in these criteria. For information's sake, the author has drawn three types of arm sections in *Fig.* 225, representing the commonly used designs, comprising the cross type, the H type and the large flanged I type. The bending axis for all three sections is indicated by the lines x—x, and the area of all three amounts to about 28 sq. in. each. The section moduli or the moments of resistance for the three types are as follows :

Cross type	22·4 cub. in.
H-type	28 ,,
I-type	64 ,,

all referred to the axis x—x.

This will show that for equal area of the arm section, the proportion in strength is as 35 : 45 : 100. The I-type, having the flanges parallel to the bending axis, is thus the strongest.

The *Prevailing Bearing Reactions* have to be ascertained at the outset when designing shafts, journals and bed plates. In particular, where upward thrusts exist, material of sufficient strength has to be used, as in bearing caps and bearing bolts. In *Fig.* 226 is shown such a diagram of reactions for a gear of 36 in. × 84 in. mill. The Corliss engine, 30 in. × 60 in., drives two mills, the other mill drive being the opposite of the one shown. For the acting forces a convenient scale of a certain fraction of an inch for each 1000 lbs., to correspond with the size of the drawing, has to be established.

As the acting forces in *Fig.* 226 are drawn to scale, it will be seen that those forces on the intermediate gear are relatively small compared with those on the main gear. The engine produces a couple of forces *a* for each mill, which is reacted by the intermediate gear in a couple of forces *b*. The main pinion has a couple *c*, reacted by the couple of forces *d* of the main gear.

On the intermediate shaft act the forces $P_{\prime}$ and $P_{\prime\prime}$ which combined form the resultant force *R*. The two intermediate bearings have to withstand this force plus an excess for abnormal conditions, when the mill is taking more power than the engine produces normally and thus the inertia force of the flywheel will be partly consumed.

The tooth forces can be derived from the torsional momentum of the engine, as explained with *Fig.* 182 (page 174).

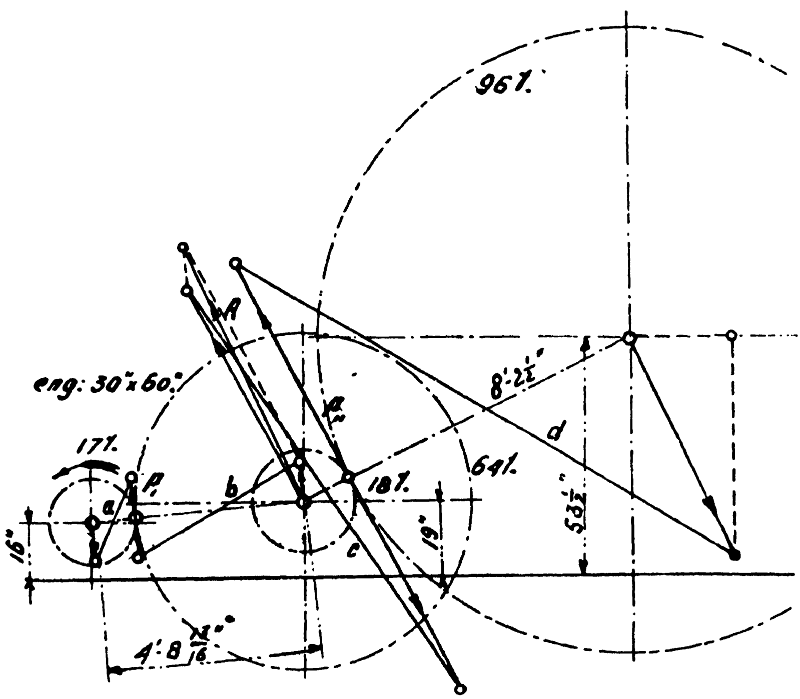


Fig. 226.—Diagram of Reactions for a Gear of a 36 in. × 84 in. Mill.

In *Fig.* 226, these tooth forces are taken to act perpendicularly on the centre line between both shafts, but with an involute tooth form a deviation of from 15 to 20° has to be taken into consideration.

The shafts are subject to combined torsional and bending stresses, and have to be calculated accordingly. From experience it is known that the three bearing shafts of compound gears are subject to breakage and they should be carefully calculated, with an allowance for the fact that the three bearings are each seldom carrying the corresponding share of the load.

In *Fig.* 227 is given the *Motor Shaft* of the gear shown in *Fig.* 221, the pinion forming an integral part of the shaft on account of the small pinion diameter. Both bearings are lined with white metal, having automatic oil ring lubrication. The electric motor is connected by a flexible coupling to this shaft and although these couplings will take up small errors in alignment, it has to be recollected that they will last longer, the better the alignment is. Misalignment will cause movement of the coupling parts and as this is done at a high frequency, the result is bound to be abnormal wear.

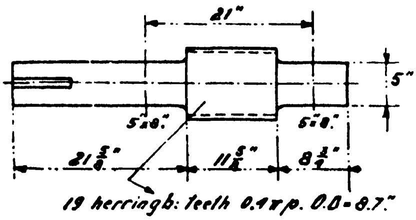


Fig. 227.—Motor Shaft for Gear.

Cycloidal Tooth Forms are in use in mill gear design, having good wearing qualities, but they are not the strongest at the tooth base and the gears have to operate at fixed centre distances. The production of cycloidal teeth by tools is less feasible than the machining of involute teeth.

Present day machine-cut teeth, therefore, are of *involute form* exclusively and design is tending to make the producing circle approach as closely as possible to the bottom circle of the teeth, so that these are not undercut and a tooth form will result that is strongest at its base. In *Fig.* 228 are shown different tooth forms of up-to-date mill gears. The bottom form *a* is of the Sykes generated type.

In the upper part is shown the average wear that will take place, approaching the cycloidal tooth form. With worn teeth the gear will become noisy and vibrations occur, which are detrimental to the life of the equipment. Pinions and gears can be reversed, as already mentioned, so that the unworn flanks can co-act. But it needs to be ascertained if the keys can be driven in from the other side, as the keyways reverse with the pinion or gear.

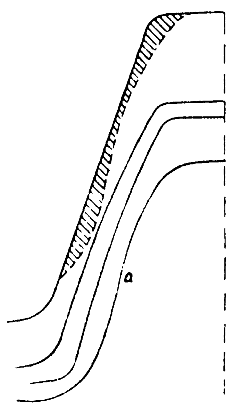


Fig. 228.—Tooth Forms of Up-to-date Mill Gears.

A well designed gear should have more than one tooth carrying the load; this can be ascertained by placing strips of writing paper on the bearing side of the teeth and turning the wheel so that the strips of paper are nipped at the intermeshing locus between the co-acting teeth. The number of paper strips grasped is obviously the number of bearing teeth. Gears with only one tooth bearing are liable to be noisy.

When new pinions are ordered, they should be designed according to the existing tooth form of the spur wheel teeth.

The back lash of machine-cut teeth is only a few hundredths of an inch and any foreign body or dirt introduced will do great harm to the teeth and may give rise to tooth lock.

3.—General Gear Data.

A few gear ratios are cited below :—

Crusher	36 in. × 84 in.	1 : 14·95	down to	1 : 25
Mill	36 in. × 84 in.	1 : 15·70	,,	1 : 30·79
Crusher	34 in. × 78 in.	1 : 19·4	,,	1 : 24
Mill	34 in. × 78 in.	1 : 15·7	,,	1 : 27·45
Crusher	32 in. × 72 in.	1 : 15·7	,,	1 : 25
Mill	32 in. × 72 in.	1 : 18	,,	1 : 25
Crusher	30 in. × 60 in.	1 : 14·9	,,	1 : 23
Mill	30 in. × 60 in.	1 : 13·7	,,	1 : 30

These data are from existing mills and crushers with double gears. In Java there are several installations having triple gears driven by higher speed steam engines and with ratios ranging between 1 : 62·5 and 1 : 75, crushers and mill gears having the same ratios.

For electrically driven mills different ratios are given under the previous sub-heading.

The *approximate weights* of gears and gear wheels are as compiled in the following table :—

		Long Tons (approx.)
Gear for 30 in. × 60 in. mill	54,000 lbs.	24·0
,, 32 in. × 72 in. mill	65,000 lbs.	29·0
,, 34 in. × 78 in. mill	108,000 lbs.	48·0
,, 36 in. × 84 in. mill	120,000 lbs.	53·5

These weights are for low type gears ; for high type the weight has to be increased by about 5 per cent.

Gear for 2 mills 32 in. × 72 in. ..	120,000 lbs.	53·5 tons
,, 2 mills 34 in. × 78 in. ..	180,000 lbs.	80·0 ,,
,, crusher and two mills 32 in. × 72 in.	170,000 lbs.	76·0 ,,
Cast steel bedplate as per *Fig.* 220..	32,000 lbs.	14·3 ,,
Cast steel spider for 36 in. × 84 in. main gear	8,800 lbs.	4·0 ,,
One-piece spur wheel, 91 teeth, 4½ in. pitch, 18 in. face	20,000 lbs.	9·0 ,,
Pinion, 21 teeth, for same	4,600 lbs.	2·0 ,,
Spur wheel with spider, 90 teeth, 5 in. pitch, 18 in. face	24,000 lbs.	10·7 ,,
Rim for same	16,000 lbs.	7·0 ,,
Disc wheel, 100 teeth, 4½ in. pitch, 20 in. face	19,000 lbs.	8·5 ,,
Do., 96 teeth, 5 in. pitch, 18 in. face	26,000 lbs.	11·6 ,,

Standards for the number of teeth and face width do not exist, as there will always be individual requirements ; but a given factory should adhere to standard circular pitch for first and second gears, the latter of larger dimensions. This will reduce the number of spares to be kept or, as the case will be in large factories, the spares will fit in different places. Also, the shaft bores and keyways for each factory should not show unpractical differences.

Below is given a list of all gears in a large Cuban cane sugar factory which has been gradually enlarged with new or existing equipment from different manufacturers. Only the circular pitch, the number of teeth and the face width are given, but there is also a difference in bores. This factory has 15 mills and 5 crushers, arranged in three tandems.

	Number of Teeth.	Circular Pitch in inches.	Face in inches.
1 pinion	19	4·500	17
1 pinion	21	4·500	15
1 pinion	23	4·500	16
2 wheels	55	4·500	16
1 wheel	74	4·500	17
2 wheels	77	4·500	16
1 wheel	85	4·500	15
1 wheel	90	4·500	16
1 pinion	20	4·943	15·750
1 pinion	20	4·943	21·625
1 wheel	100	4·943	15·750
1 wheel	100	4·943	21·625
2 pinions	23	5·000	16
2 wheels	90	5·000	16
1 pinion	17	5·125	18
1 wheel	78	5·125	18
2 pinions	20	5·500	18
4 pinions	20	5·500	19
2 pinions	22	5·500	19
4 pinions	23	5·500	19
4 pinions	25	5·500	19
1 wheel	59	5·500	16
1 wheel	82	5·500	16
13 wheels	92	5·500	18
1 wheel	92	5·500	19
2 wheels	92	5·500	20
1 pinion	21	6·000	18
1 pinion	23	6·000	18
2 wheels	88	6·000	18
3 pinions	23	6·062	19·750
3 wheels	92	6·062	19·750

CHAPTER IX.

MILL DRIVING ENGINES.

At the present day cane sugar mills are driven either by steam or electricity. Water-driven plants are fast disappearing, as the manufacturing process of sugar calls for a large amount of heating steam and therefore a steam drive is the logical consequence in sugar factories. Moreover, fuel is available in the bagasse left over from the milling performance.

However, to centralize the power input of the factory, electrification has definite merits, as this centralization uses large size prime movers developing higher efficiency and does not need steam and exhaust pipe lines to each power unit. These prime movers for generating the electrical current are also steam-driven and will be discussed later in Chapter XXXIV.

1.—Power Generated by Steam.

To Great Britain belongs the cradle of the steam engine, and to the inventive mind of James Watt we owe the use of steam above the atmospheric pressure for power generation and the expansion performance for higher efficiency.

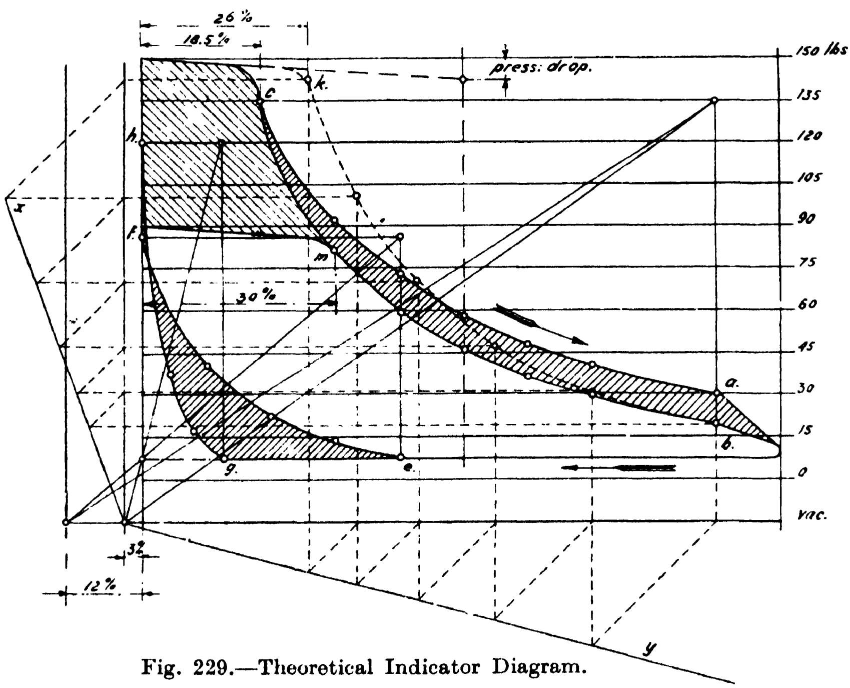


Fig. 229.—Theoretical Indicator Diagram.

The original beam engine has been replaced by the direct drive, where the piston force, acting reciprocally is transformed by means of a crosshead, connecting rod and crank into a rotatory movement.

The basic idea of the steam performance in the cylinder can be gathered from the *Theoretical Indicator Diagram*, as shown in *Fig.* 229. Let it be assumed that the steam enters the cylinder at 150 lbs. per sq. in. gauge pressure

(164·7 lbs. per sq. in. absolute pressure). Through the steam ports and the steam distribution channels a drop in pressure generally takes place and at the moment when the steam is cut off at *c* at 18·5 per cent. of the engine stroke, the pressure has dropped to 135 lbs./sq. in. The steam now expands and in the diagram the isotherm is assumed to represent the expansion performance, the product of volume and pressure remaining equal during the whole of that performance. At *a* the exhaust port is opened, whereupon the prevailing pressure drops quickly until a back pressure of about 7·5 lbs. is assumed. At the return stroke, the exhaust port remains open until *e*, where it is closed and compression of the trapped steam will thus take place. Assuming again the compression to be isothermic, the steam will be compressed until *f*, when the steam distribution gives early admission just before the piston or crank dead centre has been reached, and the pressure quickly rises to 150 lbs., to start the cycle again.

The diagram *c-a-e-f* has been drawn with an assumed dead space of 12 per cent., this being the maximum for regular slide valve engines; and to show the influence of a smaller dead space, the diagram *c-b-g-h* has been drawn for 3 per cent. dead space, as obtained with Corliss engines. It is seen here that the pressure drops more quickly, so the area *c-a-b* is the power loss. On the other hand, the compression can be much smaller and there is a gain in power, represented by the area *e-f-g*, the small area *f-h* being neglected. As the expansion loss in power will be almost offset by the gain in compression power, the total diagram areas may be considered as equal, and little difference in power output is to be expected from any different dead space of the cylinders; but it should not be overlooked that the larger dead space requires a larger steam consumption. The compression will offset this larger consumption only to a certain extent and mill engines with a small dead space should be preferred. The piston clearance is generally over ¼ in. to ensure safe operation of the engine and a large dead space must not be considered a protection against water hammer; water relief valves have to be provided on each cylinder end.

It is sometimes argued that in a cane sugar factory live steam consumption to be used for power generation has no direct bearing upon the total heat efficiency of the factory, as all the exhaust steam, containing the largest amount of latent heat, has to be used for heating purposes. This theory, however, should be accepted with great care, as it has happened before now that the boiling house could not efficiently manage the large amount of exhaust steam available and it has had to be blown off through the roof, proving a considerable heat loss. Moreover, as a consequence, the exhaust or back steam pressure is raised unnecessarily and will demand a higher steam consumption of the prime movers, so that a vicious circle is thus established.

It should also be recollected that in our modern milling plants more power is required for more intense milling, mechanical unloading of the cane, shredders or revolving knives, and improvements in other departments. Any reserve in live steam capacity will thus result in an asset in the total steam or heat balance of the factory, whereas a shortage of live steam, caused by an excess of exhaust steam, requires additional fuel besides the available bagasse, and generally becomes a costly liability.

Moreover, in *Fig.* 229 the expansion line *k-b* is drawn for superheated steam, assumed to be an adiabatic, constructed by means of the auxiliary lines *x* and *y*. The adiabatic drops more quickly than the isotherm *c-b* and a larger steam admission of 26 per cent. is required to reach the same pressure *b* at the moment when the exhaust port opens. Apparently more superheated steam is needed, but it will be erroneous to assume from this larger steam admission a larger steam consumption, as superheated steam has a much larger specific volume than saturated steam and, moreover, the power output is increased by the area *c-b-k*.

Superheated steam of 105 lbs. gauge pressure and 150°F. superheat, i.e. the temperature above the saturation temperature, has a volume of 5·23 cub. ft. per lb., whereas saturated steam of the same gauge pressure has only 4·23 cub. ft. per lb. volume. To produce 1 lb. of this superheated steam a total amount of 1266 B.T.U.'s is required, as against 1187 B.T.U.'s for saturated steam. The volume increase of (5·23 — 4·23) ÷ 4·23 = 24 per cent. therefore only requires an excess of 7 per cent. in heat value.

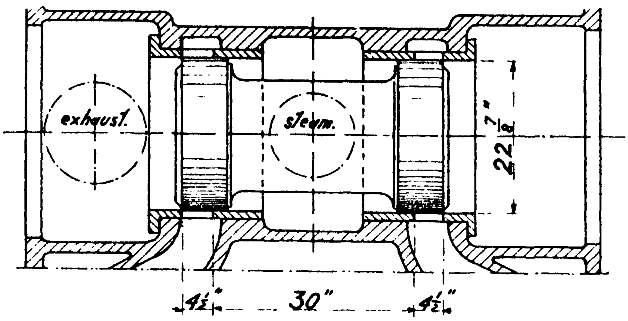


Fig. 230.—Cylindrical Slide or Piston Valve.

It is, therefore, advantageous to use superheated steam, but for sugar factory performance a superheat higher than 50 to 100°F. is not generally employed. High superheat requires special stuffing boxes, careful lubrication and valve seats of special alloys, and although these conditions can be met with in up-to-date factories, some consideration has to be given to those countries where unskilled labour handles the machinery, or existing equipment has to be used.

With high superheat of the live steam, the exhaust steam may still be slightly superheated, but this has no effect on the performance of heat transmission in the heating vessels of the boiling station, as formerly was assumed.

The danger of water hammer is not present with superheated steam.

Finally, the difference between high and low steam pressures is shown. The low pressure steam has a larger specific volume and in the diagram a 30 per cent. admission *f-m* at 90 lbs. gauge pressure corresponds with an 18·5 per cent. admission at 150 lbs. There is obviously no greater steam consumption, but the amount in power output, represented by the area *f-m-c-h*, is lost. Higher steam pressure, accordingly, will give a higher power output for the same weight of steam.

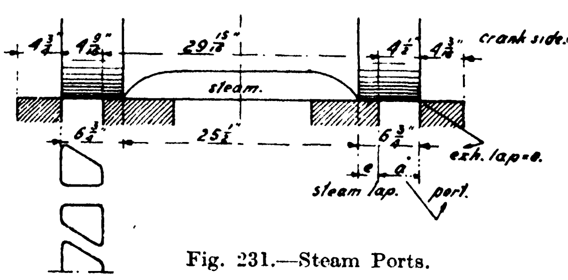


Fig. 231.—Steam Ports.

For economical operation, the engines should be designed in such a way that, under normal load conditions, the end of the expansion at *a* or *b* will approximate to the back steam pressure, thus to about 5 to 10 lbs. above, so as to obtain a quick discharge when the exhaust port opens.

This theoretical diagram has value for designing purposes, but in practice a true isothermic or a pure adiabatic expansion is not obtained, but will generally lie between or below these curves due to moist steam, heat radiation through the cylinder walls and condensation on cooled cylinder surfaces.

2.—Slide Valve Mill Engines.

The flat slide valve, as invented by WATT, was improved by his assistant MURDOCK by giving it a cylindrical shape. Flat slide valves are still in use in many mill engines, but the more modern ones, especially those of larger size, have piston or cylindrical ones. Flat slide valves can be balanced by a pressure relieving cap or a balancing device as shown in *Fig.* 53. The latter will take up any wear, which is only possible with the cap type by reducing the cap height. The piston valves are nearly balanced and will remain in this condition for a long time. Higher steam pressures or superheated steam are well handled by this type of steam distribution.

In *Fig.* 230 is shown such a *Piston Valve* for a large 44 in. × 60 in. mill engine, operating within cast iron bushings which contain the steam ports. To relieve the valve rod stuffing boxes from the live steam pressure this live steam is admitted centrally and the exhaust is released at the outer ends, so that the stuffing boxes have to act only against a low exhaust pressure. The valve rod should be supported in front and rear covers.

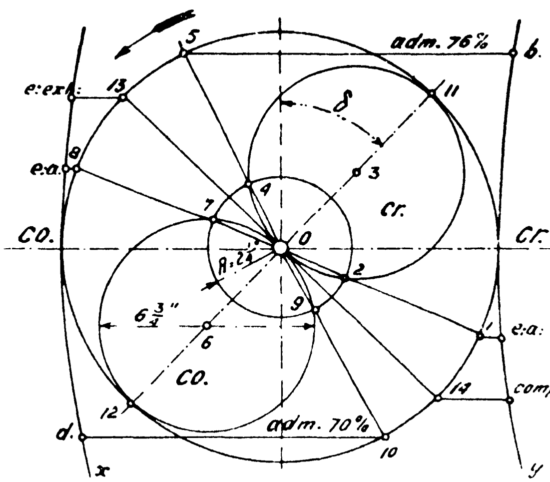


Fig. 232.—Zeuner Diagram for Slide Valves.

The piston valves are provided with self-expanding rings, and rings of small section as indicated in *Fig.* 56 will achieve perfect steam tightness under reduced friction.

In *Fig.* 231 are shown the *Steam Ports* of this valve. The steam laps are on the inside edges, whereas the exhaust laps are on the outside, or, as is the case with back pressure engines, there are no exhaust laps or they may become even negative. The ports have inclined slots, so that the self-expanding rings will easily slip over them. Sometimes drilled holes are provided in the bushings instead of slots. The holes have to be numerous enough to give sufficient free passage for the steam, so that wiredrawing will not occur beyond practical limits.

To check a given steam distribution through a slide valve, the *Zeuner Diagram*, as shown in *Fig.* 232, is very useful. The eccentricity is equal to the sum of the steam port a plus the steam lap e as a minimum and a circle 1-11-5-13 is drawn with a radius of 6¾ in. Two arcs x and y are drawn with a radius of 33¾ in., being five times the eccentricity. This is done to represent the limited length of the connecting rod, having a ratio 1 : 5 with the crank radius.

Point 1 is for early admission, being about 1 to 2 per cent. of the stroke. The line 1-0 is now drawn and consecutively the circle 2-4-7-9, having the steam lap $e = 2\frac{1}{2}$ in. as a radius. The intersection 2 will give the exact location of the centre point 3 of a circle 2-11-4, having a diameter of $6\frac{3}{4}$ in. Through the other intersection is drawn the line 0-4-5 and a steam admission or cut-off of 76 per cent. can be measured from the diagram. The advancing angle δ of the eccentric is now also obtained and it runs $90° + \delta$ ahead of the crank for outside charging valves. For inside charging valves, it is diametrically opposed, thus running $90° - \delta$ behind the crank in the direction of rotation.

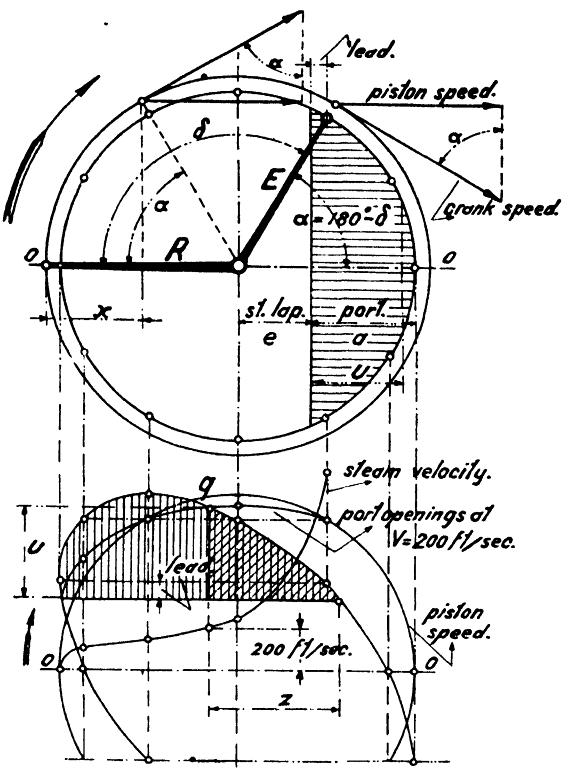


Fig. 233.—Müller Valve Diagram.

As there is no exhaust lap, the line 13-0-14 is drawn at right angles to 11-0-12 and point 13 is for early exhaust, measured on the horizontal line *e* exh.— 13 in hundredths of the diameter *CO-CR*. The compression is similarly found at 14. For the cover or head side of the engine the circle 0-7-12-9 gives the corresponding steam periods and early exhaust is to be found in point 14 and the compression in point 13.

By decreasing the angle δ, early admission is decreased, the cut-off increased, and the early exhaust and compression decreased or *vice versa*.

In *Fig.* 233 is shown at the top the widely used *Müller Diagram.* The arcs for limited connecting rod length are left out, but can be drawn similarly as in *Fig.* 232. The steam periods can be easily measured from the vertical lines at the intersections with the circle, having the radius *E*. The stroke line *CO-CR* of *Fig.* 232 now has to be drawn on this radius *E*.

Any variation in one of the four steam periods will cause a variation in the others, as already explained, and therefore two separate valves are sometimes used, one under governor control for early admission and cut-off and the other for exhaust and compression, the latter having a fixed eccentric.

Taking *R* the crank radius and *E* the eccentricity, in any position *x* of the piston or of the valve, there remains the equation :—

$$x = R \text{ (or } E) \times (1 - \cos \alpha).$$

The eccentric E is advanced by the angle δ with the crank in *the direction of rotation*, and by plotting the valve travels corresponding to the piston positions, the elliptic diagram below is easily drawn.

Furthermore, the eccentricity has to be at least, and generally is, equal to the sum of steam port and steam lap, or :—

$$E = e + a$$

and the cutting edge of the steam port can be drawn in both the circular diagrams as the ellipse.

Through the inertia of the flywheel, the crank speed V is considered as being constant. The piston speed C, derived therefrom, has to be :—

$$C = V \times \sin \alpha.$$

It is obvious that at dead centres, crank and piston speeds equal 0, but the valve speed at this moment will amount to :—

$$C_{valve} = V_{ecc.} \times \sin \delta.$$

where δ is the angular advance.

It is also clear, that when the piston leaves the dead centre, the piston speed is increasing, whereas the valve speed is decreasing. At a certain point q the steam velocity will have reached such a height that *wiredrawing* will be the result. The port area F will give the corresponding steam velocity w by the following formula :—

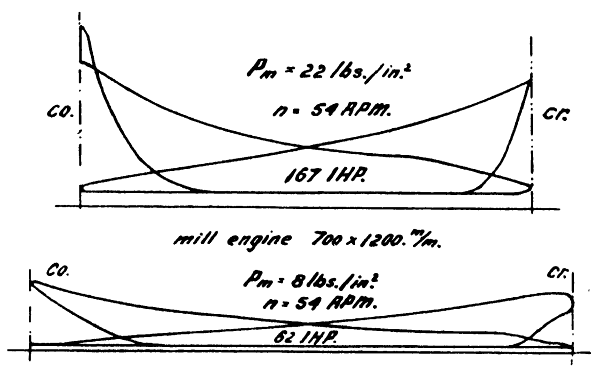


Fig. 234.—Actual Indicator Cards of a Piston Valve Engine.

$$A \times V \sin \alpha = F \times w$$

where A is the piston area. As the valve is travelling, F is not a *constant* figure, but may decrease to the amount :—

$$F_{,} = F \times \frac{E\ (1 - \cos \alpha)}{a}$$

when $E\ (1 - \cos \alpha) < a$.

The steam velocity curve can now be drawn, and it will be seen that at the later stage of admission very high steam velocities are reached.

From practical observations it has been deduced that with customary steam pressures at sugar mills, a steam velocity of 200 *feet per second* will be the limit, beyond which wiredrawing will take place.

The 200 ft. curve is also drawn and the point q is determined. Over a length z of the piston stroke, or the corresponding part u of the valve travel, there will be wiredrawing.

Many piston valve engines have a throttle-governing device and in such cases a considerable part of the steam energy is wasted for power production at the expense of steam consumption. Heat energy is still available in the exhaust of these engines, but big steam consumers are liable to upset the steam balance of a sugar factory, as mentioned before.

In *Fig.* 234 are shown the actual *Indicator Cards of a Piston Valve Engine*, 700 mm. dia. by 1200 mm. stroke. For the full load diagram the compression on the cover or head side is too large, and the advancing angle of the eccentric should be decreased. Wiredrawing is the reason why the cut-off is difficult to determine. The empty load diagrams of the engine, gear and mill are shown in the lower part. The latter cards have been taken with a smaller reduction gear on the indicator.

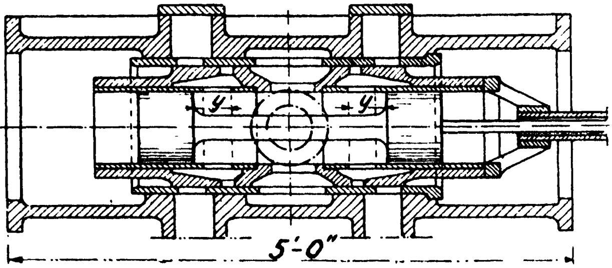


Fig. 235.—Interjacent Double Valves.

To improve the performance in respect to steam distribution, the flat slide valve has been arranged in such a way that two coacting valves, one sliding on top of the other, have been designed in Germany by MEYER. The main valve, being the lower one, controls early admission, early exhaust and compression, whereas the top one is only for cut-off. These Meyer valves are still in use on many a mill engine and the adjustment in cut-off is done by hand.

The same principle but with governor-controlled cut-off was devised in America by RIDER, but has been brought to greater perfection by European designers who have replaced the plain unbalanced valves by piston valves.

A set of such double valves for fixed cut-off is shown in *Fig.* 235. The main valve, viz. the outer one, accomplishes the early admission, the exhaust and the compression, whereas the inner or expansion valve *only* accomplishes the cut-off of the steam and sets therefore the degree of expansion.

The design of *Fig.* 235 is for a mill engine, 30 in × 48 in., driving a crusher and first mill of a cane grinding tandem. Both valves are operated by separate eccentrics, the main valve having the same angular advance as would be the case with a single piston valve and giving a large cut-off.

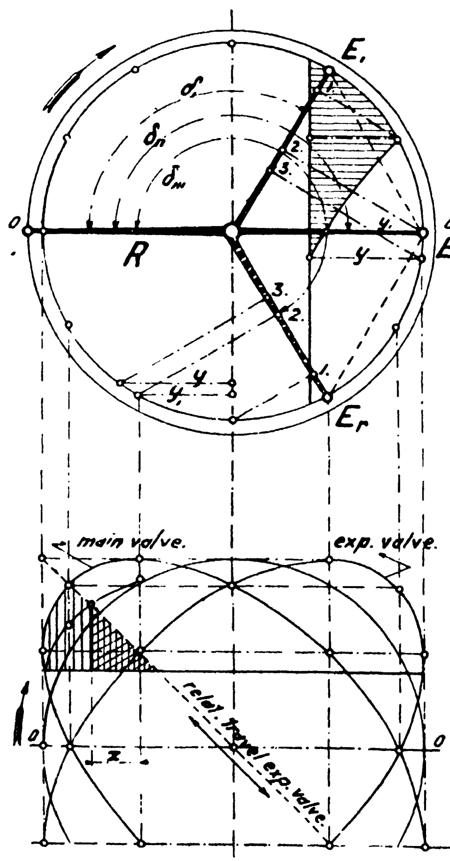


Fig. 236.—Double Valve Diagrams.

The eccentrics are mounted on the crank shaft, as shown in the top part of *Fig.* 236; the main valve eccentric at $E_{,}$ and the expansion eccentric at $E_{,,}$. For *reversible* engines, as most sugar mill engines are, the eccentric $E_{,,}$ is generally advanced 180° on the crank. The main valve has a *double* set of eccentrics with reversing link, but the expansion valve through its *neutral* position needs only *one* eccentric. If this condition is not fulfilled, the expansion valve also has to have two eccentrics and, moreover, a reversing link.

The expansion valve rod of *Fig.* 235 is located within the tubular main valve rod. This is not a necessity, as the expansion valve can be operated from the rear and the main valve from the crank side of the steam chest or *vice versa*.

From *Fig.* 236 it is seen that, the main valve still being on its outward stroke, the expansion valve has already started the return stroke and a high *relative* velocity will prevail at the moment when the expansion valve cuts off the steam admission.

The moment where the steam will be cut off is determined by the distance y of the cutting edges of expansion and main valves.

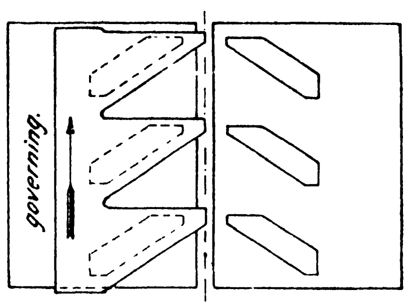


Fig. 237.—Rider Cut-off Valves.

The valve diagram in *Fig.* 236 is drawn identically as in *Fig.* 233. The main valve travel is shown by its corresponding ellipse and the *absolute* travel of the expansion valve also. As this absolute travel of the latter is of no importance for our purpose, but rather the *relative* movement, i.e., the difference between both, it will be easily seen that the relative eccentricity E_r formed by the parallelogram $E_{,}$—$E_{,,}$—E_r—O will give the difference between the valve travels of $E_{,}$ and $E_{,,}$. In the diagram the relative valve travel of the expansion valve takes place on the dotted diagonal line, derived the same way as both ellipses.

Considering the diameter E_r as the piston stroke, the point 3, being the point of cut-off, will give the distance y, and point 2 yields the distance $y_{,}$, whereas at 1, $y = 0$. With no admission y will be negative.

The distance z where wiredrawing will take place is considerably reduced as compared with *Fig.* 233.

Instead of having a fixed point of steam admission, the cut-off might be conveniently *variable under the influence of the governor.* For this purpose RIDER has designed valves with inclined admission channels for the expansion valve, as shown in *Fig.* 237, where the developed surfaces are shown.

The writer has rebuilt several mill engines with these Rider cut-off valves, but great care has to be taken to give proper design, as otherwise heavy wiredrawing will occur and the advantages of these valves be greatly nullified.

In *Fig.* 238 is shown a flat slide valve, rebuilt into a set of Rider cut-off valves with automatic expansion. The main valve is of the balanced flat port type, whereas the expansion valve is adjacently located in the same valve hood. It should be recollected that the valve hood does not participate in any movement and therefore only *absolute* velocities prevail. The relative eccentricity E_r does not apply in this case.

In *Fig.* 239 are shown the *Indicator Cards* of a mill engine, 28 in. diam. by 48 in. stroke, with *Adjacent Double Valves* as shown in *Fig.* 238, and they reveal that wiredrawing on the head or cover side is heavier than on the crank side, but the steam distribution is better than that shown in *Fig.* 234. The power developed by the engine when the diagrams were taken had been low for its size, and the expansion line on the crank side drops below the exhaust line and thus a negative power performance is indicated by the loop in this diagram.

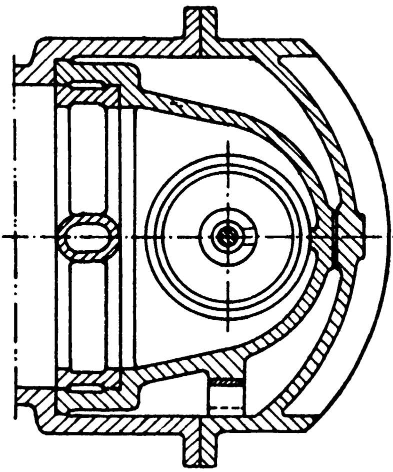


Fig. 238.—Adjacent Double Valves.

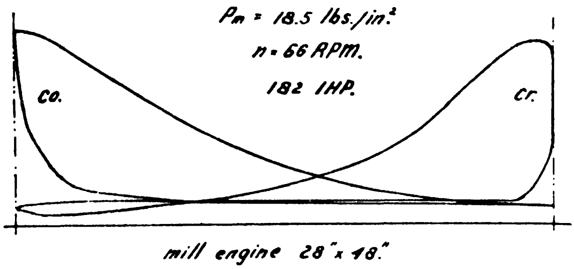


Fig. 239.—Indicator Cards of a Rider Cut-off Mill Engine.

The engine of which the valve chest is shown in *Fig.* 235 has been provided with Rider cut-off valves and the results of the design by the writer are revealed in the indicator cards of Fig. 240. The boiler pressure being 115 lbs., there was a considerable drop in pressure through the narrow steam line to this engine. Although the engine had been designed originally for only 43 r.p.m., speeds up to 65 r.p.m. were obtained with a good operating standard of the valves.

The lower sketch of *Fig.* 240 shows an indicator card of the same engine but at a lower steam pressure. The developed horse-power is nearly the same and the wide range of cut-off is performed without difficulty. Both indicator cards were taken by the customer's staff.

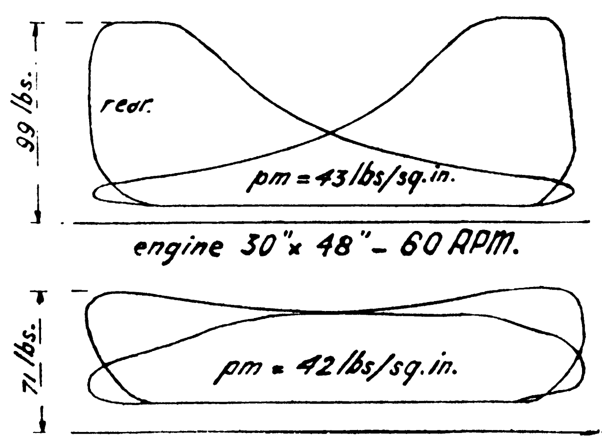
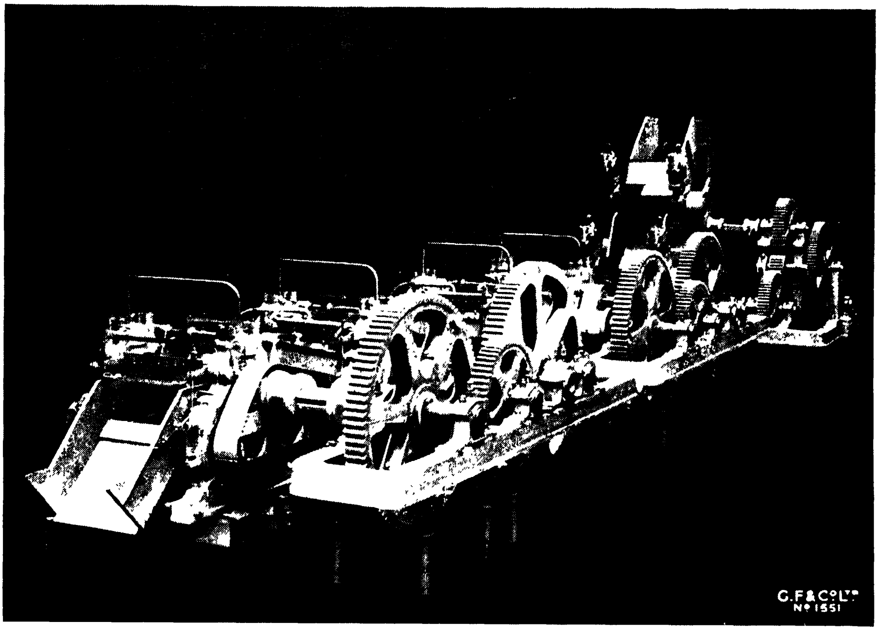
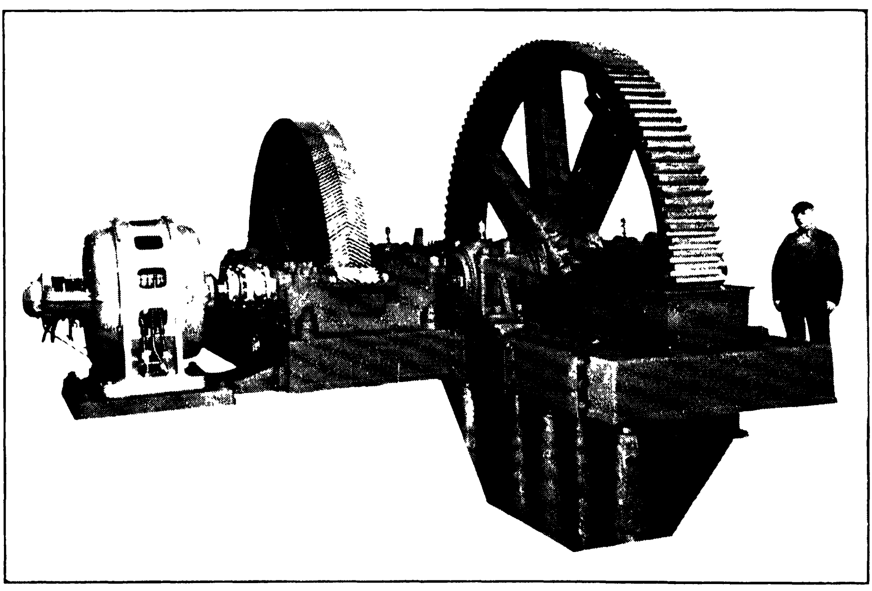


Fig. 240.—Full Load Diagrams with Rider Cut-off Valves.

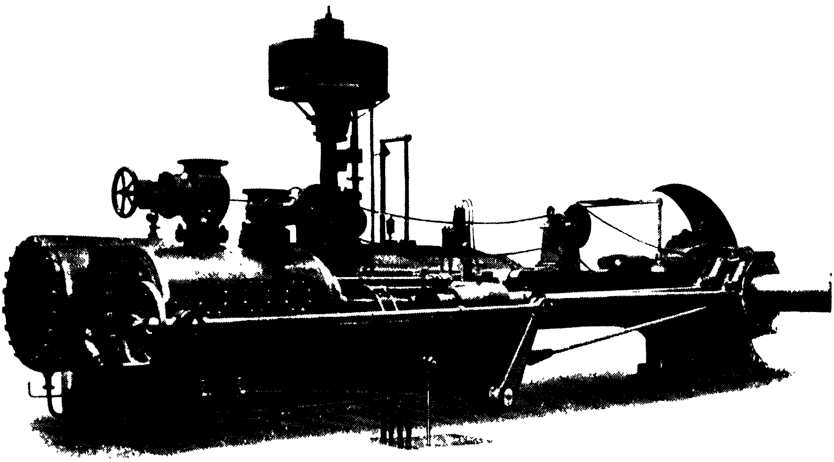


Complete Gear Assembly of a 14-Roller Milling Plant, 26 in. x 48 in.
(*Geo. Fletcher & Co., Ltd.*)

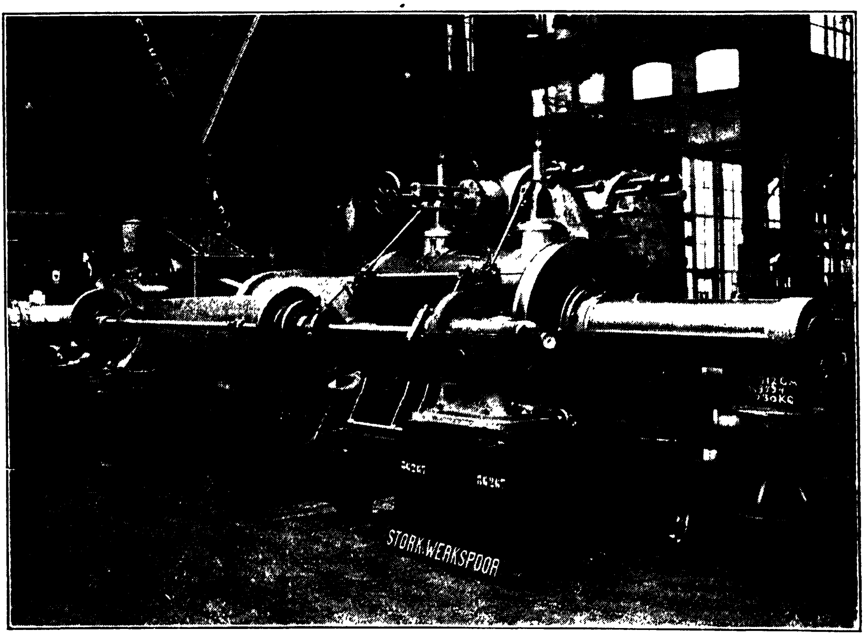


Compound Gear for Large Sugar Mills.
(*Farrel-Birmingham Co., Inc.*)

PLATES 42 & 43.



CANE MILL DRIVING ENGINE WITH RIDER PISTON VALVE GEAR.
(Fawcett, Preston & Co., Ltd.)



POPPET VALVE MILL ENGINE WITH OIL GOVERNOR.
(Gebr. Stork & Co.)

The empty load diagram, *Fig.* 241, was taken by the author and it will be seen that admissions of 5 per cent. are efficiently achieved.

The power output of the engine was considerably raised, to about 320 h.p., far above what previously could have been obtained with a single piston valve and throttling governor. Existing single slide valves can be rehabilitated, as shown, to nearly the same extent as their competitors, the Corliss and poppet valves, for use in sugar factories.

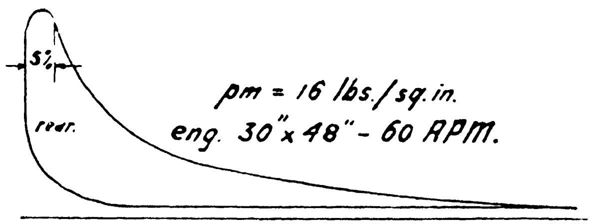


Fig. 241.—Empty Load Diagram.

To be completely independent with the four periods of steam distribution at each cylinder end, four-way steam distributions have been designed, although the piston valve engine ranks high in reliable performance. The types treated in the following sub-headings are used in sugar factory practice.

3.—Corliss Mill Engines.

The Corliss engine, of American design, was patented in 1849, and its original conception was of such outstanding excellence that only minor changes have since been made by way of improvement. Its greatest application has been in the U.S.A., for in Great Britain and on the European continent its adoption has not been so general.

A *Corliss Cylinder* of a 30 in. × 60 in. mill engine in actual operation and measured by the author is indicated in *Fig.* 242. The steam valves are at the top, whereas the exhaust valves are located at the bottom end. These Corliss valves are rotating or swinging slide valves of cylindrical shape and are easy to manufacture.

The steam connexion for the inlet or steam valves for the cylinder of *Fig.* 242 is located on top, whereas the exhaust connexion is at the bottom. The casting is symmetrical on the vertical centre line and the front cover is pressed between the engine frame and the cylinder.

The valve chambers are cast integrally with the cylinder, which is a better construction than to have them in removable covers. Double port valves are used, so the full port openings are reached with half the valve travel. The exhaust valves receive the pressure from the top and although a larger dead space will result, they nevertheless are always pressed against their seats.

The trunnion for the wrist-plate is attached by a circular plate on the mid-centre of the cylinder. This wrist-plate operates the four valves by connecting bars.

The weakest spot of the cylinder is at the locus of the steam ports, as a considerable amount of metal is taken away and the connecting ribs *a* are reinforced by throughgoing tap bolts.

In the exhaust valve chambers a connexion is made for a 2 in. relief valve, for draining off condensation water and thus avoiding the effect of water hammer, which has destroyed many a steam cylinder.

Corliss valves require generous lubrication for good operating service, but they will keep steam-tight very well and for low speed engines they form an ideal system of steam distribution. The power consumption to operate the valve gear is small, as may be ascertained from the hand operation of the wrist-plate, when the engine is drained or started.

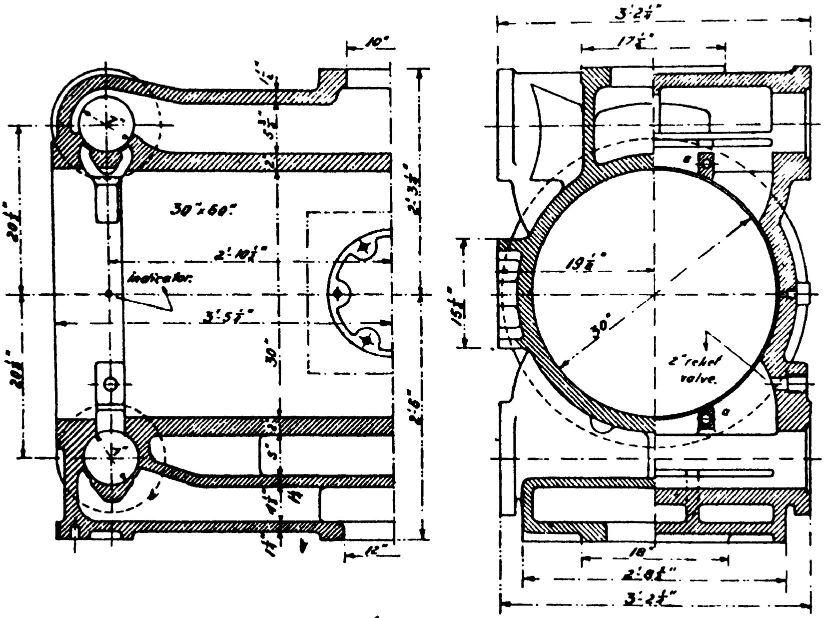


Fig. 242.—Corliss Cylinder.

A *Corliss Exhaust Valve* is shown in *Fig.* 243. It is a simple casting and the four rings, integrally cast with the valve, give sufficient bearing surface. The valve is operated by the T-head of the valve stem, which engages in its front face groove.

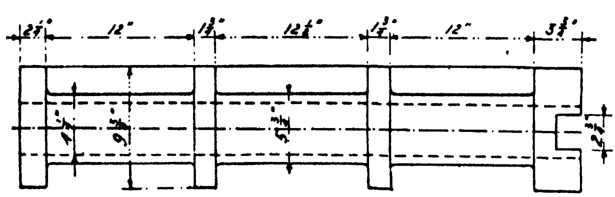


Fig. 243.—Corliss Exhaust Valve.

The cross sections of both *Steam and Exhaust Corliss Valves of the Double Admission Type* are drawn in *Fig.* 244. They belong to a 46 in. × 60 in. sugar mill engine. The exhaust valves will remove any condensation water from the cylinder as soon as they open. When worn the latter will tend to leak and the steam will blow through to the exhaust, for which reason this type of exhaust valve sometimes is provided with spring-loaded shoes on the lower side.

The dead space of Corliss cylinders is small and generally not over 3 per cent. of the cylinder volume in good size mill engines, as compared with 6 to 12 per cent. in poppet and slide valve engines.

The *Valve Releasing Gear* of a Corliss engine, having 22 in. diam. and 42 in. stroke, for driving a cane crusher, is shown in *Fig.* 245. It is practically the same construction as when invented by HENRY CORLISS in 1849.

On the valve stem is keyed a lever *a* which carries the engaging link block and the dash-pot rod. By engaging the hook *c*, which is pivoted to the bell crank *b*, the lever *a* is lifted and the valve stem rotated, thus opening the valve. As soon as the hook *c* comes in contact with the cams on the cam ring *d*, the hook is released and the valve is closed by the dash-pot action.

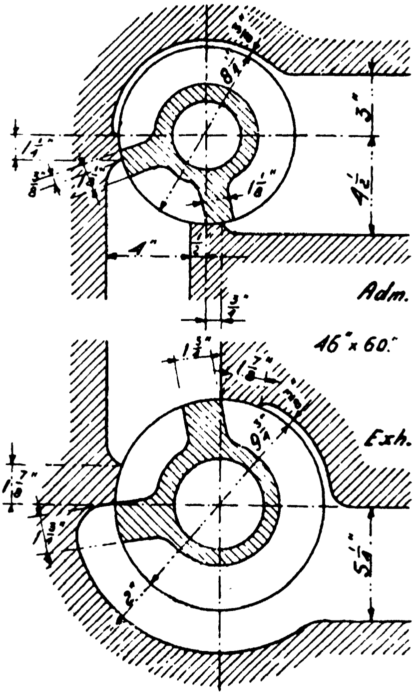


Fig. 244.—Steam and Exhaust Corliss Valves of the Double Admission Type.

The bell crank *b* is operated by the wrist-plate rod, whereas the cam ring *d* is connected to the governor rod.

The cam ring *d* has two cams, one for high positions of the governor thus for controlling speeding or slowing down of the engine, whereas the other cam is for emergency and will only be operated by the breakage of the governor belt or by hand, when a quick stoppage of the engine is desired.

The bell crank *b* bears on a hub, cast integrally with the bonnet, so there are no reactions on the valve stem proper.

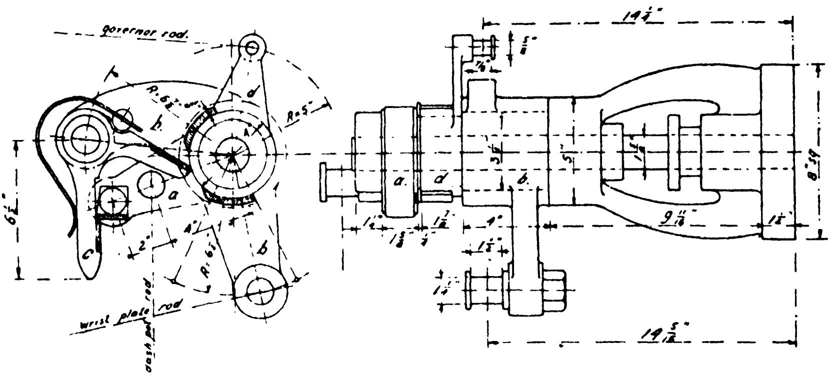


Fig. 245.—Valve Releasing Gear of a Corliss Engine.

A stuffing box is provided on the valve stem, but a steam-tight collar is forged on it, which pushes against a polished surface on the inside of the bonnet.

The stuffing box, therefore, needs only to be lightly packed and the free movement of the valve stem will not be impaired.

The dash-pot generally has a differential piston, the ring-shaped cylinder being for vacuum and the cylindrical space as an air cushion when the valve is released. The vacuum exerts a suction, so the valve will close rapidly when released.

Leather cups or cast iron piston rings are used for sealing this differential or trunk piston.

The *Corliss Valve Motion Diagram* is very interesting. From *Fig.* 246 it is seen that the wrist-plate has as its object the changing of the angular movements of the eccentric. The closing angle $\alpha_{,}$ is smaller than the corresponding wrist-plate angle α, whereas $\beta_{,}$ is larger than β. The opening velocities thus will be higher.

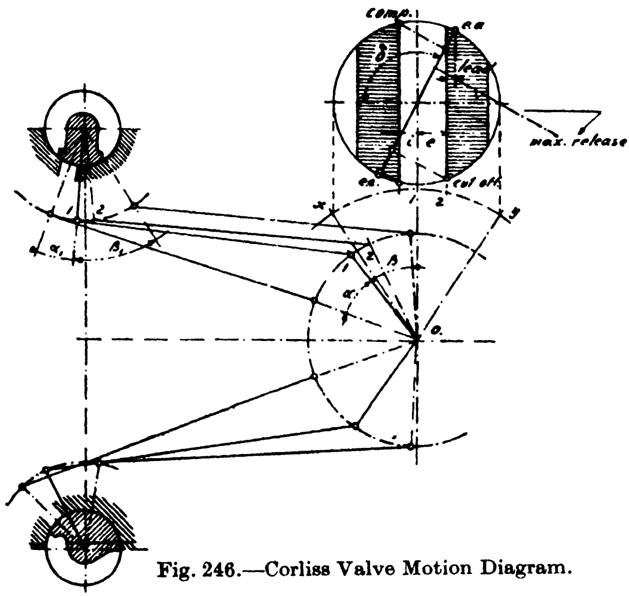


Fig. 246.—Corliss Valve Motion Diagram.

Above the wrist-plate travel x-y is drawn the Müller diagram and the steam valve is in cutting position, when the corresponding distance 1-2 has been travelled by the wrist-plate. This distance obviously is the steam lap on the valve, attaining the value e in the diagram. Correspondingly, the distance i is the wrist-plate travel necessary for putting the exhaust valve in cutting position, thus equivalent to the exhaust lap.

The steam valve should have opened a small amount when the piston is in dead centre, and this lead is also to be found in the diagram. The four steam periods are thus to be found on the stroke line *ea-ee*. The advancing angle of the eccentric δ is also easily found.

The Corliss gear being a releasing one, it will be obvious that the release has to take place before the eccentric is in dead centre, as the hook c from *Fig.* 245 will not meet the cam of the cam ring d on the return stroke. In *Fig.* 246 this moment of maximum release has been indicated, and to have a large cut-off the angle δ is made nearly 90° ; but then early admission, compression and early exhaust have attained a very small value or may not exist at all or have even become negative. This is a drawback of the one-eccentric Corliss gear. When the governor does not release at all, nearly a full stroke admission is obtained and speeding up results, but as soon as the governor effects release, the admission is reduced to less than 50 per cent. This causes the hunting of the engine, due to irregular steam admission.

The *Two-eccentric Corliss Gear* has separate eccentrics for the steam and exhaust valves and thus double wrist-plates. The steam laps can now become negative and a large cut-off or admission completely under governor control is achieved. This double eccentric arrangement is not found on all mill engines, but should be applied when the engine is small for the performance to be delivered and large cut-offs are thus required.

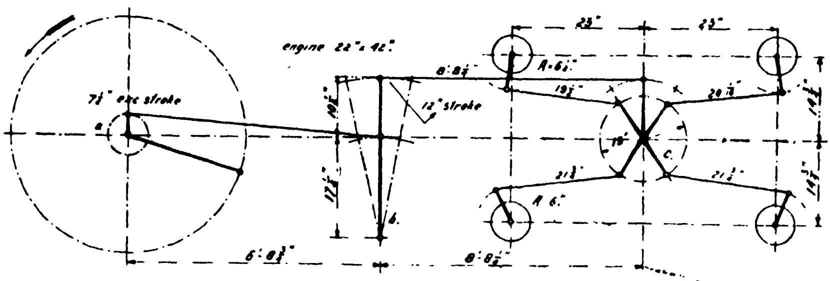


Fig. 247.—Corliss Gear for a 22 in. × 42 in. Engine.

In *Fig.* 247 is shown the general arrangement of the *Corliss Gear for a 22 in.* × 42 *in. Engine*. The single eccentric is connected to a rocking lever which has a detachable connexion to the wrist-plate. The rocking rod can be disconnected and by inserting a hand bar in the wrist-plate, the engine can be operated by hand, which is very convenient for draining or starting the engine, and also for reversing when this is needed.

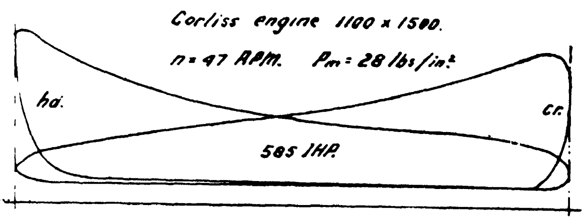


Fig. 248.—Non-Releasing Corliss Indicator Card.

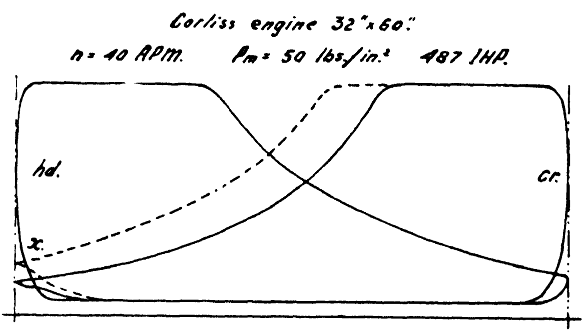


Fig. 249.—Releasing Corliss Indicator Card.

The four valves are connected to the wrist-plate and all rod heads have adjustable bronze bearings.

The Corliss steam engine, although not as simple in construction as the slide or piston valve engine, is as fully reliable and the valves will keep tight for a long run. For higher speeds and superheated steam it is less suitable and sometimes unfit.

In *Fig.* 248 a *Non-releasing Corliss Indicator Card* is shown for a Corliss mill-engine 1100 × 1500 mm., the engine thus being regulated by a throttle governor. Wiredrawing is evident and the diagram resembles the piston valve diagram. All the indicator cards mentioned were taken by the author.

A *Releasing Corliss Indicator Card*, also of a mill engine in actual operation, having 32 in. cylinder diameter and 60 in. stroke, is to be seen in *Fig.* 249.

It is a very good diagram and it shows the high mean pressure available at only 100 lbs. boiler pressure. There is no early admission and but small compression, but the exhaust valves open late, so there is retention of the exhaust steam as shown by x in dotted lines. The author has met cases where this exhaust retention has caused the engine to stall and in such instances it has to be corrected.

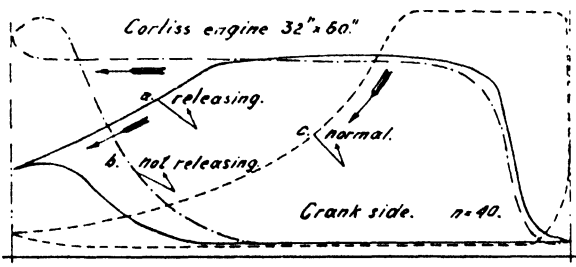


Fig. 250.—Diagram of Releasing Corliss Indicator Card.

Sometimes indicator cards are a puzzling proposition, and in *Fig.* 250 is shown a diagram from the same engine as mentioned in *Fig.* 249. The first diagram taken is indicated by full lines and, for the sake of understanding, arrows are drawn to give the direction in which it has been traced. The engine had a tendency to stall at dead centres and a non-releasing stroke then followed, as indicated by chain dotted lines. In both cases admission came too late, after the piston had travelled nearly 15 per cent. of its stroke. Moreover, the opening of the exhaust valves also was too late and the steam caused heavy pounding of the engine to an unpleasant degree. At *a* there has been expansion but at *b* the full pressure steam has been compressed, which explains the stalling tendency at dead centres. As soon as the steam pressure from the boilers dropped 10 lbs. the engine came to a standstill.

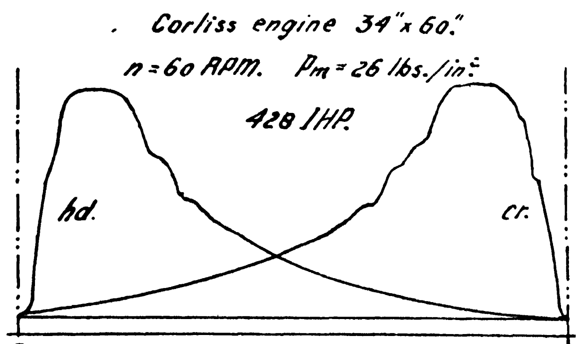


Fig. 251.—Average Corliss Diagram.

In such cases the best way is to trace how anormal diagram should look, as the factory engineer, when seeing the indicator cards, had taken the performance as reversed. With the dotted lines *c* is shown the indicator card after the author had removed the cause—a loose eccentric. The manufacturer of this special engine did not provide a key for the eccentric but only a set screw and although the shaft had been drilled to receive this screw, the latter had broken.

As the set screw of the eccentric allows different valve settings it is obvious that it is not always adjusted properly, and in *Fig.* 251 is shown the *Average Corliss Diagram* as the author has found it on many a mill engine. The admission comes too late and there is no compression nor early exhaust. The author, therefore, favours having a key in the eccentric, so that the factory engineer has only to adjust the rod lengths.

4.—Poppet Valve Mill Engines.

The poppet or drop valve was invented around 1800 by HORNBLOWER in Great Britain, and improved by SULZER in 1865; but it has not attracted the general interest in that country that it has obtained in continental Europe.

The poppet valve is an ideal medium for steam distribution, as it needs almost no lubrication and there is practically no wear. As soon as the valve has reached its seat, all motion is stopped and there is no friction between valve and seat. The poppet valve on account of its light weight is especially adapted for high speed, and high superheat has no influence on its operation.

In *Fig.* 252 is shown a *Lentz Poppet Valve*. As there are very many poppet valve arrangements of different designers, the Lentz gear has been taken as a representative one, being one of the most ingenious designs in existence.

The valve is double-seated and nearly equilibrated, as the steam pressure only acts on the ring-shaped surface between the diameters d and d_1, the valve seats having a width of about $\frac{1}{8}$ in.

The free valve area is only to a small extent obstructed by the hub, the ribs and the valve body proper, the valve lift therefore being a little less than one-eighth of the mean valve diameter. A higher lift would not give more valve opening.

The valve is guided by a large stem or cylindrical guide, attached to the bottom of the valve seat basket, and it is freely bolted to the valve spindle, so as to turn easily. A special feature is the labyrinth seal of the valve spindle. The valve spindle bushing as well as the spindle proper is ground to true dimensions, the latter being provided with grooves at regular intervals. Any steam passing along the spindle will expand in the grooves until the pressure is thus lost. For lubrication, a central oil chamber is connected to a forced feed lubricating pump.

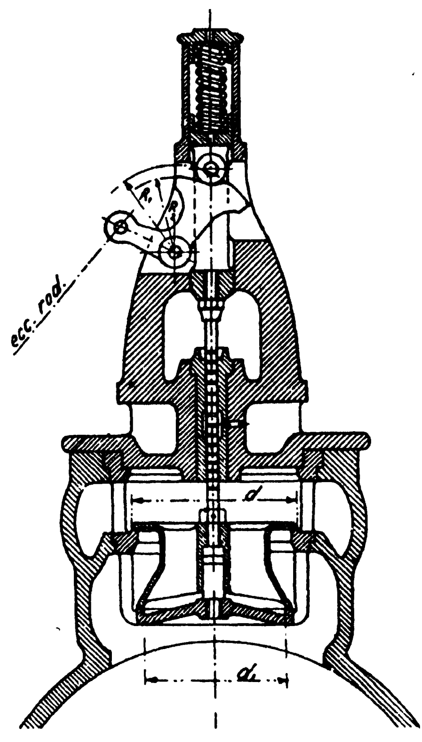


Fig. 252.—Lenz Poppet Valve Arrangement.

The valve chair contains the spindle roller guide and the valve lift is achieved by the swinging of a bell crank lever, having case-hardened bushings and a case-hardened roller path. This roller path takes a shape formed by two radii $R_,$ and $R_{,,}$ and the connecting curve between both has a bearing upon the acceleration of the valve in opening or closing and the lateral thrust upon the roller guide. The roller guide has a long bearing surface and is spring-loaded.

In the closed position of the valve, the case-hardened roller should just keep clear of the bell crank lever or cam, which can be ascertained by rotating it by hand. As the roller remains in contact with the bell crank cam, no air cushion is required for the closing period of the valve as is the case for releasing valve gear. A drip lubricator is placed on top of the valve chair.

All the four valves required are operated by rods connected to eccentrics mounted on a side shaft parallel to the engine axis and driven by a bevel gear from the crank shaft, having the same number of revolutions as the latter.

The exhaust valves have fixed eccentrics, so their valve lift is always the same, whereas the steam valves have variable eccentrics under control of a shaft governor on the side shaft.

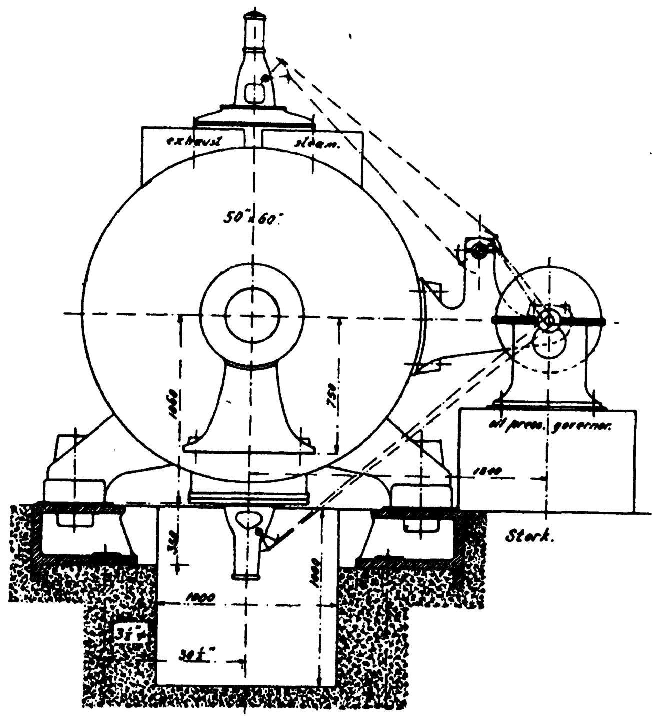


Fig. 253.—Poppet Valve Cylinder 50 in. × 60 in.

In *Fig.* 253 is shown a large *Poppet Valve Cylinder*, 50 in. × 60 in., for an existing mill engine as supplied by the author, and the location of the side shaft is clearly shown. The steam valves are operated by one eccentric by means of an inter mediate shaft. This main eccentric has a variable eccentricity under control of a potential oil pressure governor, mounted on the same side shaft.

The cylinder is secured on the existing foundations by means of a special cast iron bedplate, to make this arrangement possible. The existing foundation bolts were shortened by an oxy-acetylene torch and new threaded ends welded on. The welding has to be done very carefully and the author has provided in several instances drilled bushings for this purpose, which are welded to both pieces and which attain nearly the same strength as the full bolt area.

As the valves are arranged vertically, there is practically no wear on the spindles or the valve seats, and the author has seen poppet valves still in operation after 15 crops, and the engines running well over 60 r.p.m., without any detectable wear on the valves and seats proper. Spindles and bushings suffer from dust or dirt and oil incrustations, as do the bell crank cams, which have to be renewed sometimes.

When the engine and valve gear are kept in proper condition there are few or no repairs necessary for long periods, and the poppet valve engine is as reliable as a Corliss valve one.

Instead of the mechanical operation of the valves by eccentrics from the above-mentioned side shaft, *hydraulically operated valves* are now being manufactured by European machinery manufacturers. A special designed oil pump supplies oil under pressure to four pistons on top of the four valve chairs. The valve lift can be made of such a magnitude that free admission is obtained even with a small cut-off. The regulating is done by shortening the effective piston stroke of the driving oil pump so that the valve piston will lift for a shorter period.

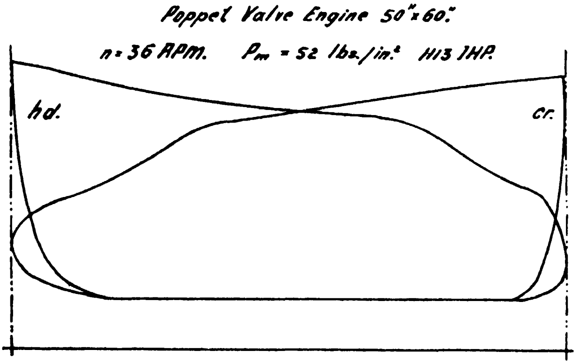


Fig. 254.—Indicator Card of Engine in Fig. 253.

Good operating results are reported from these hydraulically-operated poppet valves, as there is no wear on the operating gear which is immersed in oil. In Java the hydraulic poppet valve engine is used for driving cane mills.

Before starting the engine, the oil pressure prevailing in the system has to be raised to working pressure by means of an electrically-driven auxiliary pump. It is considered an advantage to have this oil pressure as low as possible, and the lowest operating pressure is given as between 100 and 120 lbs./sq. in.

From an existing hydraulic poppet valve engine, having one cylinder, 114 lbs. steam gauge pressure, 164°F. superheat and 10 lbs. back pressure, a thermo-dynamic efficiency of 89 per cent. has been achieved under official test, which means that 89 per cent. of the heat lost by the steam going through the engine has been actually converted into the equivalent mechanical power.

In *Fig.* 254 is shown an indicator card of the engine mentioned in *Fig.* 253; the latter was run at low speed, to obtain a large cut-off. Early exhaust and compression are taking place to time and the wiredrawing during the admission period is probably caused by too narrow steam lines or steam valve area. The engine has run up to 65 r.p.m. satisfactorily.

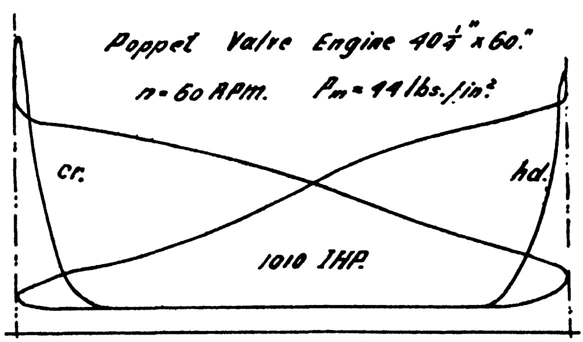


Fig. 255.—A Good Indicator Card of Poppet Valve Engine with Mechanical Gear.

A good indicator card of another poppet valve engine with mechanical gear, taken by the author, is shown in *Fig.* 255. The engine has cylinders, 40½ in. diameter and 60 in. stroke, running at 60 r.p.m. Although the compression is a trifle too high, the power loss caused through it is negligible. As the engine had originally been designed for 45 r.p.m. there is wiredrawing in evidence during the admission period.

For the control of a given valve gear the corresponding valve lifts for each position of the piston stroke can be drawn. This valve lift diagram has to contain the valve lifts under different positions of the governor.

5.—Mill Engine Details.

Most steam engine details will be found in engineering handbooks and only a general review is here given of special mill engine details.

A *Closed Engine Frame* of an up-to-date high speed mill engine of European design is shown in *Fig.* 256. The crank motion is completely enclosed and forced lubrication applied for the main bearings and the connecting rod and crosshead. At *a* a stuffing box with metallic packing is provided, so that the splashing oil will not reach the front cylinder cover, and any condensation from the piston rod stuffing box cannot mix with the oil and cause emulsification. As the enclosed engine parts are dustproof, there is little wear due to this source.

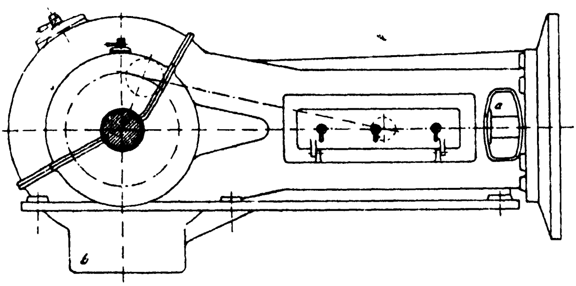


Fig. 256.—Closed Engine Frame.

At *b* the oil is taken from the oil pan after having been strained and sometimes cooled, and is re-pumped into the cycle by a forced feed oil lubrication pump. As a very generous supply of oil is pumped through the bearings, these engines run very well. The crank is of the double-throw type and the hood over the crank case can be removed; moreover, inspection doors are provided at the principal spots.

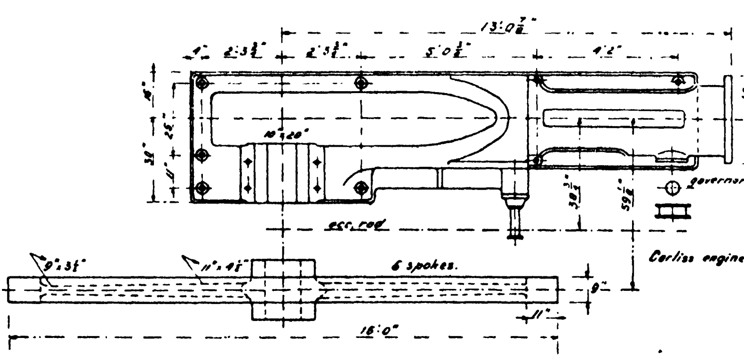


Fig. 257.—Open Girder Bedplate.

The double-throw crank shaft is generally made with two equal size main bearings and, having one enlarged crank shaft end, the engine can be used for left-hand or right-hand drive of gearing and mills.

The engine frame is very rigid, as it is *over the full length bearing* on the foundation, and unilateral stresses, caused by the main bearing reactions, are avoided.

An *Open Girder Bedplate* for a 22 in. × 42 in. Corliss mill engine of American design is shown in *Fig.* 257. The crosshead path is well supported over the full length. This bedplate is cast in one piece but, for large mill engines, the crosshead guide or path is bolted to the main bearing bedplate. The pivot for the rocking lever, as well as the flange for the governor, is also shown. The flywheel is made in two halves for easy transport, but too many parts should be avoided, as every joint might become a weak spot. The two parts are connected by four hub bolts, which should be heated before insertion, so that they will shrink when cooled, whereas the rim has a bolt on each side for erection purposes and a shrunk bar with two T-heads on each side of the rim.

Other flywheel rim connexions are made by means of a heavy pin with two keys or shrunk rings of oval form. Rim connexions, generally, have to be made very solid and in case an existing engine has to run at higher speeds, the stresses in the connecting members, caused by the centrifugal force (which increases with the square of the peripheral rim speed) should be carefully checked. In cases where it was desired to know if there was any play in the flywheel rim connexions, the author has fastened a glued paper strip, such as can be bought at any stationery shop, around the flywheel. After this had dried, the engine was put to work at the desired speed, and if any member had an elongation, the paper strip broke at the spot. If the paper does not readily adhere to the rim, a double layer of the glued paper should be applied.

For turning the engine over by hand, holes or teeth are provided on the outer periphery in which to insert a hand bar ; this is kept supported in a special pedestal for the purpose.

The flywheels of mill engines should have sufficient rim weight to ensure that a degree of uniformity of 1 : 50 is obtained for the normal speed. The flywheel inertia has to equalize the irregularities of the piston force, which causes an uneven tangential crank force during a full revolution and in dead centres even attains zero. Without proper flywheel inertia uniform running performance of the engine is not possible.

Variations in load, i.e. a difference in the required engine couple, have to be regulated by the governor and it should be recollected that a heavy flywheel will assist the governor, but will not take over its duty, nor will a good governor compensate for the effect of too light a flywheel.

With a constant engine couple, the flywheel performance is proportionate to the square of the engine speed and with equal power output even to the third power. This will indicate that high speed mill engines have the advantage for varying milling conditions, as a considerable flywheel inertia can be stored.

The crank of this engine is of the disc type, which is favoured in British and American designs. The disc is generally made of cast iron, but the author has supplied several in cast steel to replace cast iron ones. The counterweight for the crank proper and a part of the connecting rod inertia can be conveniently arranged in the crank disc.

Bayonet Engine Frames are designed by many manufacturers where the main bearing is connected unilaterally with the crosshead trunk, just as a bayonet is attached to a rifle.

Such a main engine bearing, as shown in *Fig.* 258, is composed of four parts, each being of cast iron, lined with white metal. As the bearing will wear sideways on both sides, two wedge pieces are provided, to take up the clearance caused by the wear on the white metal, as otherwise the engine will start knocking. The wedge pieces can be adjusted by nuts, which have a thrust ring held by a flange on the top cap.

Sometimes one wedge piece is provided and on wear a filler has to be placed on the opposite side. One-side adjustment, nevertheless, is liable to throw the engine shaft out of its true centre.

Vertical wear has to be taken up by fillers laid between the top and the side segments, or by planing off the material. As mill engines have an average speed not over 60 to 75 r.p.m. there is little danger that the shaft will lift under operation, when there is a small clearance on the top.

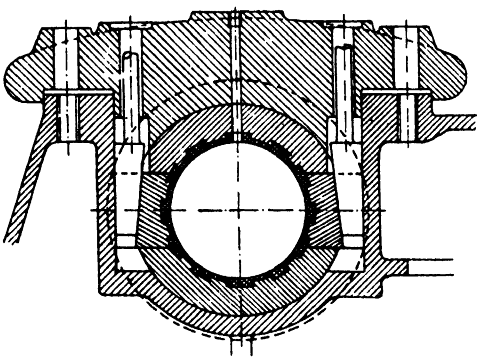


Fig. 258.—Main Engine Bearing.

When the bearing has worn down, it should be re-filled with a first-class white metal, and the cast iron segments, after having been cleaned of their old white metal and any intruding oil rinsed out with gasoline (petrol), should be tinned before the white metal is poured in. A wooden core or a piece of heavy size pipe will take the place of the shaft and distance plates of sheet iron can form the division between the segments. Sufficient material has to be poured to give a clean cut on the perforator or lathe. The segments should be heated before the white metal is poured in, so as to obtain a better adhesion.

Connecting Rods are made of high tension steel and are provided with closed heads, as shown in *Fig.* 259, having bronze or white metal lined cast steel bushings for the crank pin end, adjusted by a wedge on the front end. A filler of thin brass sheets between the two bushing halves can be removed in case of wear. The cross pin end also is adjusted by a wedge and the latter is of semi-cylindrical form for easy manufacture of the rod and even distribution of stresses. The bushings are of bronze.

For double-throw cranks and also for single-throw crank engines, the well-known marine head is extensively used, as it will allow easier erection. The closed head connecting rod has to slip sideways over the crank pin. Both types of rods can be made of sufficient strength and although the author has seen the marine type replaced by the closed head type in a case where the marine head bolts had broken (due to water hammer) it is doubtful whether a closed head type rod would have escaped breakage under such conditions.

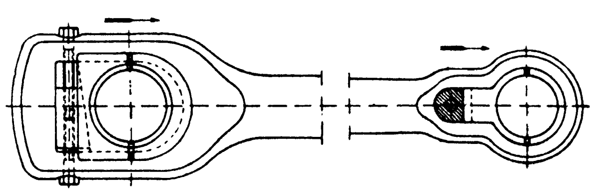


Fig. 259.—Connecting Rod.

Special care has to be taken that both bushes are adjusted in the direction indicated by the arrows in the figure, as the rod length otherwise will change under wear and thereby the piston clearance will sometimes become dangerously short at one end of the cylinder.

The *Crossheads* in our present-day engines all have cylindrical shoes, as shown in *Fig.* 260, the engine frame being bored simultaneously when the connecting flange to the cylinder is turned, and thus good alignment will be obtained.

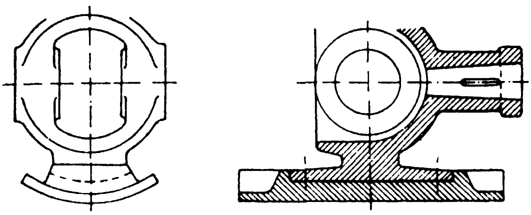


Fig. 260.—Crosshead.

The crosshead body is made of cast steel and the shoes of cast iron. Bronze shoes or cast iron shoes lined with white metal are sometimes used, but this is not a necessity. Fine grained cast iron will form a hard and polished wearing surface in the course of running the engine. It should be repeated that the engine has to be well aligned, as the author knows by experience that most of the crosshead troubles come from this source. The connexion of the crosshead and piston rod is by means of a cone and a key. A conicity of about 1 : 25 gives a firm connexion, but has the disadvantage that it is sometimes hard to loosen the rod, as the adhesion is very firm. A set of counter keys, therefore, should be supplied by the manufacturer for dismantling the engine during the dead season, so that the piston can be subject to a careful inspection.

Other manufacturers employ straight piston rod connexions, a stop being provided inside the crosshead body, so that the rod can be easily removed when the key has been taken out. The key of this arrangement has to be made of ample strength as it will suffer more than that with a conical connexion.

In *Fig.* 261 is shown a cast iron crosshead of American design noted by the author, having crosshead shoes adjustable by means of an enclosed wedge piece. As the normal pressure of the crosshead is downwards for engines turning over, it is upwards for engines turning under and the adjustable shoe should be on the wearing side. As the manufacturer aims to provide for both directions of rotation, the wedge pieces are supplied for both shoes. In *Fig.* 260 thin sheet packings have to be laid between the shoes and the crosshead body in case of wear, and the crosshead has to be taken out for this purpose.

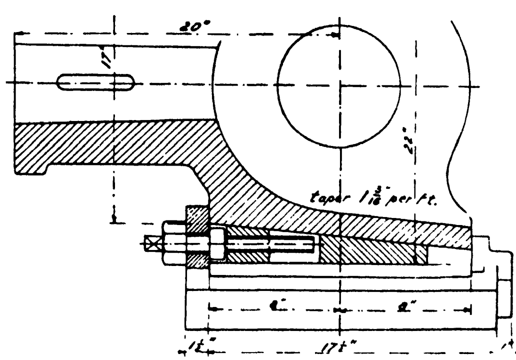


Fig. 261.—Cast Iron Crosshead of American Design.

The conical rod connexion does not allow adjustment of the piston clearances and, therefore, American designers prefer the threaded connexion with a shallow safety nut. This makes a convenient connexion, but dangerous fillet stresses, causing early fatigue of the material, result, especially when the piston is not properly guided and very small deflections are taking place at each stroke. The author has seen several breakages of piston rods at the threaded part. Keyed rods may break at the locus of the key, this being the weakest spot of the rod, but with conical attachment this seldom will happen.

Piston Rods are usually made of high tensile carbon steel or nickel steel; in any event a fatigue-resistant material of sufficient hardness should be selected.

The piston rod should be supported preferably at both ends for mill engines of medium and large size and a rear guide should be provided. As the purpose of the rear guide is to ensure less wear on the piston path inside the cylinder, it will be obvious that the weight of the piston and rod must be properly supported by the crosshead and the rear piston rod guide. A small deflection will therefore always take place and the stuffing boxes ought to make allowance for this.

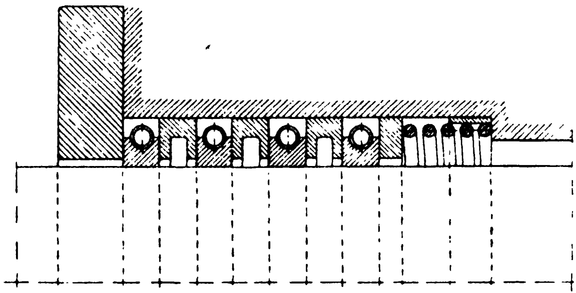


Fig. 262—Metallic Packing for a Piston Rod.

In *Fig.* 262 is shown a *Metallic Packing for a Piston Rod* as supplied by the author, which will need no adjustment and will allow for small deflections of the piston rod.

The packing rings are of Parson's white bronze or similar material, tangentially cut in two or three sections and held together each by a surrounding spring of non-corrosive material. Between the packing rings are cast iron distance rings, provided with expansion chambers for the escaping steam. Good lubrication and good drainage of accumulating oil and condensed steam have to be provided. The rods have to be ground to exact dimensions and be well polished, and dust or dirt should be kept away from the rod, as it will cause grooves which destroy any packing. A spring supplies the necessary pressure to hold the packing rings tightly together.

Sometimes the piston rod is supported in the bottom bushings of the stuffing boxes and these should be of ample proportions for their purpose. But lubrication of these bushings is rather difficult.

For engines not having a rear piston rod guide, the lower part of the piston has to be turned according to the cylinder diameter for about one-third of its periphery, so that there will be sufficient bearing surface. A groove lined with Parson's white bronze or similar anti-friction material is sometimes cut into the piston body. But it should not be forgotten that lubrication inside the cylinder is difficult and beyond inspection, and wear of the hot cylinder walls will take place rapidly under unfavourable conditions.

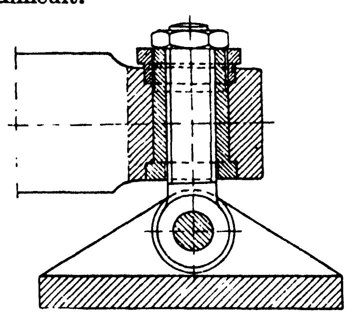


Fig. 263.—Rear Guide Shoe.

A *Rear Guide Shoe* of the author's design is shown in *Fig.* 263. The pin connexion will allow for small deflections of the rod and prevent the tipping of the shoe, which will scrape the oil in one direction. The eye bolt has to be fitted tightly in the rod bore, as any play will cause heavy wear on this bolt or the threaded bushing, through the reciprocatory movements. The shoe can be easily attached, by turning the rod 90°, as the shoe is of cylindrical shape.

To regulate the power requirements under varying load every mill engine is provided with a *Governor.* Two methods are employed for the purpose, one by varying cut-off, which is the most desirable for economic steam consumption, and the other way by means of a throttling governor, which reduces the steam pressure by wiredrawing, while having a fixed cut-off. Varying cut-off is achieved by Rider piston valves, or Corliss and poppet valve gear.

The Porter centrifugal governor, although obsolete for other engine types, is still favoured by several designers for sugar mill engines. Centrifugal force throws two balls out of their path and this movement is transferred to the governor lever connected to the variable cut-off gear. A counterweight steadies the movement as otherwise the governing action would be erratic. An oil brake is also provided for the same purpose and to prevent the engine hunting.

In Europe, spring-loaded governors designed by HARTUNG, JAHN, HARTNELL and PROELL are used for mill engines. Shaft governors are especially used for poppet valve gear, whereas the well-known Pickering governor of British design occupies a predominant place in throttling regulation, as a reliable performance is given by this type.

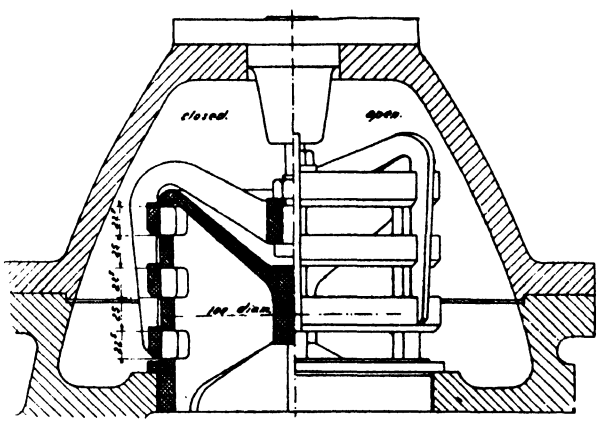


Fig. 264.—Balanced Throttle Valve.

Throttling governors should have a well-balanced throttle valve, as otherwise the steam pressure will counteract the opening or closing and thus the engine may start to speed up or slow down.

The author has had unfavourable experience with a 34 in. × 60 in. mill engine of the piston valve type, which had the regulator acting on an unbalanced throttle valve; this resulted in some instances in the engine accelerating to over 100 r.p.m. and then rather abruptly slowing down. The milling performance, of course, became very irregular and less efficient. A *Balanced Throttle Valve* of the author's design, shown in *Fig.* 264, was therefore installed to overcome the difficulty.

The brass seat of the valve is of the sleeve type, being closed on top and having three horizontal ports. The sleeve is reinforced on the inside with six ribs of sufficient strength. Over this inner sleeve is mounted a bell-shaped outer sleeve, also provided with three ports, but so arranged that the openings of the inner sleeve can be covered by the strips of the outer one. The latter is raised or lowered by the centrifugal governor of the engine above the valve housing.

It will be obvious that there is no steam load on the valve, as it is completely balanced and no side thrust can act on the bell sleeve. Moreover, the three ports will achieve a quick closure, giving sufficient steam passage at reduced lift. Since it was installed, the equipment has given satisfactory regulating performance to the engine.

Milling requires sometimes great variety in the speed of the driving engines, and the centrifugal governor will not cover a large range. An *Oil Pressure Governor* for mill engines has accordingly been designed by Continental manufacturers for this purpose (it had already been used in elementary shape by American manufacturers on vacuum pump engines). In Java the oil pressure governor has found many an application and the author has supplied several in Cuba.

In *Fig.* 265 is shown a simplified design of such an oil pressure governor[1] which will easily explain its fundamentals.

A geared oil pump is driven by, and in proportionate speed to, the mill engine, and it will be obvious that a larger quantity of oil is delivered when the engine attains a higher speed. This oil is led through piping to the governor inlet c as shown in *Fig.* 265 and it will leave by the needle valve *a*, which has been graduated for normal speed. When the engine accelerates a larger amount of oil has to flow through the needle valve, but this can only be achieved by a higher oil pressure, as is evident from the laws of hydrodynamics.

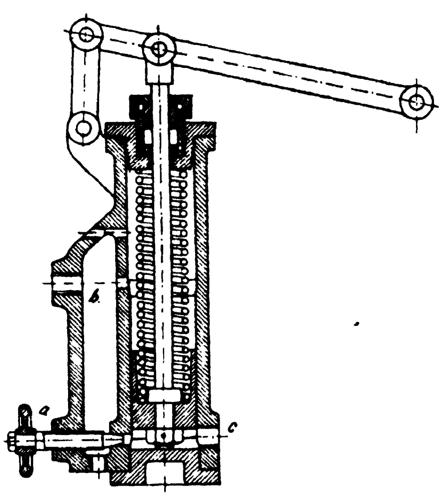


Fig. 265.—Oil Pressure Governor.

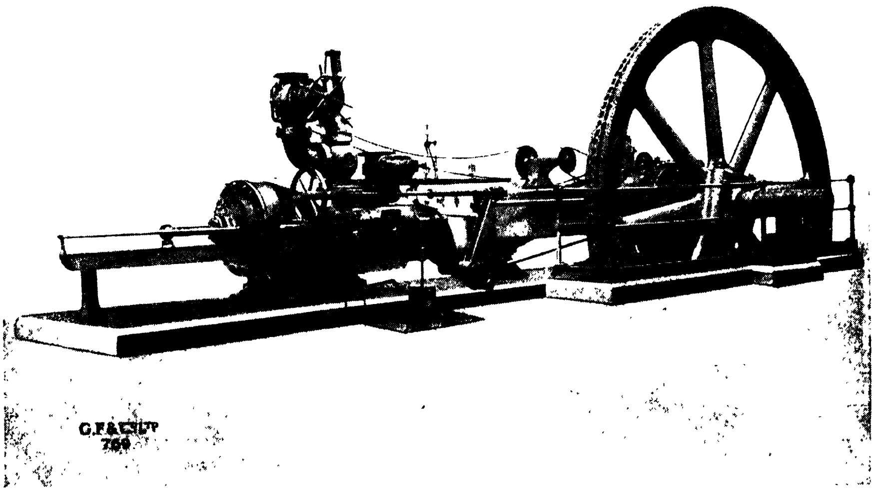
This increased pressure will raise the spring-loaded piston, which in its turn will change the position of the governing lever. The piston is 80 mm. in diameter and the normal oil pressure on which the needle valve is adjusted is about 45 lbs./sq. in. Any increase in speed will very soon raise the oil pressure up to 75 and 90 lbs. and the governing force thus to about 220—350 lbs. which is far in excess of what can be obtained with a centrifugal governor.

The governor piston has labyrinth grooves, but a slight leakage of oil may occur, as a liquid like oil does not expand like steam and this oil leakage will be released, together with the outgoing oil, at *b* which leads to the suction pan of the oil pump. At the top is an air vent and the packing on the piston rod is against leaking oil. It will be obvious that no pressure must be allowed in the discharge pipe line for the oil, as it will act on top of the piston.

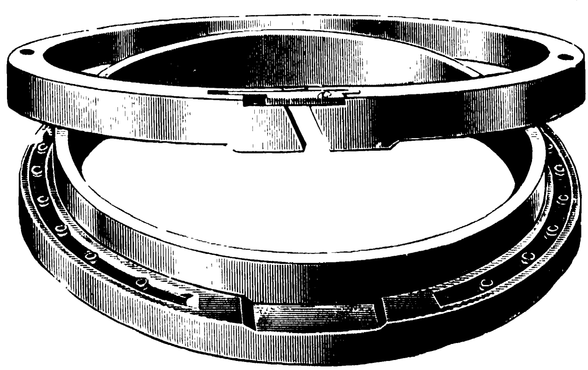
Improved constructions, aiming at nearly constant spring pressure and better oil release at raised piston position, are now made by different manufacturers and very wide variations in engine speed, e.g., between 36 and 72 revolutions per minute, have been noted in practical milling operation by the author.

Reversing of mill engines is effected by means of the Stephenson link for slide valve, piston valve and also for Corliss valve gear. Poppet valve engines have the reversing gear on the side shaft with the angularity of the eccentrics changed by about 100° against the direction of rotation. Some manufacturers apply a differential bevel gear for the purpose, whereas others provide a claw clutch which can be engaged for either forward or backward rotation.

[1] See A. E. KLAY, *Het Archief*, 1932, p. 934.

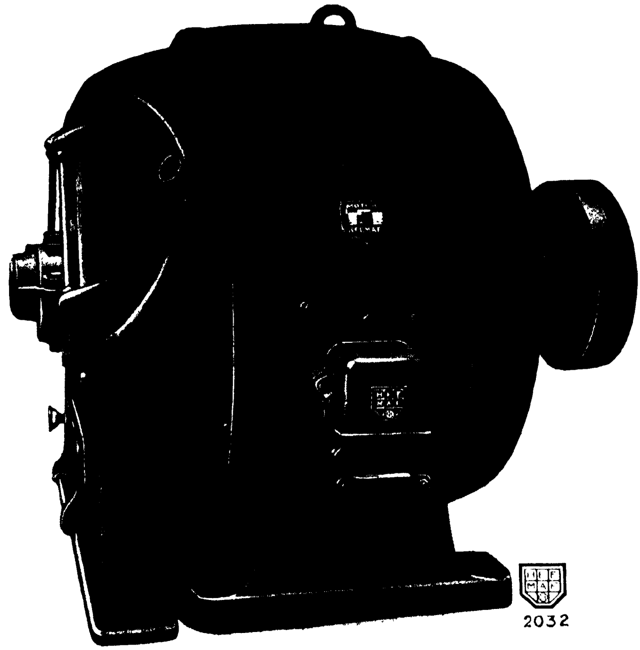


PISTON VALVE MILL ENGINE WITH REVERSING GEAR.
(Geo. Fletcher & Co., Ltd.)

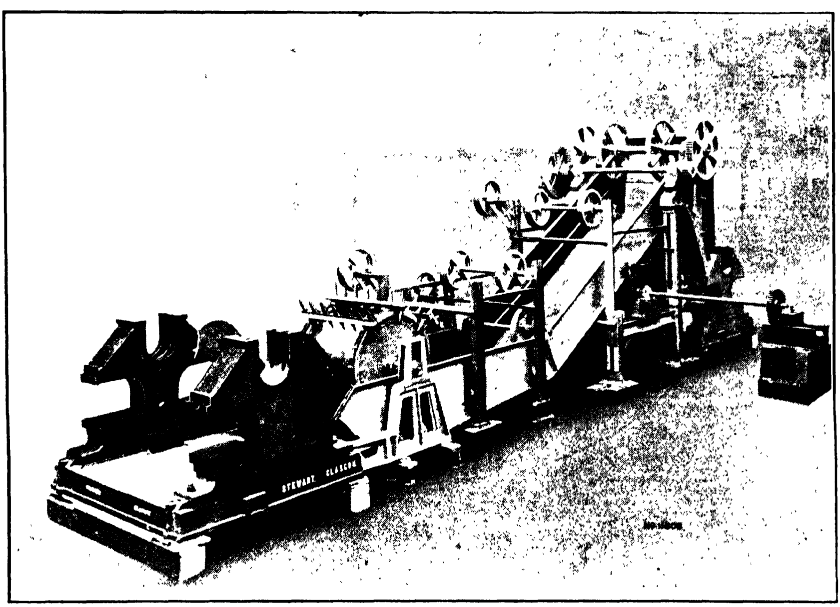


SUPER LIMIT PISTON RINGS.
(Lancaster & Tonge, Ltd.)

PLATES 46 & 47.



DIRECT CURRENT MOTOR FOR SUGAR MILL DRIVE.
250 H.P., 462 r.p.m. 440-Volt.
(*Heemaf N.V.*)



MACERATION BATH TYPE INTERMEDIATE CARRIER.
(*Duncan Stewart & Co., Ltd.*)

As the reversing of sugar mill engines is not a very frequent occurrence, many Corliss gears are reversed by hand operation and this system can be applied to slide and piston valves as well.

Before starting up the engine, steam should be admitted to warm it up, so as to reduce steam entrance condensation. Every engine should be started slowly until all condensate has been removed and the danger of water hammer is thus removed.

Pistons of mill engines are preferably of one-piece construction, as these are the lightest, a point of importance for higher speeds. A reinforced construction, applying ribs and steel studs between the two piston walls, has given good operating performance. Piston rings generally are of the self-expanding type, made of cast iron of less Brinell hardness than the cylinder casting, so that the piston rings, not the cylinder, will have the bigger share in the unavoidable wear. The one-piece piston ring should have an overlapping lip at the splitting point and the expansion pressure should be a gentle one. It is sometimes argued that one-piece piston rings will wear the cylinder oval, but this need not be feared with well designed one-piece rings.

6.—Power Development and Steam Consumption.

It is good practice to have indicator cards taken at regular intervals from the steam engines in a sugar factory and especially from the larger ones, as they will reveal defects in the steam distribution which generally are the cause of a larger live steam consumption.

The mean pressure of a given diagram can be easily calculated according to SIMPSON'S rule, dividing the diagram in ten vertical strips and measuring the mean height of each. The sum of the ten measurements, divided by ten, will give the mean effective pressure to the same scale as the indicator card has been taken.

As any mechanical work performed is the product of a force multiplied by the distance covered in a certain unit of time, it is obvious that the indicated power output of a given engine amounts to :—

$$\text{i.h.p.} = \frac{p_m \times L \times A \times n}{33{,}000} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (71)$$

where p_m = the mean effective indicator card pressure in lbs./sq. in.
L = the length of a double stroke in feet (four times the crank radius).
A = the piston area in sq. in. after deduction of the rod area.
n = the number of revolutions per minute.

As 1 h.p. equals 550 ft./lbs. per second, it will be 33,000 ft./lbs. per minute.

For metric conditions the formula will read :—

$$\text{i.h.p.}_{metric} = \frac{p_m \times C \times A}{75} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (72)$$

where p_m = the mean effective indicator card pressure in kg./cm.^{2}

C = the mean piston speed in metres per sec. $\left(\frac{S \times n}{30}\right)$

A = the piston area in sq. centimetres.
(S = the stroke in metres = twice the crank radius).
(n = the number of revs. per min.).

One metric horse-power equals 75 kgm. per sec. or 4500 kgm. per min. and the formula can be written similarly as (71) in this way :—

$$\text{i.h.p.}_{metric} = \frac{p_m \times L \times A \times n}{4{,}500} \quad \ldots\ldots\ldots\ldots\ldots\ldots \text{(72a)}$$

in which L is twice the engine stroke in metres and the other values of the metric system are as mentioned for (72).

The *Mechanical Efficiency* of a mill engine is generally between 85 and 93 per cent., according to the make, and poppet valve engines rank highest in this respect. From 15 to 7 per cent. of the power input is thus lost in friction of the engine parts. It has to be remembered that at low speed and with no load the friction resistance of the mill engines may be larger than when under full load.

The *Thermo-dynamic Efficiency* depends on very many factors such as throttling, degree of cut-off, compression, insulation, dead space, piston speed and superheat; and it may differ for two engines of the same size but of different make.

For overall calculations the following table is given, although it cannot be applied in all instances :—

Full admission engines (duplex pumps)	0·25 to 0·35
Slide valve engines with throttling governor ..	0·40 to 0·50
Piston „ „ „ „ ..	0·45 to 0·55
Rider „ „ automatic cut-off	0·55 to 0·65
Corliss „ „ „ „	0·60 to 0·70
Poppet „ „ „ „	0·65 to 0·75
Curtis steam turbines	0·45 to 0·55
Reaction steam turbines	0·55 to 0·65
Special precision turbines	0·75 to 0·80

These special thermo-dynamic efficiencies are given for back pressure engines as used in sugar factories. As the heat drop between incoming and outgoing steam is not a large one, the efficiency generally is high.

The *Steam Consumption* of a given mill engine can only be accurately measured by condensing the exhaust steam and weighing it, if the moisture of the live steam entering the engine is known. This method, nevertheless, can be applied but seldom to sugar mill engines, and the steam consumption has to be calculated from the indicator diagrams, as from those the moment of cut-off, the moment of compression, the entrance and exhaust pressures, as well as the developed indicator h.p., can be obtained.

The total steam consumption of an engine, without considering the steam pipe condensing loss, is composed of three parts, according to HRABAK :—

$$C = C_1 + C_2 + C_3 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(73)}$$

where C = the total steam consumption or steam weight.

C_1 = the weight of the entering steam reduced by the weight of the steam remaining in the cylinder during compression.

C_2 = the loss by condensation of the steam through radiation and contact with surfaces below the steam temperature.

C_3 = the steam loss through leakage of the steam distribution units and the piston and piston rod.

C_1 can be calculated, but C_2 and C_3 are empirical data.

When the steam is cut off, a volume in the cylinder is occupied consisting of the piston area A, multiplied by that part of the stroke S_1 travelled when the steam is cut off *plus* the dead space DS given in percentages of the stroke. Thus with an engine having 40 in. stroke, 25 per cent. cut-off or admission and 5 per cent. dead space, the piston area, reduced by the piston rod area, has to be multiplied by $(0{\cdot}25 + 0{\cdot}05) \times 40 = 12$ in. This volume in cub. in. has to be multiplied by the specific weight of the entering steam W_1 which is to be found in engineering handbooks, given in lbs. per cub. ft. for British and American practice, and in kg. per cub. metre for metric calculations. Proper conversion, of course, has to be made.

Similarly, when the exhaust port is closed for compression when the piston still has to travel a part of the stroke S_2, the volume of the entrapped steam is equal to the piston area multiplied by $S_2 + DS$; and to obtain the steam weight of this volume, it has to be multiplied by the specific weight of the exhaust steam W_2 and the formula may be written :—

$$C_{1s} = A \times [(S_1 + DS) \times W_1 - (S_2 + DS) \times W_2]$$

per stroke of the engine and as $2n$ strokes are covered per minute, the steam consumption per i.h.p./hour will amount, on inserting the factor 1728 for conversion of cub. in. into cub. ft., to :—

$$C_1 = A\,[(S_1 + DS)\,W_1 - (S_2 + DS)\,W_2] \times 2n \times 60 \div (N_i \times 1728).$$

As $N_i = p_m \times S \times A \times 2n \div 33{,}000$, the formula will transpose to :—

$$C_1 = \frac{1145 \times [(S_1 + DS)\,W_1 - (S_2 + DS)\,W_2]}{p_m \times S} \quad \ldots . \; (74)$$

S_1, S_2 and DS are measured in inches, W_1 and W_2 in lbs. per cub. ft., p_m in lbs. per sq. in., and S (the engine stroke) in feet.

For condensation losses 30 to 40 per cent. has to be added for saturated steam and 15 to 20 per cent. for superheated steam. In case of superheated steam, the exhaust steam temperature has to be measured to see if it is still superheated or already saturated.

The leakage loss is about 2 to 5 per cent. of the amount of C_1.

Another way of calculating the steam consumption is with the *Entropy or Heat Diagram*, where the heat contents are drawn as absissæ and the entropy, being the proportion of the amount of B.Th.U.'s per unit of weight and the absolute temperature, as ordinates. From this heat diagram, drawn to scale, the heat drop by adiabatic expansion can be measured at once.

In the following table are given the different data for the calculation of the steam consumption by means of the heat diagram :—

Live steam pressure	100 lbs. 7 kg./cm.^{2}	gauge pressure
Exhaust steam pressure	7 lbs.	0·5 kg./cm.^{2}
Temperature of superheated steam	428°F.	220°C.
Superheat	90°F.	50°C.
Heat drop for saturated steam....	122·4 B.Th.U./lbs.	68 cal./kg.
Heat drop for superheated steam ..	131·4 B.Th.U./lbs.	73 cal.kg.

1 calory $=$ 3·986 B.Th.U.

1 metric h.p. $= 75 \times 3600 \div 427 = 632$ cal./hr.

1 h.p. $= 550 \times 3600 \div 777{\cdot}64 = 2546{\cdot}2$ B.Th.U./hr.

1 h.p. $=$ 75·94 kgm. $=$ 1·013 metric h.p.

By dividing the heat amount of an i.h.p. by the heat drop Δ, the amount of steam per i.h.p. hour will result for the ideal engine, having 100 per cent.

efficiency. For engines having a thermo-dynamic efficiency η_t, the steam consumption will thus be :—

$$C = \frac{632}{\Delta \times \eta_t} \text{ kg. steam/hour per metric i.h.p.} \quad \ldots \quad (75)$$

$$C = \frac{2546 \cdot 2}{\Delta \times \eta_t} \text{ lbs. steam/hour per i.h.p} \quad \ldots\ldots\ldots\ldots \quad (76)$$

For saturated steam conditions as mentioned above, the steam consumption per hour for metric and British h.p. can be tabulated as follows :—

Thermal efficiency 0·30	..	37·2 kg.	or	69 lbs.
0·40	..	23·0 kg.	or	52 lbs.
0·50	..	18·6 kg.	or	42 lbs.
0·60	..	15·5 kg.	or	35 lbs.
0·65	..	14·3 kg.	or	32 lbs.
0·70	..	13·3 kg.	or	30 lbs.
0·75	..	12·4 kg.	or	28 lbs.
0·80	..	11·6 kg.	or	26 lbs.

With high superheat, still higher thermo-dynamic efficiencies and thus lower steam consumptions may be reached, but the above-mentioned cover the steam consumption of the average sugar mill engines.

7.—Electric Motors for Mill Drives.

Three-phase alternating current was used for the first electric mill drive in Cuba by American manufacturers about 20 years ago, but direct current has been used in Java and Peru since about 1925.

The advantages of electric mill drive have to be sought in the first instance in the centralization of the power production on the premises and thus a high efficiency, due to the large units employed, may be obtained. Moreover, the maintenance of the electric motors is low as compared with individual steam engines, which require more lubrication, packing, etc. Pipe lines for live steam and exhaust are of course not required as with individual steam drives, and heat losses through radiation are thus reduced.

The efficiency of the electric drive, nevertheless, is reduced through the conversion into electric current at the generator and the conversion from current into mechanical power at the motor, and the higher efficiency of the large prime mover will not always offset this difference. As steam turbines are widely used for generating electric current, it should not be overlooked that back pressure turbines with not too high an initial steam pressure have thermo-dynamic efficiencies generally well below those of the larger size steam engines required for individual steam drive. Electrification, therefore, for small power consumers like pumps, etc., has indisputable advantages, but for mill drive its merits are not generally accepted.

Three-phase alternating current is so convenient that it has replaced direct current installations in nearly all industries, and it is also very adaptable for the machinery in a sugar factory, as well as for transforming low to high voltages for power supply at far distant pumping stations or for domestic service at the "colonias" or field workers' dwelling-houses.

Direct current, nevertheless, has special advantages in respect to speed regulation and has found application for mill drives as well. In *Fig.* 266 is shown the *Direct Current Scheme* according to WARD-LEONARD. The turbine drives the main generator for the mill motors through a reduction gear. This generator supplies current at varying voltage and thus varying speed to all the connected motors. The field current for the individual motors and the generator is developed at a constant voltage in a special exciter mounted on the same shaft, and regulation of the individual motor fields is thus possible without loss of power. The power output of each motor can be lowered or increased above the normal rating as desired and within large limits, although the windings or coils have to be designed for these larger current inputs, especially when the latter are desired for longer periods of time.

About 20 per cent. of the individual speed above or below normal can be obtained, whereas the general regulation will allow for about 30 per cent of the normal speed.[1]

The combined efficiency of generator and mill motors is about 83 per cent., so 17 per cent. of the power input is lost for D.C. power transmission. The generator speed has to be low to allow the commutator to operate well and 750 r.p.m. is usually adopted. The voltage varies between 295 and 460 volts.

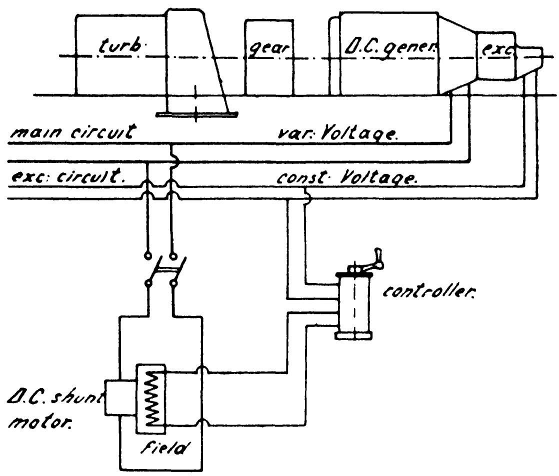


Fig. 266.—Direct Current Scheme.

The motors are *D.C. Shunt Motors*, separately excited, having a commutator like a dynamo. The cost price of D.C. motors is higher than that of A.C. ones, and prices should be compared before the selection is made.

The power output of the individual motor can be measured by volt and ampere readings, as there is no other than Ohm's resistance :—

$$h.p. = \frac{I \times E \times \eta}{746} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (77)$$

where I = the intensity of the current in amperes.

E = the electro-motive force in volts.

η = the electro-mechanical efficiency of the motor (about 0·92).

One h.p. is equal to 746 watts and a metric h.p. is equal to 736 watts. For caloric calculations one kilowatt/hour is equal to 860 cal. or 3411 B.Th.U.'s. (when calories are given, kg.cal. are meant).

An *Alternating Current Scheme* is shown in *Fig.* 267 and it has found exclusive application in Cuba, where most electrically-driven mills are to be found.

The prime mover is directly connected to the alternator, and the exciter for the alternator field is mounted on the same shaft as the alternator.

As its name indicates, the alternating electro-motive force swings between a positive and a negative maximum and the number of these reversals or

[1] See the article of J. W. DEN HAAN, Fourth Congress of Intern. Soc. of Sugar Cane Tech., Puerto Rico, 1932, abridged in *Int. Sugar Jl.*, July, 1932, page 260.

periods per. sec. is called the *frequency* of the A.C., and depends upon the number of revolutions and the number of double magnetic poles, each composed of one north and one south pole, according to the formula :—

$$F = \frac{P \times n}{60} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (78)$$

where F = the frequency per second.
P = the number of double poles.
n = the number of revolutions per minute of the alternator.

The frequency for British and Continental standards is 50 per second, whereas in American practice 60 periods per second is common usage. The driving engine or turbine thus has to revolve at a rate of some multiple of 50 or 60, respectively, thus :—

Freq. = 50. N = 3000 - 1500 - 1000 - 750 - 600 - 375 - 300 - 250 - 200, etc.
Freq. = 60. N = 3600 - 1800 - 1200 - 900 - 720 - 450 - 360 - 300 - 240, etc.

The number of revolutions for A.C. motors can also be derived from (78), but an electrical slip of about 5 per cent. has to be allowed for. By changing the frequency of the alternator, the speed of all the connected motors will change correspondingly.

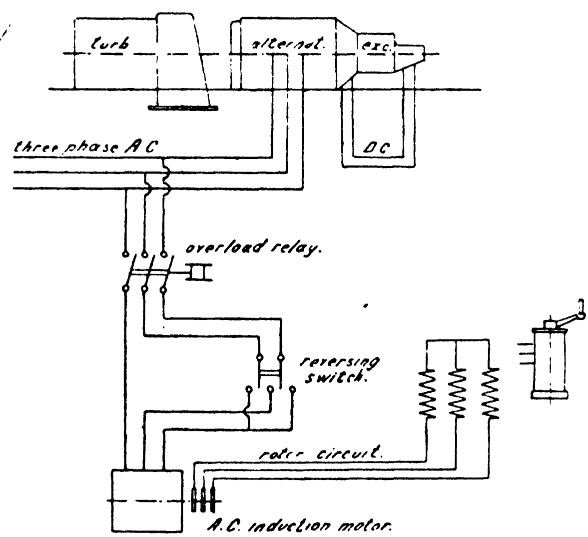


Fig. 267.—Alternating Current Scheme.

The *Slipring A.C. Motor* is used for mill drives ; the current is supplied to the stator windings, whereas the rotor windings receive induced current by rotation within the stator field. These motors therefore are called *Induction Motors.* The induced current of the rotor is led through three sliprings to an oil-immersed controller with resistances, so the starting current will be reduced below normal operating current strength and a good starting torque obtained. As soon as the motor has attained full speed, the rotor windings are short-circuited and the motor can run without the sliprings, which have only the definite object of starting the motor.

Speed regulation of the slipring motor is possible by applying resistances in the rotor circuit, which means a loss of power, as the current in the resistance is merely converted into heat. These resistances will reach large dimensions and can be used for short periods only and moreover the resistance has to vary with the load, a smaller load requiring a larger resistance for constant speed. This regulation, therefore, is not an economic nor a practical proposition ; the A.C. motors are normally run at their full speeds corresponding with the frequency. Moreover, any speed regulation is only downwards.

The *Variable Frequency System* changes the turbine speed and thus the frequency of the A.C. current, and is only for general regulation of the speed of all the connected motors.

The slipring motor has a good starting torque and is very reliable for continuous service, as there are no commutators but only sliprings.

A more ingenious A.C. motor is the *Squirrel Cage Motor*, where the current is applied to the stator windings and the induction current is produced in round bars, placed in slots at the periphery of the rotor. These bars are connected on both sides to copper rings, which rotate with the rotor. The squirrel cage motor, therefore, is the simplest A.C. motor existing, but it has the inherent disadvantage that the starting current is very high, up to 4 to 6·5 times the normal load current, and therefore it is only used for small units, which are switched direct on the main line.

For larger units star-delta controllers can be used, but the starting current will still remain three to five times the normal load current.

The common squirrel cage motor is thus quite unfit for mill drive, and for other medium and high power requirements; and to overcome this drawback, BOUCHEROT in 1898 invented a squirrel cage motor having two sets of bars in the rotor; the outer ones of small area have a large resistance and small reactance, whereas the inner ones have a large area for small resistance and high reactance. The combined effect of both has given the motor a greatly reduced torque-current characteristic. This Boucherot motor has since been improved and nowadays a special *Double Squirrel Cage* design is on the market, as shown in *Fig.* 268, the light copper or aluminium bars being embedded on the outside of the rotor armature and the heavier bars nearer the centre. This motor has a torque-current characteristic as good as that of the slipring motor and is started by an oil-immersed star-delta controller. It has replaced many a slipring motor and is now built up to units of 3000 h.p.

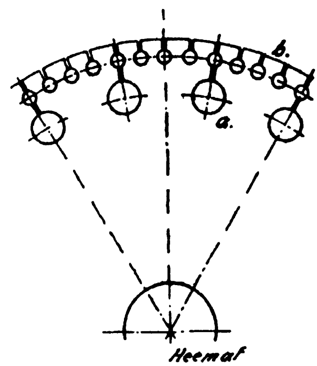


Fig. 268.—Special Double Squirrel Cage Motor.

The author has installed several of these motors and has found them to provide very good operating performance for carrier, centrifugal and pump drives and owing to their simple construction, they may be used as mill driving motors as well in the future.

All electric motors for sugar mill drives should be of the drip-free type, having special insulation for the tropics, where the moisture content of the air is higher than in more temperate zones and a temperature of 40 to 50°C. under full load not exceeded. All electric motors are cooled by an air current and dust will be sucked in and adhere to the coils, so should be removed at regular intervals by dry compressed air. Totally enclosed fan-cooled A.C. motors are not used for mill drives and the normal voltage is 440.

During the dead season the mill drive motors have to be well protected from moisture and before starting the crop it is good practice to run the motors 24 hours without load, so that they can acquire a temperature favourable for drying, thus removing any moisture that may have entered into the insulation of the coils.

A damp motor should not be started under load, as it will show a high current consumption (high amperage) and may even develop a short circuit, which would burn the motor windings and make costly re-winding a necessity.

The power output of the A.C. motor can be most conveniently measured by an A.C. wattmeter, or it can be derived from the average ammeter and voltmeter readings between each phase, from the formula :—

$$h.p. = \frac{I \times E \times \cos \phi}{746} \times \sqrt{3} \times \eta \quad \ldots\ldots\ldots\ldots\ldots \quad (78)$$

cos ϕ is called the *power factor* and is the ratio of the effective power in watts to the volt amperes. The power factor varies with the load and the kind of current consumers used, and in good installations will vary between 0·7 and 1·0. It has a bearing on the efficiency of the equipment but not in proportion to it, although a low power factor may cause a low efficiency.

CHAPTER X.

IMBIBITION AND MACERATION EQUIPMENT.

PRINCIPLES — IMBIBITION AND MACERATION APPLIANCES — IMBIBITION AND MACERATION PUMPS.

1.—Principles of Imbibition.

The purpose of imbibition and maceration is to exhaust the sugar left in the bagasse after the primary mill pressings. Even if all cane tissue cells have been ruptured by pressure or killed by heat, a 100 per cent. efficiency can never be obtained and the water content of the bagasse will contain sugar in solution ; so imbibition or maceration will reduce the amount of sugar in the bagasse down to the lowest practical limits—about 2·5 per cent. with imbibition, or 1·5 per cent. with maceration being the minimum now obtained.

By imbibition is understood the spraying on the bagasse of the imbibition agent, this being water, diluted juice of a subsequent mill or sweet-water from the filter-presses or other process work. Maceration is the drenching or soaking of the bagasse in the maceration liquid.

With imbibition or maceration there is only a process of sugar extraction by dilution, and osmotic action through the cell walls does not take place, due to the shortness in time of the leaching process, which has to be continuous like cane grinding itself.

Cold imbibition or maceration, therefore, will not extract any sugar from cane cells which have not been ruptured, whereas hot maceration will kill the protoplasm of the living cell at about 60°C. (140°F.) ; but to achieve this the imbibition liquid must have a temperature of about 85°C. (185°F.).

Cold or hot imbibition below this optimum temperature gives equal results, according to PRINSEN GEERLIGS, but hot maceration will extract a part of the reversible colloids from the bagasse, and this may cause additional work at the clarification station.

An exact mathematical synopsis of the maceration performance cannot be given, as all the cane tissue cells are not ruptured or killed, and the good effect of maceration or imbibition has to be judged from the lost juice in final bagasse, as explained in Chapter VII.

Hawaii has been noted for its most efficient application of imbibition for some decades past ; but nowadays due regard is paid to the merits of the process in all countries.

The quantity of water added, the dilution, amounts to from 10 to 35 per cent. on weight of cane, or between 100 to 350 per cent. water on dry fibre weight, which figures are to be taken as the practical limits. High imbibition necessitates a larger amount of water being evaporated and for economical factory operation this has to be done without the expense of additional fuel. A triple effect evaporation, therefore, will allow less imbibition than a quintuple effect evaporator. Moreover, factories with a good heat balance, having an excess of bagasse, can be more generous with the amount of imbibition water they use and thus have the benefit of less juice lost in bagasse. In any event, the most efficient degree of application of the imbibition water should be employed.

In many factories the quantity of imbibition water used is calculated from the Brix of the normal and diluted juice ; but a more exact figure can be obtained by measuring the imbibition water by means of a water meter or better by weighing it.

2.—Imbibition and Maceration Appliances.

Imbibition water can be applied very easily by means of a perforated pipe over the width of the intermediate carrier where it is desired, and it is good practice to have the water enter by a T-piece midway of the carrier width so that an even distribution may be obtained. The area of the perforations should be less than the pipe area, so that a certain degree of pressure can be maintained.

There are installations having two or three pipes of smaller diameter, so that the amount of water to be applied can be changed by cutting out one pipe, in which case the pressure in the others will not be impaired.

For imbibition, pure cold water is used or hot water from the boiler feed line or the last evaporating bodies. In most well operated sugar factories the boiler feed-water is kept at a temperature of about 95°C. (204°F.).

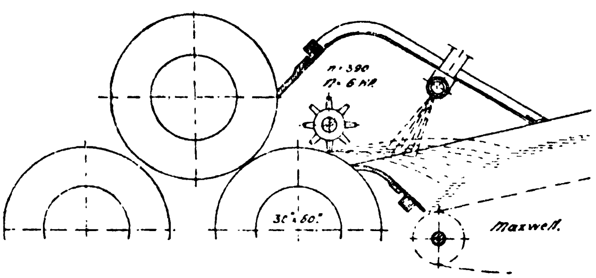


Fig. 269.—Revolving Macerator.

The perforations should be of about $\frac{1}{8}$ in. diam. to allow the water to be distributed in small jets, as larger jets might not be absorbed by the bagasse, and so would leak through the blanket.

A still better method is to use *spray nozzles* or atomizers so that the spray can be readily absorbed by the bagasse.

Of equal importance to a good distribution of imbibition water is effective penetration of the imbibition liquid into the bagasse. As dry fibre or bagasse will absorb five to ten times its weight in water, according to the fineness or degree of disintegration of the material, it will be obvious that the imbibition water, generally, is only absorbed by the top layers. In *Fig.* 269 is shown a *Revolving Macerator*, doing 390 r.p.m. which throws the bagasse in a finely divided state against and through an atomized jet of water, so as to achieve a thorough penetration.

A further method for good penetration is to compress the bagasse slightly after (or while) the maceration or imbibition water has been added. The RAMSAY macerator is composed of two scraper plates, between which the bagasse emerges from the mill under light pressure. In these scraper plates are recesses over the full width, where water is applied under pressure, thus penetrating into the bagasse.

The BURUNAT maceration roller aims at the same result by compressing the bagasse on the back roller, after it is released by the mill and the water has been sprayed on. The roller is driven from the top roll of the mill and is pressed down by springs.

The LUCE maceration carrier passes the bagasse between two carrier aprons, which move at the same speed, one below and the other on top of the bagasse. As the aprons approach each other to the exit, the bagasse is compressed and the penetration of the imbibition liquid is achieved.

Of British design is the *Maceration Bath Carrier* shown in *Fig.* 270 and used in many Australian sugar mills. It is a very efficient piece of apparatus as the bagasse is immersed in the maceration bath by means of a large revolving drum, so it cannot float but is pressed down into the liquid and thoroughly soaked.

After the bagasse leaves the bath on the inclined carrier slope, the superfluous moisture is drained off over a perforated double bottom. The bagasse is dragged along by wooden scraper slats, attached to two strands of detachable chain.

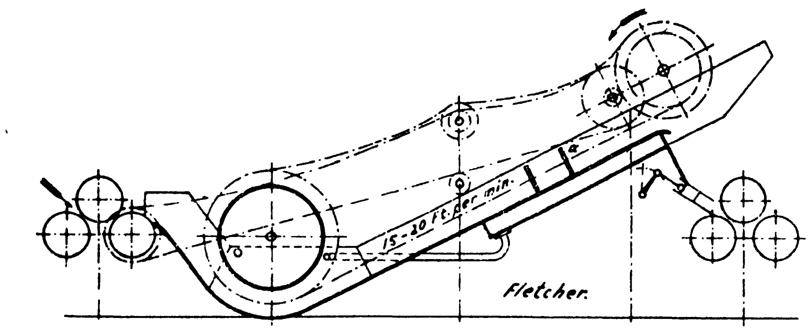


Fig. 270.—Maceration Bath Carrier.

The temperature of the bath can be kept as desired, although a high temperature will give rise to heavy mist.

As the soaked bagasse is difficult to feed to the next mill, a mechanical pushing arrangement is provided, which will force the feed into the mill. Scraper slat carriers or drag carriers have the inconvenience of delivering the feed by heaps and hence a good-sized feeding chute for the next mill has to be provided.

The sucrose extraction performance of the maceration bath carrier ranks high, and it can be applied in those instances where a sufficient centre distance —about 40 feet—between the consecutive mills exists. The mills must have good juice drainage for their front rollers, as the latter have to deal with a very large amount of juice.

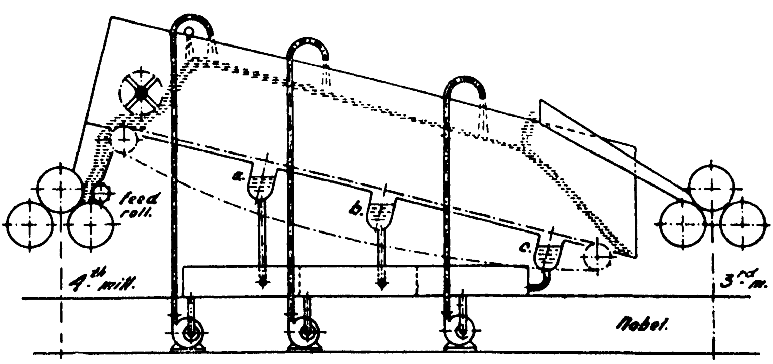


Fig. 271.—Drenching Maceration Carrier.

In Java several installations have been made of a *Drenching Maceration Carrier*, as shown in *Fig.* 271, these requiring also a large centre to centre distance of about 27 ft. between the mills, resulting in large mill houses. Generally, the maceration carrier is applied before the last mill only.

Very favourable results are reported of its performance and with a 12-roller tandem without crusher, the lost juice in bagasse has been reduced from 59 to 26 per 100 fibre, or about 1·5 per cent. sucrose in bagasse. The heat losses or steam consumption are high, and 10 per cent. of the total factory steam is sometimes required, which fact has militated against its general extension.[1]

As will be seen from the figure, the bagasse emerges from the previous mill between two scraper plates, well above the carrier end. Maceration water is applied at the rate of about 150 parts per 100 fibre at the upper end of the carrier and the liquid is drenched through the heavy bagasse layer from 3 to 5 ft. thick; and once it starts to leak through it is recovered in a receptacle *a* in the carrier bottom, which discharges into a tank below, whence it is pumped by a centrifugal pump of the unchokeable type and returned to the carrier at a lower spot. This cycle is repeated three times at *a*, *b* and *c*. The fourth mill also delivers its juice to the first tank below *a*. To prevent fermentation of the large amount of diluted juice in circulation, lime is added at the first cycle. The overflow from *a* is re-collected at *b*, and from *b* at *c*. From the tank below the latter, the excess juice is pumped for imbibition before the third mill.

The bagasse is subjected to the leaching action for about 20 minutes, and a large amount of it has to be stored on the carrier, which has therefore to be of substantial design.

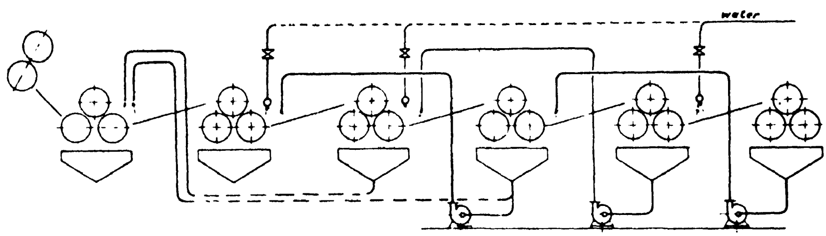


Fig. 272.—Compound Imbibition.

The power needed for driving the carrier and the three pumps amounts to about 25 h.p. for a 6-foot carrier and 1200 tons daily grinding capacity.

As the last bagasse contains only a small amount of sucrose, the roughening action of the juice on the mill rollers is less and a coarse roll material has to be used in the last rollers. The mill setting for drenched bagasse at the high temperature of about 85°C. (185°F.) has to be generous. A feed roller is set in front of the last mill, so as to dispose of the largest amount of diluted juice. The bagasse of the last mill is generally of low moisture content.

There is a difference of opinion as to how the imbibition water had best be applied. In Java the standard practice of *Compound Imbibition* is shown by the dotted lines in *Fig.* 272 for a four-mill tandem. The imbibition water is added after the second and third mills and this has the advantage that relatively small amounts of water are applied to each of the last mills, and no special drainage of the mill rollers is required. The diluted juice of the 3rd and 4th mills is pumped behind the first mill.

For a larger tandem, having six mills and one crusher, the author has installed a system of compound imbibition, as shown by the full lines in the same figure. All the imbibition water up to about 200 per cent. on fibre is applied after the 5th mill and the last mill juice is pumped unstrained, by means of an unchokeable pump, to behind the 4th mill, the 5th mill juice behind the

[1] See MAXWELL, "Modern Milling," p. 326.

3rd and the 4th mill juice behind the 2nd mill. Between the 1st and 2nd mills the cush-cush or trash from the main juice strainer and the secondary strainer is brought on to the carrier. This arrangement has given good results with a grinding capacity of about 2400 tons per 24 hours, the mills being 78 in. in size and about 2·4 per cent. sucrose in bagasse has been obtained, the moisture being around 50 per cent.

3.—Imbibition and Maceration Pumps.

The diluted juice for imbibition purposes has to be pumped from the mill from which it emerges to a previous one. This diluted juice is generally strained by a drag slat strainer and the trash delivered by a scroll on to the carrier. For this performance, any standard type of flywheel, duplex or centrifugal pump will be satisfactory, and a bronze pump body should be selected to avoid the corrosive action of the juice.

As the juice strainers may become a source of contamination of the juice, and also to avoid wear on the perforated sheets, scraper slats and chains, they are omitted in modern installations and the juice with the contained trash is pumped by *Unchokeable Centrifugal Pumps* to the previous mills. In *Fig.* 273 is shown the impeller of such an unchokeable pump as designed by the author and of which many have been put into satisfactory operation. These pumps are also called clogless or chokeless pumps.

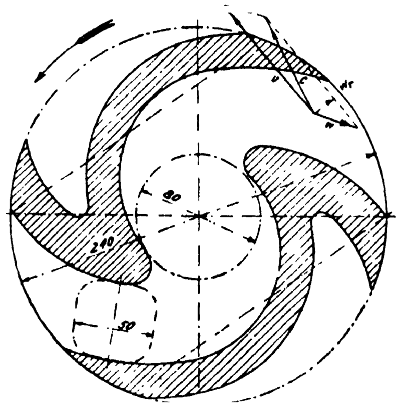


Fig. 273.—Impeller of Unchokeable Centrifugal Pump.

The centrifugal pump creates an increasing velocity of the liquid towards the discharge periphery of the impeller, so as to convert the thus created kinetic energy into potential energy or pressure, With the unchokeable pump the area through which the liquid has to pass cannot be reduced when the periphery of the impeller is approached, otherwise choking will occur, the trash being pumped with the liquid, so this forms an apparent contradiction in the design.

It will be seen from *Fig.* 273 that the radial and discharge velocities on the outer periphery of the impeller, as shown in the diagram, are kept low. This, nevertheless, has a bearing on the efficiency of the pump, but fortunately the pump head is generally small and the power input low too, hence the reduced efficiency for direct connexion with high speed motors may be neglected.

The impeller is of a closed design, as an open impeller might get the trash caught between the impeller and the pressure plate, after wear has taken place, and a braking action would thus be established. The passage in the impeller is indicated by the dotted square and pieces of trash up to 50 mm. (2 in.) will freely pass. The suction diameter is 80 mm. ($3\frac{1}{8}$ in.).

In *Fig.* 274 a sectional view is given of such an *Unchokeable Pump* for imbibition purposes, as supplied by the author. The pump casing as well as the impeller are entirely of bronze, whereas the shaft is of stainless material. The latter is supported in a bronze bushing next to the impeller and a splash ring is mounted on the shaft, so as to prevent eventual leakage of juice into the roller bearings. These pumps are made with one or two sets of roller bearings, and they can be driven by a belt or by an electric motor.

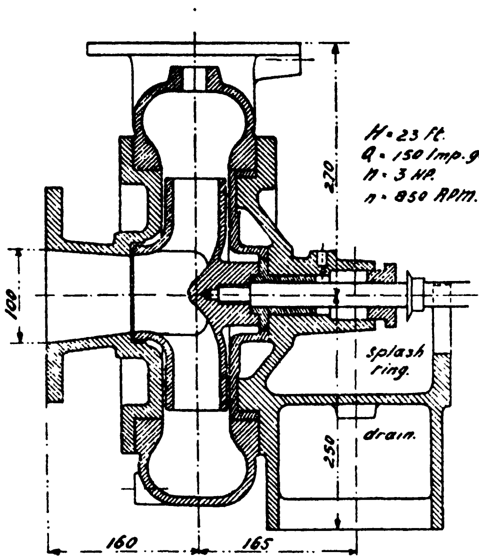


Fig. 274.—Sectional View of Unchokeable Pump.

Special care has to be taken that fine trash cannot enter around the shaft, as it will cause considerable wear in the bushing. The shaft is lubricated through a grease cup.

These pumps have worked very satisfactorily and large amounts of trash, up to about 20 per cent. of the volume, can be easily handled. A free passage of 2 in. × 2 in. is sufficient for imbibition purposes, as the cane is already disintegrated and the trash thus sufficiently reduced in size. It is of importance that the pumps have an unobstructed suction so that juice and trash combined will easily flow to the pump.

Valves in the discharge or suction lines cannot be allowed and the author has even had to remove straightway valves. The discharge pipe should only

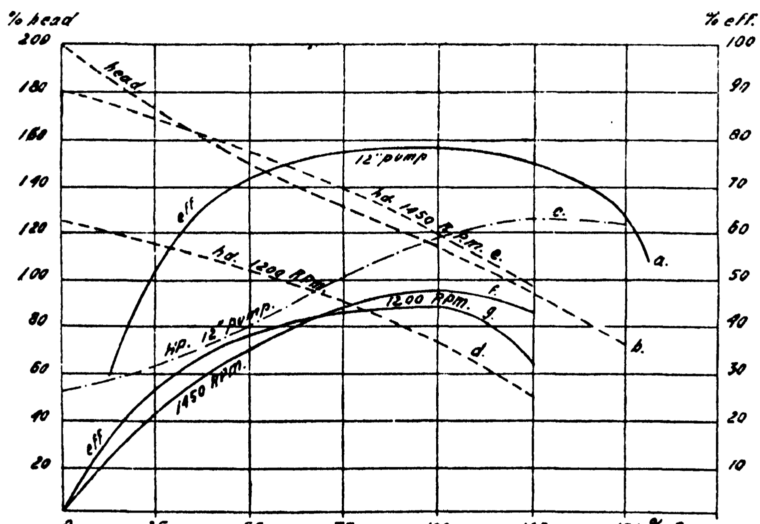


Fig. 275.—Pump Characteristics.

have long radius bends and be of a larger size than the pump discharge. Copper pipes have given a very good performance.

The wear on the impeller and the casing is of course heavier than when a pure liquid has to be pumped, since the trash has an abrasive effect.

As the pump flow cannot be regulated by valves, a variable speed motor for D.C. or A.C. current can be used to advantage in some instances.

The *Pump Characteristics* for unchokeable pumps are shown in *Fig.* 275, giving the head, output and efficiency under different conditions. The curve *a* represents the efficiency for a 12 in. sewage pump, whereas *b* and *c* are respectively the head and power input curves for this same slow speed pump.

The curves *e* and *d* represent the head for a 3 in. imbibition pump at 1450 and 1200 r.p.m. respectively, whereas *f* and *g* represent the efficiency for the same number of revolutions.

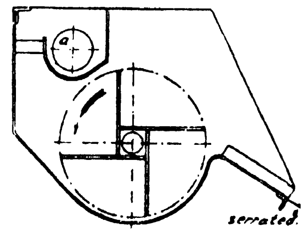


Fig. 276.—Distribution Trough for Unstrained Maceration.

A mean efficiency of a little below 50 per cent. is practically the best that can be obtained.

The distribution of the imbibition juice with trash over the carrier has caused difficulties ere now, as an even distribution can be impaired by the floating trash. The author has tried out different designs, and the *Distribution Trough* for unstrained maceration juice, shown in *Fig.* 276, has given good operating performance. The juice enters at *a* and as the pump flow is irregular, due to the bits of trash carried along, it first enters an equalizing gutter from which it overflows into the main trough. The gutter has welded strips of flat iron, to connect it to the main trough.

The juice flows down the slanting chute of the main trough and the floating bagasse is pushed by revolving flat irons, 1 in. × ½ in., welded in a spiral form 2 in. apart around the shaft.

The equalizing gutter has its edge cut at the end opposite to the feeding pipe, and the shaft revolves at about twice the r.p.m. of the driving mill roller.

Another arrangement is a fan-like discharge plate, which is placed in a slanting position, the juice entering at the top under a cover plate.

Distributing troughs with a revolving drum, having straight ridges longitudinally arranged, have also given good results.

CHAPTER XI.

JUICE STRAINERS.

PURPOSE—MILL OR COARSE STRAINERS—SECONDARY OR FINE STRAINERS.

1.—Purpose.

The juice extracted by the mills has a proportion of trash in it and sometimes includes small pieces of cane which have fallen through the clearance of the chute bottom plates, the trash knife clearances, or else are scraped from the Messchaert grooves, the latter producing a considerable amount of fine trash.

With the use of revolving knives or shredders, the amount of fine bagacillo in the juice is also increased and will be drained with the latter.

Formerly, the juice had to pass through piston pumps, these being the sole type of pump then employed, and it is obvious that any trash would cause trouble as it would stick to the ribs of valves or the seats, or between the valve discs and the seats, and thus prevent the pump functioning properly. Even the standard centrifugal pump of later design will choke with trash, as the passages of the impeller contract towards the periphery. Hence, as a logical consequence, straining of the mill juices becomes a necessity.

With the use of unchokeable pumps, as dealt with in the previous Chapter, the mill strainer may well become obsolete, only a coarse rake being necessary for removing whole pieces of cane, which possibly could not pass any pump or along any pipeline.

But the trash includes a part of the rind, containing the yellow pigment saccharetin, and other colouring matter,[1] so with less trash there will be less colouring matter in the juice. The heating of the alkaline juice will cause decomposition of the cellulose, and this has to be removed at the clarification station or it will become a melassigenic agent in the massecuites.

Moreover, the tubes of the heaters and evaporators will form incrustations at an increased rate, so the trash should be removed for efficient factory operation, as this will result in a better manufacturing process and a better product.

Formerly, fine straining was done after the clarification, but it is better that it should be done in the earlier stages of the process, thus immediately after the juice leaves the mills. Moreover, straining the hot clarified juice will produce a considerable fall in temperature and a corresponding loss of heat, which should be avoided.

The settling of juice without "bagacillo" has sometimes occasioned difficulties in the settling tanks in raw sugar factories, but when this occurs a small amount of superphosphate (1 lb. per 10 tons of cane ground) will produce a good defecation.[2]

Filter-presses will not work properly when fine bagacillo is not present, and this has induced many a raw sugar factory in Cuba to eliminate the filter-press station completely by returning the mud of the settling tanks to the

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1 See HARLOFF & SCHMIDT, "Plantation White Sugar Manufacture," pp. 11-12 and 14.

2 See the article of J. C. GONZALEZ, 4th Conference of the Association of Cuban Sugar Technologists, 1930.

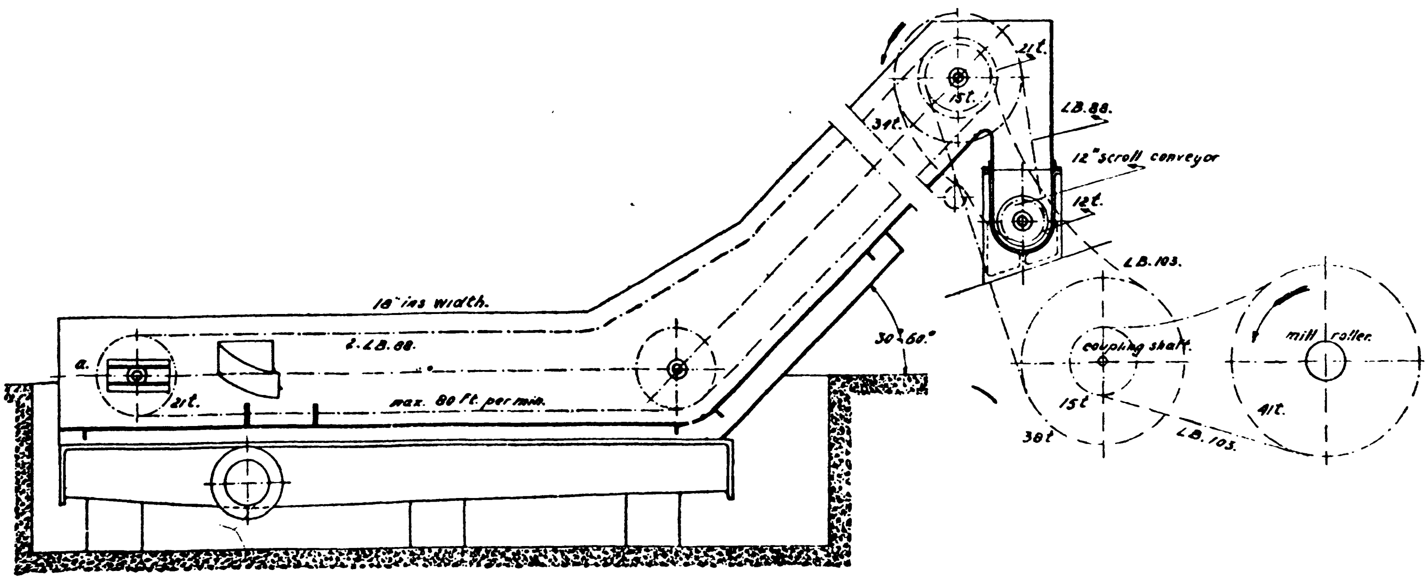

Fig. 277.—Drag Carrier Juice Strainer.

bagasse blanket of the intermediate carriers. The author knows one instance where the filter-presses have been taken completely out of the factory, giving corresponding economy in maintenance and operating expenses of the filter-press station.

2.—Mill or Coarse Strainers.

The juice leaves the juice pans or trays underneath the mills along cast iron, galvanized sheet iron, or preferably copper juice gutters, having a slope of about 1 per cent. for proper gravitation. These gutters generally discharge into a *Drag Carrier Juice Strainer* (often termed a "Cush-Cush" or Trash Strainer and Elevator), the common type of which is shown in *Fig.* 277, as noted in an existing installation by the author.

The juice is discharged by a spreader gutter in the direction of the carrier travel, close to the left end, so that it will fall upon the perforated brass plate, the juice being drained and the trash scraped off by the passing wooden scrapers attached to the carrier chains at regular intervals of about 18 in. to 24 in.

The trash will drain over the horizontal perforated plate and also on to the inclined part of the trash elevator and will discharge into a scroll conveyor, arranged at right angles to the strainer axis.

The sizes of the chains and sprockets are stated in the drawing, and as the drive is from a mill roller, a friction clutch is mounted on the intermediate shaft, so that the strainer can be stopped even when the mill is running. It is, nevertheless, better practice for medium and large installations to have the strainer driven independently by a small engine or electric motor.

Two strands of drag chain are provided, on which are fitted wooden scraper slats about ½ in. to ¾ in. thick, the width of the strainer bein· from 18 in. to 30 in. according to the standard width of the perforated plates. The chains suffer heavily from corrosion through the acid juice and also from the grit carried along with the trash ; and the author has known cases where malleable iron chains of the detachable type lasted only for one crop. Some manufacturers now make these chains also of bronze or a non-corrosive and hard iron alloy, with greatly improved wearing resistance.

The straining area necessary depends on several factors, as follows :—

1.—Size of perforations.

2.—Speed of carrier slats.

3.—Increased amount of trash through shredders, Messchaert grooves, etc.

For normal straining, i.e., for juice going to the factory and not used for imbibition purposes, about 1 square foot straining area per ton of cane ground per hour is considered normal practice, but the author knows several instances where 1 square foot had to take care of three to five tons cane ground hourly.

The chain speed ranges from 50 to 100 ft. per minute and a small-sized strainer can be improved in most instances by a higher chain speed.

The perforations of the strainer plates vary from 100 to 625 per square inch, but it should be recollected that very small perforations are easily clogged and special arrangements have to be made to clean these perforated plates regularly for continuous service.

The free area of the perforations is only a small percentage of the total straining area, as may be learned from the following table :—

With	625	perforations per sq. in.	0·020 in.	dia.,	free area	is	19·62%
„	400	„ „	0·027 in.	„	„		22·88%
„	225	„ „	0·045 in.	„	„		35·77%
„	144	„ „	0·057 in.	„	„		36·73%
„	100	„ „	0·063 in.	„	„		31·17%
„	64	„ „	0·085 in.	„	„		36·33%

Where no secondary straining is provided, 144 to 225 holes should be the lowest allowable limit.

Assuming a hydrostatic head of only 0·04 ft., the effluent velocity will amount to : $\sqrt{2\,g\,h} = 1{\cdot}6$ ft. per sec., g being the gravity acceleration = 32·1 ft./sec.^{2}

The contraction of the juice flow through these perforations is considerable and obstruction of these latter will take place owing to the presence of the gums commonly found in all mill juices ; the author has taken the coefficient C_c for combined contraction and obstruction as 0·08 for perforations over 0·05 in. whereas for finer perforations this value should be reduced to about half.

The juice flow through the strainer plate can thus be written :—

$$Q_s = V_g \times C_c \times C_o \times 3600 \times 6{\cdot}24 \text{ Imp. gals./hr./sq. ft.} \quad (79)$$

where : Q_s = Flow rate in Imp. gals. per hour and per sq. ft. strainer area.

V_g = Gravity speed in ft./sec. (1·6 ft./sec. as above).

C_c = Combined contraction and obstruction coefficient (0·08 as above).

C_o = Coefficient of free perforation area (mean 0·3 (30%) as above).

and 6·24 Imp. gals. is the equivalent for 1 cub. ft. For U.S. gals. the value 6·24 should be changed to 7·48 and for litres to 28·3. One square foot strainer surface, therefore, would screen 863 Imp. gals. per hour, and considering that one ton of cane gives one ton of diluted juice ≃ 210 Imp. gals., a capacity of four tons cane per square foot will result per hour.

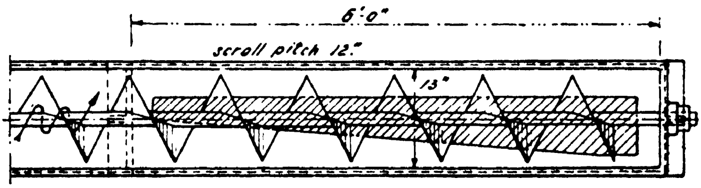

Fig. 278.—Trash Scroll Conveyor.

The perforated brass sheets have a thickness varying with the perforations, $\frac{1}{16}$ in. being the maximum for coarse strainers. These sheets have to be well supported by backing frames underneath, forming squares of 6 in. to 12 in. as otherwise the plates will bulge and the cleaning effect of the scraper slats be greatly impaired.

A tension device for the drag chains is provided at *a* and as the juice falls on a cast iron tray underneath, inspection and cleaning under the screen plates are difficult, since slime and fungi grow sometimes 6 in. thick under these plates, thereby becoming a centre of contamination for the juice. A good steaming device through perforated tubes is essential, but still better is it to have a gap between the strainer and the juice tray, to permit cleaning with a hot water hose at regular intervals, on top as well as underneath the strainer plates.

The TUINICU strainer has the strainer plates on bronze frames which can be taken out sidewards and thus be very easily replaced and cleaned.

In *Fig.* 278 is shown a *Trash Scroll Conveyor* designed by the author, the trough being made of cast iron and the scroll of welded sheet iron, having

1 in. total clearance in the former. The opening for distribution of the trash over the carrier width is not located centrally but to one side, as the trash will be carried on the moving side of the scroll and not centrally over the trough bottom. At the end a free discharge is provided. Sometimes an adjustable plate is laid under the opening to equalize the trash distribution.

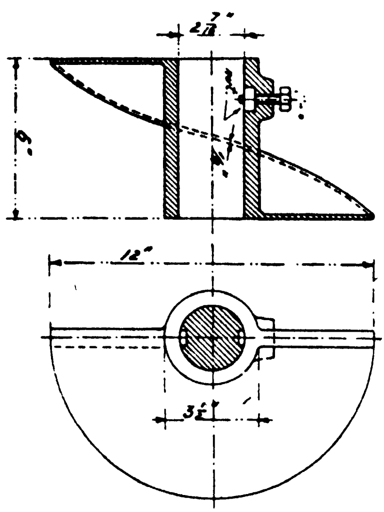

Fig. 279.—Bronze Scroll.

Instead of a scroll, a drag conveyor with wooden slats has been provided, but this requires more space and must be driven at right angles to the mill axis.

The amount of trash generally is heavy when cane disintegrators are present and as juice is absorbed about five to ten times the dry fibre weight by the trash, imbibition cannot be applied in all instances on the carrier, where the trash is discharged.

In Cuba there are installations where the trash is discharged in front of the crusher, which is feasible in those cases where revolving knives are employed. The author also has made such an arrangement with good operating results.

The scroll is subject to heavy wear and corrosion, and *Bronze Scrolls,* as shown in *Fig.* 279, are now available. Each casting forms half the scroll pitch and they are mounted over the shaft, being fastened by a set screw with a countersunk nut.

The direction of movement of the trash and that of rotation of the shaft are the factors deciding if a right or a left hand scroll has to be provided.

Instead of perforated plates, the author has supplied *Split Sieves* as shown in *Fig.* 280, made of brass or monel metal, having openings of 0·02 in. (0·5 mm.). The sieves are made of specially wound and stamped wires with internodes at 2¾ in. distance, in which ½ in. brass pins are riveted, so as to form slabs of the required width. At ⅝ in. distance, little lugs are stamped for maintaining the correct openings. The experience has been that these sieves will outlast several perforated sheets, the straining surface always being flat and not bulging, but they sometimes clog and so have been discarded; but it is interesting to know that in one mill, where the author has supplied these sieves, they were kept in satisfactory operation, doubtless due to the perseverance of the chief engineer.

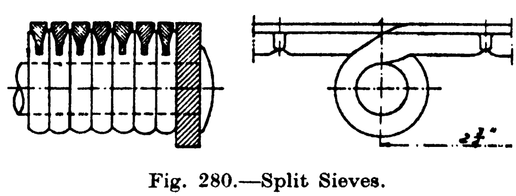

Fig. 280.—Split Sieves.

For use with unchokeable pumps and to strain out the large particles of cane, thus omitting the coarse strainer, these split sieves can be used to advantage, with openings of about ¼ in.

In Java *Grasshopper Strainers* are used and are reported to give good operating performance, the principle being the same as applied to sugar conveying. The trash removal is not so rapid as with scraping slats, and additional area has to be provided.

3.—Secondary or Fine Strainers.

In most cases hitherto, the juice has traversed a secondary strainer after having passed through the coarse primary strainer, but in the future these primary strainers may be eliminated by employing unchokeable pumps.

The performance of the fine strainer is considerably more difficult than is the coarse straining, as the surface tension of sugar juice is higher than that of pure water and thus the contraction through the very fine mesh of the straining wire gauze is considerable. Moreover, the gummy or colloidal matter in the juice may stick to the gauze and thus obstruct the passage of the juice.

For fine straining perforated sheets are not used, as the smallest perforations known to the author are 0·020 in. in diameter, having only 625 perforations per square inch, whereas wire gauze of 40, 60 80 or 100 mesh or more, measured to a linear inch, has respectively 1600, 3600, 6400 and 10,000 openings per square inch, from 0·015 to 0·0055 in. square, thus being of considerably more efficient straining performance.

The free area for juice passage is as follows :—

40 mesh,	wire diameter	0·01000 in.	40·80 per cent.
60 ,,	,,	0·00800 in.	33·87 ,,
80 ,,	,,	0·00575 in.	29·59 ,,
100 ,,	,,	0·00450 in.	30·25 ,,

The material for these wire gauzes is preferably phosphor bronze or monel metal, the latter being a natural alloy of copper and nickel, found as a mineral in U.S.A. These metals have a corrosion-resisting capacity, the monel metal being the better.

Being of small load resistance, the fine wire gauze has to be supported by a backing sheet of greater strength. The use of coarse 4-mesh wire gauze has not given good results, as the bearing is unequal and the fine wires wear rapidly over the coarser ones. Copper plates having 25 perforations of 0·150 in. diameter per sq. in. have given very good results, as well as the *Square Perforated Backing Plates*, as shown in *Fig.* 285, having $\frac{3}{8}$ in. square holes on $\frac{1}{2}$ in. pitch.

These perforated plates give a free passage as follows :—

25 round holes per sq. in.	44 per cent.
4 square holes per sq. in.	56 ,,

Fine strainers, therefore, have the juice passage reduced first by the copper backing sheets and then by the fine wires, the total free passage being about 15 per cent. of the straining area.

The contraction and obstruction of the juice flow through gauze is considerably higher than with perforated sheets, and in case a sufficient hydrostatic head is not available a gentle pressure is advisable, if only to overcome the frictional resistance, as otherwise very large straining areas are required. Moreover, the area must be dimensioned according to the easy or retarded removal of the trash, and in case of revolving strainers, to the peripheral speed.

In *Fig.* 281 a *Vibrating Fine Juice Strainer* is shown, which has been furnished to several Cuban mills for secondary straining. The straining surface is only 15 sq. ft. and the author has seen 550 Imp. gals./hour per sq. ft. gross strainer area being passed through such a strainer, having 40-mesh wire gauze.

As fine wire gauze when supported by coarse mesh wire gauze has shown heavy wear, a better performance might be obtained by using the above-mentioned perforated copper sheets as a backing.

The strainer is operated by a vibrator, based on the principle of unbalanced centrifugal force, as shown in *Fig.* 282. In a cast iron cylinder mounted in a spherical centre bearing, lined with white metal, the stator windings of an electric motor are attached. The rotor is supported in ball bearings, which

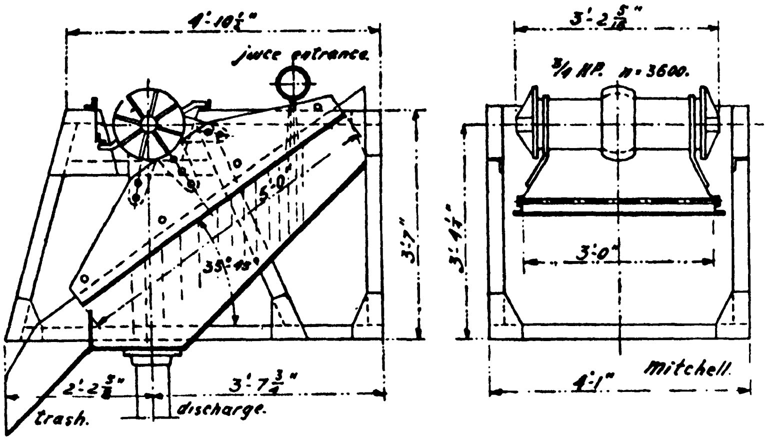

Fig. 281.—Vibrating Fine Juice Strainer.

are firmly connected to this cylinder and the long shaft carries on its ends ball cages (one ball of 1½ in. to 2 in. diameter in each casing) diametrically opposed to each other. As soon as the motor starts to rotate, the centrifugal forces will cause a vibrating action of the cylinder. The latter is firmly connected to the strainer trough, which thus participates in the same vibrations. According as the frequency of the A.C. is 50 or 60, there will be 3000 or 3600 vibrations per minute.

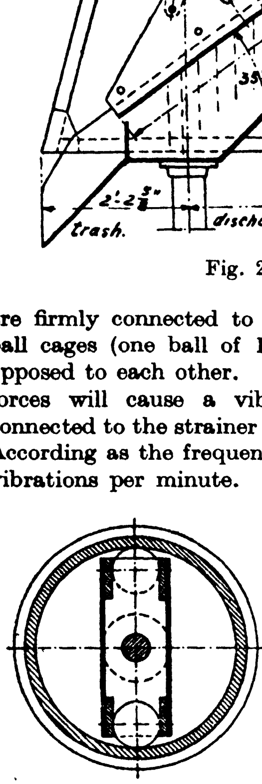

Fig. 282.—Unbalanced Centrifugal Force Vibrator.

The *Electro-magnetic Vibrating Strainer* has an electro-magnet for causing a rapid reciprocal movement. The magneto is connected to the wire screen, which is held in "drumhead tension." The aim of this design is to minimize the strain on the fine wire gauze and the author has seen good straining performance with this type, the power input being 0·75 kw. for a 3 ft. × 8 ft. strainer.

Another type of vibrating screen is the *Grasshopper Short Stroke Strainer*, working at 1200 r.p.m., which gives a more gentle motion to the wire gauze area. All the above strainers have an inclination of about 35° and the trash is discharged at the lower end.

The *plansifter* device as used for sifting flour in the flour mills has also been adapted for juice straining and a favourable riddling action is obtained, the sieves needing only a slight inclination.

As a variation from vibrating action, and in order to guard the fine wire gauze against heavy wear, *Revolving Fine Juice Strainers* have now obtained an important vogue. For coarse straining the revolving strainer of hexagonal pyramidal shape, having a horizontal axis, has been in use for a long while, but in Hawaii the drum type gained a footing about 15 years ago for fine straining.

In *Fig.* 283 is shown the *Flushing Type of Fine Juice Strainer* as designed in Hawaii; it has been applied in many installations in that country as well as abroad.

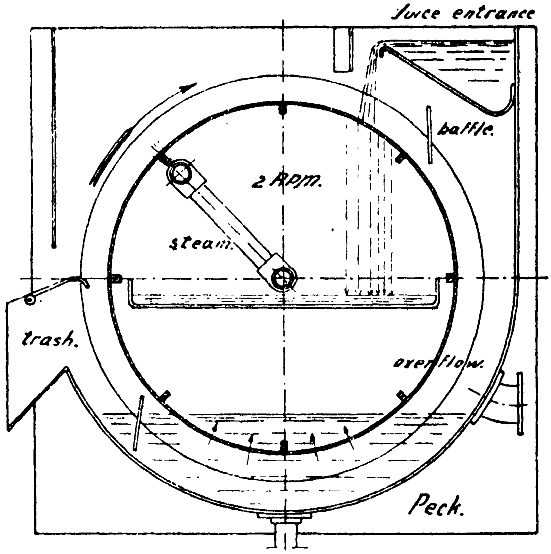

Fig. 283.—Flushing Type of the Fine Juice Strainer.

The juice enters a trough at the upper right hand side and flushes the drum which is lined with fine wire gauze, the "bagacillo" remaining on the outside and the clear juice falling through on a flat tray, located at a short distance below the drum centre, so that the central pipe may clear it. After one revolution is nearly completed, the bagacillo is thrown off by a steam jet from a perforated pipe arranged within the drum.

Any trash or bagacillo falling off into the surrounding casing is scraped by two baffles, which revolve with the drum, and is conveyed to the trash gutter, a hinged scraper being used for cleaning the baffles. In case any clogging of the wire gauze should occur, the casing around the lower half of the drum gets filled by juice and a small hydrostatic pressure will force this juice inside the drum where it is readily directed to the screened juice discharge.

To prevent flooding, an overflow outlet is attached to the casing which discharges into the tank for unstrained juice.

The strainer is about 6 ft. in diameter and is cleaned with muriatic acid solution. The capacity for continuous service is about 150 Imp. gals. of juice per hour per sq. ft. straining area, using 60 to 100-mesh wire gauze.

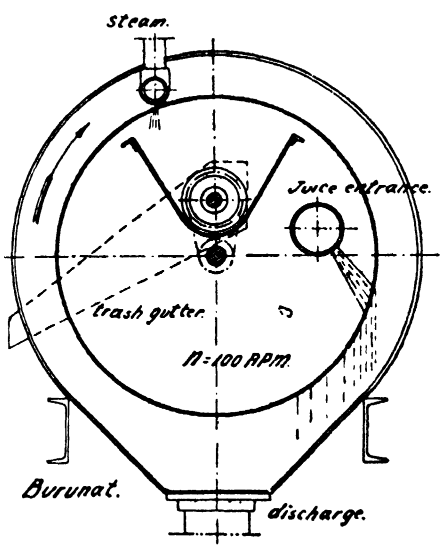

Fig. 284.—Inside Charging Revolving Drum Strainer.

Another strainer of the revolving type, also using a steam jet for sweeping off the bagacillo, is the *Inside Charging Revolving Drum Strainer*, shown in *Fig.* 284, which has found several applications in Cuba since 1928. The author saw the first strainer of this type, made from an old centrifugal, 30 in. × 18 in., having the shaft horizontally arranged.

The drum is closed at one end, the other end having a rim about 3 in. high, whereas the wire gauze is kept between two perforated plates of iron, having similar perforations as in *Fig.* 285, but of larger size. The juice enters the drum casing unilaterally and the steam jet is applied on the outside of the drum, using reduced live steam of about 35 lbs. gauge pressure.

As the speed is about 100 r.p.m. and the larger size strainers are about 4 ft. in diam., a centrifugal force of 6 lbs. for each lb. of juice and trash exerts a gentle pressure on the wire gauze, which pressure the steam jet has to counteract.

The drum with the wire gauze is enclosed in a sheet iron casing, which has to be dismantled when replacing the screens.

In case the screen clogs, the juice will overflow the rim on the drum and thus may escape unscreened.

The trash falls on a small scroll conveyor, driven by the main shaft through a chain reduction gear. The bottom part is perforated and the trash is delivered with its natural moisture content, i.e., about 10 : 1 on dry matter.

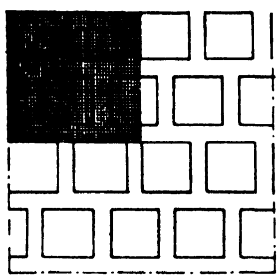

Fig. 285.—Backing Plate.

For cleaning, the discharge pipe is closed and the casing filled with a solution of caustic soda, so as to immerse the lower part of the drum, and is then rotated for a few hours. This cleaning has to be done about every six or nine hours and a spare strainer needs to be at hand, so that they can be used alternately.

As a consequence of the action of the steam jet, the wire gauze suffers, and carbonization or hardening of the gummy matter cannot always be avoided.

Due to its low initial cost, this strainer has found application in several mills.

To avoid the incrustating effect caused by the steam jet, the author patented, and put into operation in 1928, an *Immersing Rotary Strainer* as shown in *Fig.* 286, which has since been installed in several sugar factories in Cuba and Mexico.

The drum, which is 4 to 6 feet in diameter, is provided with six or eight easily removable sections carrying fine wire gauze and backing sheets, and it is mounted in two large journals or collars attached to the tank walls, so that the trash conveyor is self-supporting outside the drum. The juice entrance is bilateral, and a special construction provides for juice pressure regulation and non-clogging operation of the juice charging pipe. The number of revolutions of the drum is kept to such limits that no appreciable centrifugal forces will act on the trash.

The lower part of the drum immerses in the juice, whose level can be kept at will by an adjustable overflow during operation of the strainer, so the outside pressure will float the bagacillo on the inside of the drum. It is now easily caught by the inside perforated baffles and by means of the revolving drum is deposited in the centrally arranged trash conveyor, which has a perforated bottom, and leads outside the drum. The trash can be discharged through a tube of about 6 in. diameter, having a slope of 1 in 4.

At the right hand side is a water pipe *a* having perforations or nozzles for spraying hot water under pressure (boiler feed-water) for periodical cleaning purposes. Sometimes a spray is also provided for disinfection with formaldehyde. In case the wire gauze should clog, the drum fills with juice until it overflows through the large journals, which are provided with drip pans, so ready warning is given. The hot water spray will dislodge the adhering trash in a few revolutions of the drum and restore the normal functioning of the strainer.

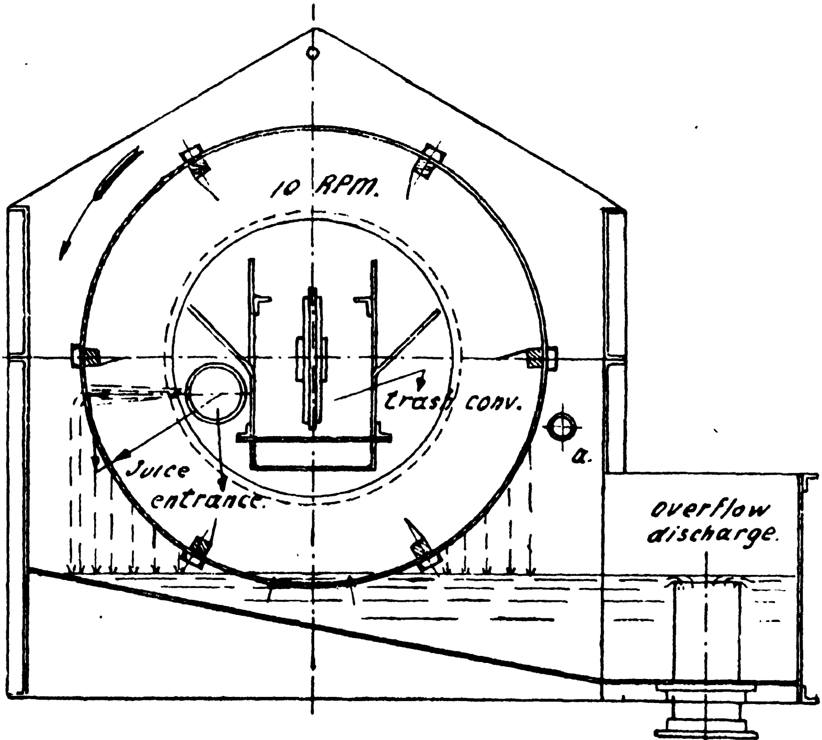

Fig. 286.—Tromp's Immersing Rotary Strainer.

The frames with the fine wire gauze can be changed in about 15 minutes and the author has put this strainer in continuous operation for 21 days, without having need to change the wire gauze frames. The cleaning of the latter is done in a tank with caustic soda and afterwards washing in clean water. The author has instances on record where one wire gauze set has lasted nearly two crops, so a very low maintenance charge is ensured.

The trash conveyor is driven by bevel gear from a side shaft, which drives also the drum by means of a detachable chain.

A capacity of 200 Imp. gals. of juice per sq. ft./hr. has been attained. The straining area is about two to three times as large as the previously-mentioned strainer, and maintenance cost is thus greatly reduced.

CHAPTER XII.

STEAM BOILERS.

STEAM GENERATION—FUELS USED—COMBUSTION—FURNACES—TYPES OF BOILERS—GENERAL BOILER DATA.

The steam generating plant of a sugar factory is of paramount importance, as on its efficiency may depend the economic heat balance, where any liability generally causes outlay for additional fuel.

1.—Steam Generation.

Water is transformed into steam, after the necessary heat energy has been supplied for the change into the third aggregate state. A three-fold amount of heat is required for this transformation :—

(*a*) Heat for raising the temperature of the water to the boiling point.

(*b*) Latent heat for transforming water at the boiling point into steam.

(*c*) Heat necessary for the mechanical performance of increased volume, as steam will occupy more space than the water whence it has been produced.

The total heat available in the steam is the sum of these three quantities, but it should be recollected that the liquid heat (*a*) is not always recovered in heating bodies, as the condensation water will leave with a high temperature.

The specific volume of water is lowest at 4°C. or 39·2°F., whereas at higher temperatures it is expanding and the specific volume thus increases, up to 1·39 for 294°C. (560°F.).

An Imp. gallon of water weighs 10 lbs. at 62°F. and occupies 277·4 cub. in., whereas a U.S. gallon of water weighs 8·3 lbs., occupying a volume of 231 cub in. The metric system is the most practical, 1 litre or 1 cub. dm. of water weighing 1 kilogram at 4°C.

Under an atmospheric pressure of 760 mm. (29·92 in.) mercury, the boiling point is reached at 100°C. or 212°F., whereas the freezing point is at 0°C. or 32°F. Under a lower pressure than the atmospheric one, the boiling point will be reached at a lower temperature, and a higher temperature is required for pressures above the atmospheric one, as may be found in the steam tables of any engineering handbook.

The specific heat of water changes very little with the rise of temperature and may be taken as being unity for overall calculations, thus one *British Thermal Unit* (B.Th.U.) is required to raise one pound of water one degree Fahrenheit. In the metric system, a rise of 1°C. per 1 kg. of water will require one calory.

Steam is considered *saturated* when, on being brought in contact with water of the same temperature as the steam, there is no evaporation of this water.

From the steam tables for saturated steam is taken :—

Gauge Press. lbs./sq. in.	Temp. °F.	Heat of Liquid B.Th.U.	Total Heat of Steam B.Th.U.	Spec. Vol. cu. ft. per lb.
0	212	180	1150·4	26·79
100	337·9	308·8	1188·8	3·89
150	365·9	338·1	1195·0	2·758
200	387·9	361·3	1199·2	2·141

from which it will be gathered that the latent heat is far in excess of the sensible heat and heavy heat losses are caused by loss of steam.

In the metric system, there is a handy formula from REGNAULT for calculating the total amount of heat available in the steam, when the temperature t is known in °C., thus :—

$$Q = 606{\cdot}5 + 0{\cdot}305\,t \quad \ldots\ldots\ldots\ldots \quad (80)$$

Q being measured in calories per kg., equal to 1·8 B.Th.U. per lb.

Superheated Steam is understood as steam at a higher temperature than saturated steam at the same pressure ; thus when brought in contact with water of saturation temperature, evaporation will continue until the steam becomes saturated again. Superheated steam therefore is moisture-free and for sugar-house practice a reasonable superheat is advisable, as it will supply dry steam to the prime movers and thus avoid the danger of water-hammer, and moreover a reduction in steam consumption results, as explained in Chapter IX.

The amount of additional heat to be supplied for superheating saturated steam depends on the specific heat of the steam, varying between 0·463 and 0·635 for pressures above the atmospheric, as employed in sugar-house practice.

Superheated steam occupies a higher volume than saturated steam of the same gauge pressure, and from the steam tables the following data are taken:

Gauge Press. lbs./sq. in.	Superheat °F	Temp. °F	Total Heat of Steam B.Th.U.	Spec. Vol. cub. ft. per lb.
0·3	50	263	1174·2	28·40
	100	313	1197·6	30·46
	200	413	1244·4	34·53
100·3	50	388·1	1216·9	4·20
	100	438·1	1243·1	4·51
	200	538·1	1293	5·09
150·3	50	416	1225·2	2·99
	100	466	1252	3·21
	200	566	1302·5	3·64
200·3	50	438	1231·6	2·33
	100	488	1259	2·51
	200	588	1309·7	2·84

For sugar-house practice 100°F. (55°C.) superheat may be taken as a fair average, as it does not need much heavier insulation of the pipelines than is needed for saturated steam and requires no special provisions at the steam engines, as the steam will arrive there nearly in a saturated state, and dry steam is thus obtained. Where electrification with a centralized power plant is considered, higher superheat can be freely employed, so there the exhaust steam will be at, or slightly above, the saturation temperature.

Moist Steam contains a certain percentage of water, generally dragged along from the boilers or caused by condensation in pipelines and engines, as well as by the expansion performance in the latter. The heat content of moist steam is less, as well as the specific volume, and, moreover, the steam lines have to be drained to avoid the danger of water-hammer. In addition, more water is required for the production of moist steam and as a great part of the moisture is drained by traps to the sewers, larger amounts of "make-up" water are required.

In normal sugar-house practice, where there is no superheating equipment, 3 to 5 per cent. moisture will be present in the live steam and 10 to 20 per cent. moisture in the exhaust steam, according to the heat drop experienced in delivering mechanical power in the steam cylinders.

For measuring the amount of moisture for percentages below 4 per cent., an apparatus has been designed, wrongly called a *Throttling Calorimeter*,[1] as shown in *Fig.* 287. The steam is taken from the main steam line at a point where there is no water dragged along at *a* and on passing a valve *b* is led to a T-piece c, having a thermometer bulb inserted. The thermometer at this spot will thus give the temperature of the live steam extracted at *a*. Good insulation of the apparatus is essential. At *d* is a pair of flanges, having a plate attached between, with an orifice of $\frac{1}{8}$ in. diameter and an elbow connects with another T-piece *e* (where another thermometer bulb is inserted), which connects to the free atmosphere without causing any back pressure. The thermometer bulbs are filled with oil.

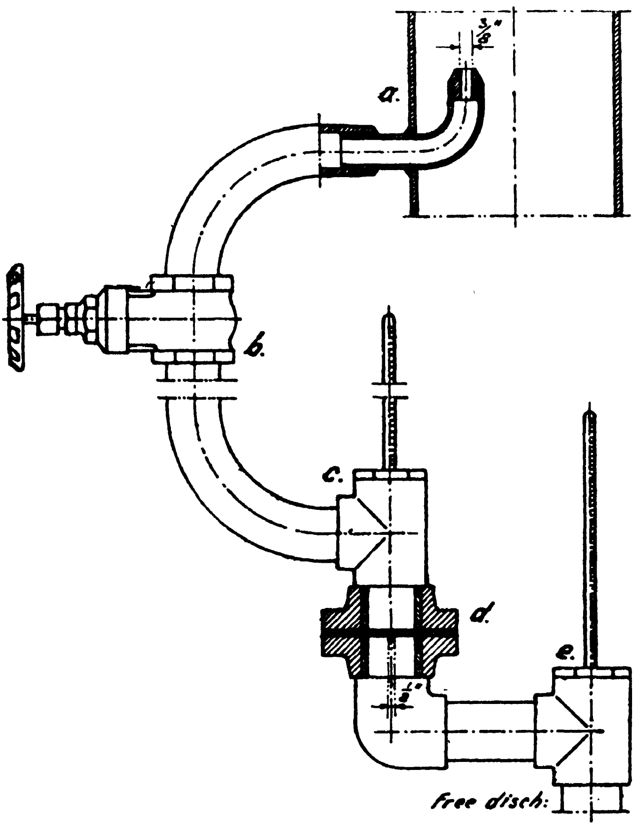

Fig. 287.—Throttling Calorimeter.

By passing through the orifice the steam will be superheated, as may be easily gathered from the fact that the total heat at e.g., 100 lbs. pressure, is higher than the one at 0 lbs. gauge pressure.

The specific heat of steam at 0 lbs. is equal to 0·47 and the temperature rise above 212°F. at *e* with the above-mentioned pressures will thus amount to : (1188·8 — 1150·4) ÷ 0·47 = 81°F. when dry steam is extracted. As soon as there is moisture in the steam, the latent heat freed will cause evaporation of this moisture and the temperature at *e* will thus drop. At 100 lbs. gauge

[1] See also "Steam," published by the Babcock & Wilcox Co., New York, 1930, page 123.

pressure, the latent heat is 888 B.Th.U. and for evaporating 1 per cent. moisture, 8·88 B.Th.U. are required and there remains a temperature rise of 38·4 — 8·88 = 29·52 divided by 0·47 or 62°F. above 212°.

For exhaust steam a separating calorimeter should be used before the throttling one is applied, as the moisture contents otherwise will be too high to be measured. For pressures above 100 lbs. the following table will be found useful :—

Gauge lbs./sq. in.	Temp. *c*.	Temperature *e* at a Moisture Content of: 0%	1%	2%	3%	4%
100	338	293	275	255	237	218
150	366	305	288	270	252	233
200	388	316	298	280	263	244

Obviously, there will be no readings below 212°, as the discharged steam might be already moist and thus the instrument would not give a true indication.

To compare the performance of different boilers under different pressures a uniform basis is applied in respect to British-American practice, so all steam figures are reduced to "from and at 212°F.," thus being at atmospheric pressure. Hence all steam figures have to be multiplied by an *evaporation factor* F., which is derived from :—

$$F = \frac{H - h}{970{\cdot}4} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (81)$$

where H is the total heat of the steam above 32°F. and h the liquid or sensible heat of the feed water, 970·4 being the latent heat in B.Th.U. for one lb. of steam at 212°F.

2.—Fuels used.

For cane sugar factories *bagasse* (in some places called *megass*) is the natural fuel, but additional fuels for firing under the factory or locomotive boilers are wood, coal and fuel oil.

Bagasse is composed of fibre, which contains the cellulose and other components of the cane stalk, as well as the sucrose left in the bagasse, the non-sugars (principally glucose), water and ash. The total heat value is the sum of the heat values of the components, reduced by the latent heat necessary to evaporate the water, and is found from the formula of PRINSEN GEERLIGS :

$$H = \frac{4750\,F + 3955\,S + 3750\,G - 540\,W}{100} \quad \ldots. \quad (82)$$

The total heat value is given in calories per kg. bagasse, whereas for British measures, the formula reads :—

$$H = \frac{8550\,F + 7119\,S + 6750\,G - 972\,W}{100} \quad \ldots. \quad (82a)$$

where F = percentage of fibre in bagasse.
S = ,, sucrose in bagasse.
G = ,, glucose (non-sugars) in bagasse.
W = ,, moisture.

Formula (82a) gives the heat value in B.Th.U. per lb. bagasse.

It has been found that there is only a slight difference in heat values of different kinds of bagasse when calculated on dry matter, and the deciding factor is the percentage of moisture.

The ash content is about 3 per cent., so $F + S + G + W$ is approximately 97 per cent.

From a given analysis of bagasse, the heat value calculated at 40 per cent. moisture has been shown to be 2585 cal./kg., whereas this same bagasse with 60 per cent. moisture only yields 1527 cal./kg., thus the former is 70 per cent. higher than the latter; but it should not be overlooked that the same amount of dry matter will produce 58 per cent. by weight more bagasse with the latter. The difference in absolute heat value is thus only about 7·5 per cent., but bagasse with 40 per cent. moisture will burn decidedly better than the same bagasse with 60 per cent. moisture. And with less latent heat of the moisture vapours the amount of heat carried up the chimney will be considerably reduced.

The specific weight of piled bagasse is about 10 to 15 lbs. per cub. ft., but loose bagasse will give only about 50 per cent. of these values.

Bagasse is fired in the same state as it has emerged from the last mill and any drying has to take place in the furnace. There it is obviously dried in a very efficient manner.

Actual tests have proved the utility of calculating the heat values. In Java, BOLK found that the actual heat value of an analysed bagasse amounted to 2150 cal./kg. with 43 per cent. moisture and 1350 cal./kg. with 60 per cent. moisture, both corresponding with the calculated figures. In Queensland an average of 8177 B.Th.U./lbs. on dry bagasse has been obtained.[1]

The composition of *wood* is similar to that of bagasse, the cellulose also being the most important component. Sugar is only available in certain kinds of trees to a very small extent, but wood generally contains resin, which adds to the ease of combustion. The heat value of wood does not differ greatly from that of bagasse, the principal factor being the moisture content.

Green wood just after being cut contains about 40 per cent. moisture, whereas air-dried wood only contains about 20 per cent. and the former ignites badly, as the moisture is driven to the surface as soon as it is heated. Green wood, therefore, is not very suitable for burning under the boilers.

Contrary to the belief of many an operating engineer, hard wood has about 10 per cent. less specific heat value than soft wood, but for burning under the boilers the volume of hard wood is considerably less than that of soft timber and it will last longer due to its compactness and lower moisture absorption, so hard wood is an ideal boiler fuel. Wood does not require any special features in the furnace design; a suitable firing door is the only thing to be borne in mind.

The specific heat value of different kinds of wood varies between 8300 and 9100 B.Th.U. per lb. on dry matter, but as fired the heat value lies between 4300 and 6600 B.Th.U. per lb. or 2400 to 3700 cal./kg. As an analysis of the firewood is not available at the sugar factory and sampling is not very feasible, an average heat value of 4500 B.Th.U. per lb. as fired can be taken as a conservative figure.

Hard wood gives about 1 per cent. ash and the weight if piled in logs of up to 10 in. dia. will be 30 lbs. per cub. ft., whereas soft wood weighs about 20 lbs. per cub. ft. (300 to 500 kg. per cub. metre).

Firewood is usually piled in the open air, as the hard wood employed in most instances will take up but little outside moisture, and the crop time normally falls within the dry season.

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1 See *Int. Sugar Jl.*, Sept. 1935, page 359.

Coal and coke are not used to any great extent for sugar factory boilers, as they will need special furnaces, which are not always at hand. Coal and bagasse can be fired alternately but not together, as the coal will have reached the ash cone or be discharged before it has been burnt completely. For locomotive boilers and in some cases for pump stations not close to the factory, coal can be used with advantage. Coke will prove of value, when it can be had at a reduced price as compared with coal.

Coal contains about 1 to 6 per cent. moisture and 2 to 20 per cent. volatile matter and the higher the latter percentage the longer will be the flame produced by burning, a point of special interest for sugar factory boilers, where a long path for the gases is generally provided. Moreover the volatile matter increases the heat value, which will vary between 11,000 and 14,000 B.Th.U. per lb. dry value. An analysis is generally to be had from the coal vendor.

As fired, 9000 to 13,000 B.Th.U. per lb. (5000 — 7500 cal./kg.) will be obtained. The ash content varies greatly, being between 3 and 20 per cent., but a fair average of 10 to 12 per cent. may be expected. Coal ash and cinders have value as material for road hardening, but these should be applied in the rainy season, as otherwise they will give rise to dust. For roads traversed by motor cars, it should be mixed with an asphalt road dressing.

The relation of weight to volume in the case of coal is between 63 and 75 lbs. per cub. ft. (1000 — 1200 kg. per cub. metre).

When stacking coal, it should be borne in mind that it will suffer from exposure to weather and may lose as much as 10 per cent. in heat value, especially when a high percentage of volatile matter is present. Moreover, when moisture has penetrated big piles of coal, spontaneous ignition may arise, although the author has no record of coal fires at sugar factories; the ignition temperature is around 870°F. (465°C.).

Coke has an average heat value of 12,500 B.Th.U. per lb. (5500 to 7200 cal./kg.), giving about 10 per cent. ash and weighing 22 to 32 lbs. per cub. ft. as piled. Coke is brittle and too much handling or storage in piles subject to weathering will cause a considerable loss, as the coke will disintegrate, thus making it difficult and inefficient for combustion under the boilers.

For those countries where *fuel oil* (the final residue of the crude oil cracking process) is cheap, it can be used to advantage as it is easily fired under factory boilers as well as in locomotives. Moreover, it is easily stored in tanks and is not subject to deterioration, nor is it liable to the danger of spontaneous ignition.

The heat value is almost a constant figure, being around 18,900 B.Th.U. per lb. or 10,500 cal./kg. Its smoke production is also considerably less than that of coal.

The statistical measurement under which fuel oil is sold is the barrel, containing 35 Imp gals. (42 U.S. gals.). But sale in actual barrels is not very practicable and the author has no knowledge that it is done, the transport usually taking place in standard gauge tank cars up to 12,000 U.S. gals. or in tank steamers, when the sugar factory is situated close to the coast.

The specific gravity of oil lies between 0·85 and 1·00 at 39°F. but as quoted for sale it is generally stated in degrees Baumé, it being assumed that the specific gravity :

1·0000 = 10° Bé.	0·9032 = 25° Bé.
0·9655 = 15° Bé.	0·8750 = 30° Bé.
0·9333 = 20° Bé.	0·8484 = 35° Bé.

Its viscosity is determined by comparing the fluidity of the fuel oil with that of an oil of high fluidity, like colza-oil, in Redwood or Engler viscosimeters. Under tropical temperatures, the viscosity does not cause any inconvenience when pumping and conveying through pipe-lines. Considering the viscosity degree as unity at 100°F., it is 2·5 at 80°F., 8·8 at 60°F. and 35 at 40°F., which shows clearly that viscosity decreases with a rise in temperature and the oil, therefore, should be heated to about 150°F. before it is conveyed to the atomizers at the furnace, so as to guarantee the proper atomization essential for good combustion.

Under pressure, the following amounts of fuel oil will flow through the stated pipe sections :—

Quantity.	Pipe.
100 Imp. gals./hour	1½ in.
200 ,, ,,	2 in.
300 ,, ,,	2½ in.

A velocity of the flow of 2 ft./sec. (0·6 metre/sec.) will give satisfactory results for pressure pipe-lines, whereas suction lines should be credited with about half this velocity.

The ignition point, as determined by the closed method of MARTENS-PENSKY, is about 180°F. (about 100°C.) and it should be remembered that all the oil has to have this temperature and sufficient air available to cause ignition.

3.—Combustion.

Combustion is the rapid chemical union or combination between the burnable components of the fuel and the oxygen present in the air, thus being a rapid oxidation process, whereas corrosion is slow oxidation. Heat is released with this chemical performance, and there is no destruction of the components involved, but only a transformation, and the weight of the fuel plus the weight of air supplied for the combustion, are to be found in the flue gases plus the weight of the resultant ash.

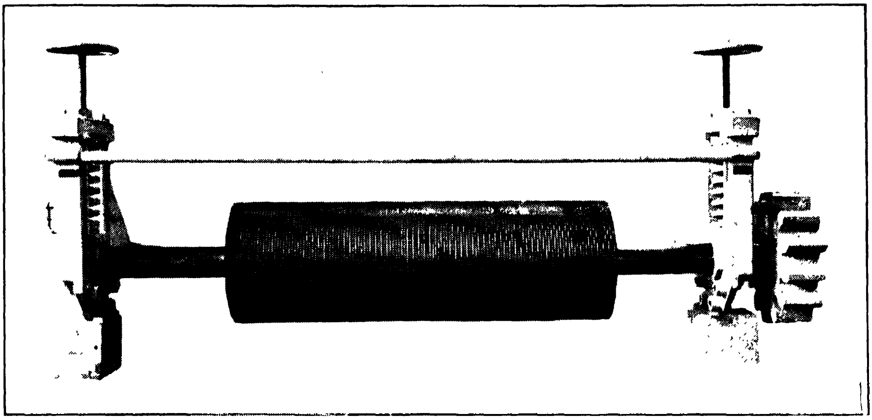
The principal burnable components in the fuels used for sugar factories are carbon and hydrogen, each atom of carbon requiring two atoms of oxygen, together forming one molecule of carbon dioxide, CO_2. Hydrogen combines at the rate of two atoms H to one atom O, forming H_2O, water, and this will leave the smoke stack as superheated steam of nearly atmospheric pressure. Only a part of the sensible heat is thus recovered for heating purposes ; the rest plus the total latent heat is lost.

The approximate chemical analysis of the different fuels mentioned is given below :—

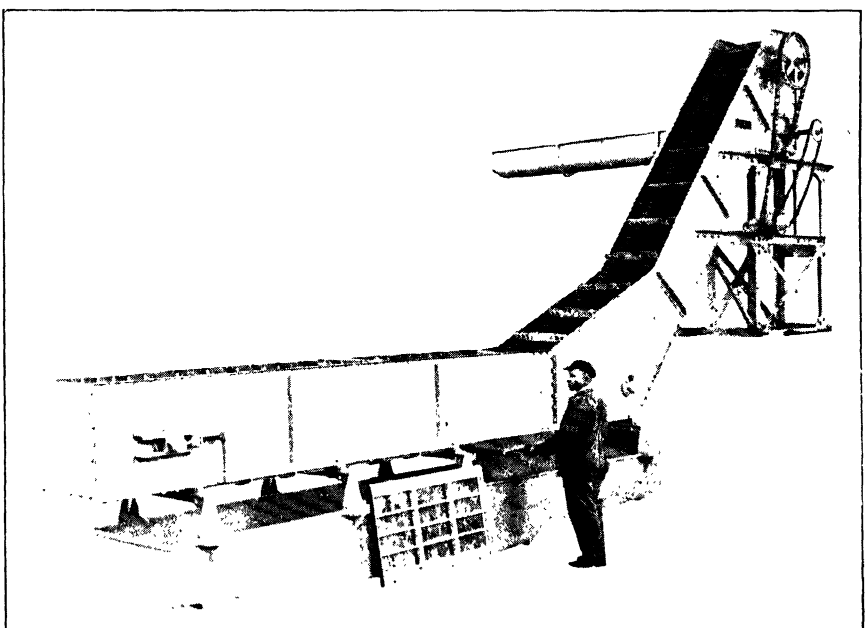
	Per cent.					
	Carbon.	Hydrogen.	Oxygen.	Nitrogen.	Sulphur.	Ash.
Bagasse (average)	45	6	47	—	—	2
Wood (soft or hard)....	50	6	43	—	—	1
Coal (average)	84	4	3	1	1	7
Coke	86	1	2	1	—	10
Fuel oil (average)......	85	11	2	—	2	—

According to DULONG'S formula, the different heat values of the components are added, thus :—

$$Q = 14{,}600\ C + 62{,}000\left(H - \frac{O}{8}\right) + 4000\ S \quad \ldots. \quad (83)$$

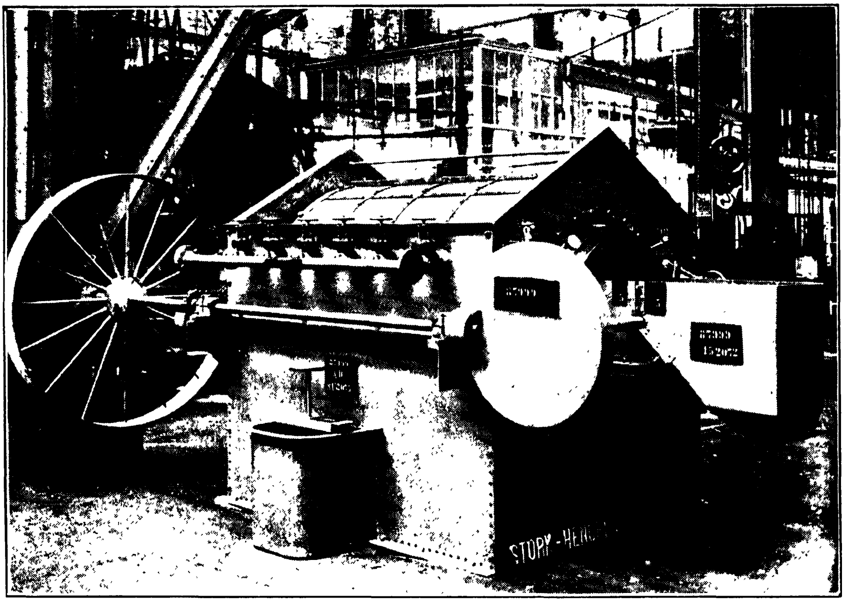


BURUNAT IMBIBITION ROLLER.
(Farrel-Birmingham Co., Inc.)

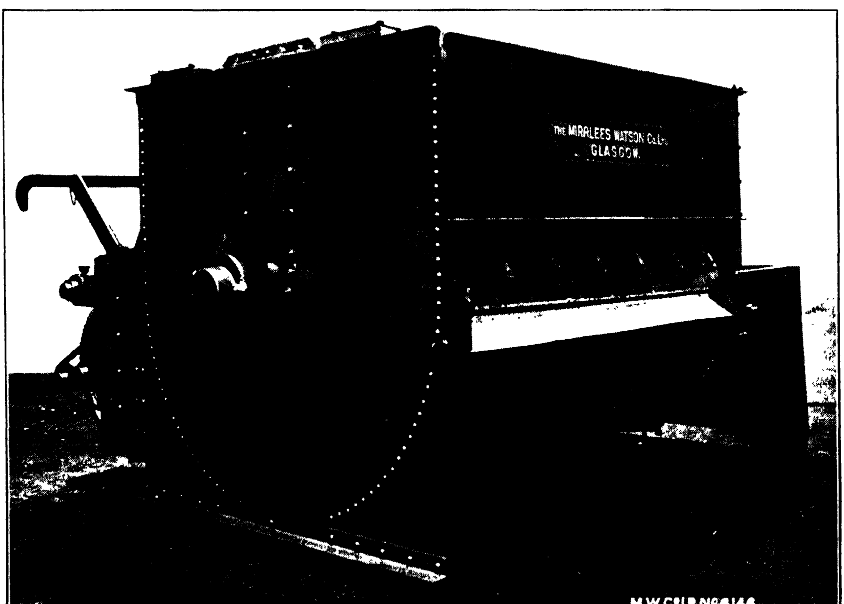


TUINUCU JUICE STRAINER WITH REMOVABLE SCREENS.
(Farrel-Birmingham Co., Inc.)

PLATES 50 & 51.



TROMP'S CUBA STRAINER FOR RAW MILL JUICE.
(*Gebr. Stork & Co.*)



PECK STRAINER FOR RAW JUICE.
(*The Mirrlees Watson Co., Ltd.*)

The amounts of C, H, O and S are given in hundredths and the total heat Q is obtained in B.Th.U. per lb. DULONG's formula does not give proper values for bagasse and wood, probably owing to the hydro-carbons present, and the heat value is decided not only by the chemical elements of the fuel, but also by the way they are combined in that fuel.

For comparison of the weights of different chemical elements and their combinations, the atomic weight has been established, and the approximate values are :—

C = 12. O = 16. H = 1.
CO (imperfect combustion) 12 + 16 = 28.
CO_2 = 12 + 2 × 16 = 44.
H_2O = (vapour) = 2 + 16 = 18, etc.

Thus 1 lb. C requires for complete combustion 32 ÷ 12 = 2·67 lbs. O and as the air contains only 23 per cent. oxygen by weight, 2·67 ÷ 0·23 = 11·6 lbs. of air have to be furnished for complete combustion of 1 lb. carbon. Similarly, 1 lb. of H requires 8 ÷ 0·23 = 34·78 lbs. of air.

The volume of the gases has as much importance as the weight, and the following weights of gases at 32°F. and at atmospheric pressure (14·7 lbs./sq. in. absolute pressure) are tabled :—

Air	0·08071 lbs./cub. ft.
Oxygen O	0·08921 ,,
Hydrogen H	0·00562 ,,
Nitrogen N	0·07807 ,,
Carbon monoxide CO	0·07806 ,,
Carbon dioxide CO_2	0·12341 ,,

As the gases occupy a volume in proportion to their absolute temperatures and also to the absolute pressures, the combination of the laws of GAY-LUSSAC and MARIOTTE may be written as :—

$$P \times V = R \times T \qquad (84)$$

where P = absolute pressure of the gas in lbs. per sq. ft. (atmospheric pressure = 2116·27 lbs./sq. ft.)

V = volume per lb. of gas in cub. ft.

T = absolute temperature of the gas in °F. = t + 460°F. (t being the normal temperature in degrees Fahrenheit).

R = gas constant as per table below :

For air	53·33
For oxygen	48·24
For hydrogen	765·80
For nitrogen	55·13
For carbon monoxide	55·15
For carbon dioxide	34·88

With these data, we are able to calculate the weight and the volume of the gases, derived from the combustion of the fuel, and also the total amount of heat produced, but we do not know the temperature of combustion. The specific heat of the gas and the temperature have to be known ; but, unfortunately, the specific heat of gases is subject to the temperature and also to the pressure. There is a divergence of opinion amongst leading authorities about the exactness of these figures, but for our present calculations, the figures of NEUMANN are given for three different temperatures at atmospheric pressure. The specific heat of air is taken as a constant value.

Temp.		86°F. (30°C.) Tropical day temperature.		520°F. (270°C.) Average flue temperature.		1920°F. (1050°C.) Average combustion temperature.
Air	..	0·240	..	0·240	..	0·240
CO_2	..	0·204	..	0·223	..	0·261
H_2O	..	0·462	..	0·467	..	0·496
N	..	0·249	..	0·254	..	0·266
O	..	0·218	..	0·222	..	0·233

With a perfect combustion of carbon, the gases will contain about 21 per cent. CO_2 by volume and 79 per cent. N. In case of the oxygen combined with the hydrogen in our fuels, 19 per cent. CO_2 is about the limit obtainable. With the Orsat apparatus combustion gases are analysed by volume, and with a measured CO_2 content of 13 per cent., about 6 per cent. O is present. This reduction in CO_2 is due to the fact that an excess of air has been given for combustion. In practice it is never possible to achieve complete combustion with just the requisite amount of air, as the mixture of fuel and oxygen is less efficient in presence of draft currents and of the huge percentage of nitrogen which has to be carried along without serving any use and which, moreover, is heated and part of its heat goes out through the chimney.

The proportion 19 ÷ 13 = 1·46 gives an approximate indication of the *Excess Air* required, being in this case about 46 per cent.

From several estimations made by the author, a few average data are offered for overall calculations :—

One lb. of bagasse having about 52 per cent. moisture, requires for complete combustion .... 2·7 lbs. air

Excess 50 per cent.	1·35 lbs. air
Total air	4·05 lbs.
Bagasse fuel	1·00 lbs.
Weight of combustion gases....	5·05 lbs.

The combustion temperature obviously will be lowered by any excess of air, as this additional amount of air has to be heated. The following approximate temperatures are calculated, but furnace temperatures generally will be lower through radiation and the moisture content of the bagasse :—

With no excess air	1650°C. or 3000°F.
With 50 per cent. excess air ..	1300°C. or 2370°F.
With 100 per cent. excess air ..	1060°C. or 1940°F.
With 150 per cent. excess air ..	735°C. or 1355°F.

The best way to ascertain when excess air is given is to observe the top of the smoke stack, from which a greyish smoke should emerge. If there is no smoke, all the fuel has been completely burned resulting in colourless flue gases, but this can be achieved only with a considerable excess of air. The Orsat apparatus will indicate in such cases a low CO_2 content and a high O content.

With a flue gas temperature of 270°C. (520°F.) the chimney loss amounts to 18·6 per cent. on dry gases, which can be reduced by installing an air-heater ; but all cannot be saved, as the flue temperature cannot always be lowered too much. The author has figures on record where 280°F. (155°C.) has been obtained with too much excess air, this being an exceptional case.

4.—Furnaces.

Bagasse burns with a long flame and sufficient space has to be provided in the combustion chamber to achieve a thorough admixture of the gaseous combustibles with the oxygen of the air. Bagasse, having about 50 per cent. moisture, has to be dried in the furnace before this dry distillation or gas formation will start, and combustion will not occur unless the proper ignition temperature is produced at the outset. Combustion gases, therefore, should not be cooled on exposed boiler heating surfaces before complete combustion has taken place.

Moreover, the refractory lining of the furnace serves the purpose of a heat accumulator to maintain the ignition temperature.

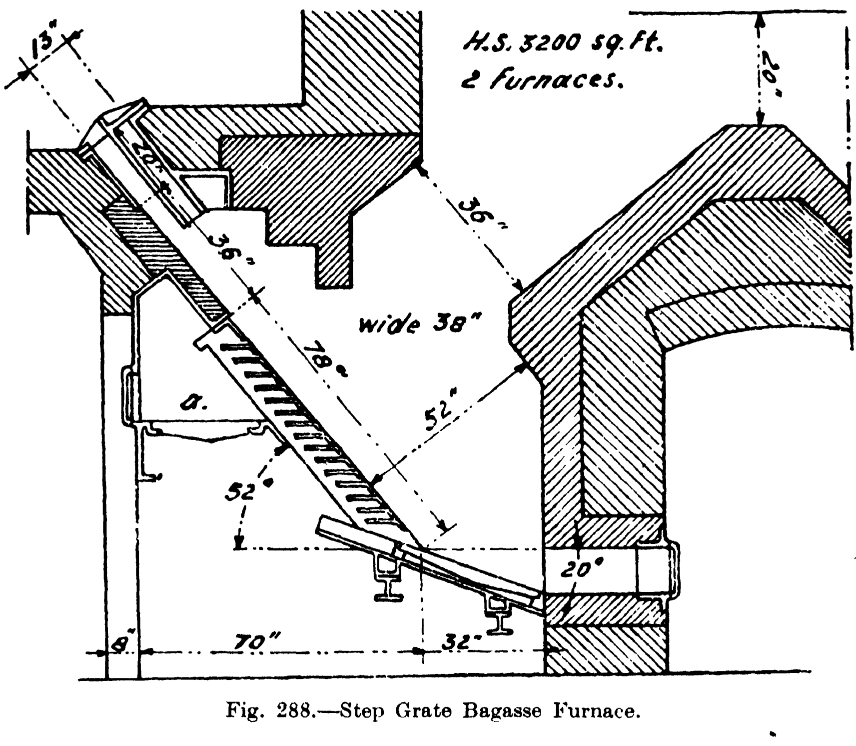


Fig. 288.—Step Grate Bagasse Furnace.

In *Fig.* 288 is shown a *Bagasse Step Grate Furnace* as developed by the Experimental Station of East Java. The step grate is used for low grade fuels in other industries and its purpose is to introduce the bagasse gradually to the zone of higher temperature. The bagasse has to be fed in a thin layer, which will not obstruct the passage of the air. Ashes accumulate on a plain grate at the lower end. Hand firing is done in those countries where manual labour is cheap and a good combustion performance in usually obtained. When starting the furnace, a fire is raised on the auxiliary grate *a* to heat the top space of the furnace and ensure the drying of the incoming bagasse. As soon as the step grate is well covered with burning bagasse, sufficient heat is developed for the drying performance on the dead plate.

The step grate slopes 52°, the most convenient angle, and the air slots are about 1 in. wide. The plain ash grate lies below, sloping about 20° from the horizontal and the ash will fall through this grate into the ash pit underneath, whence it can be easily removed.

The furnace neck serves the purpose of ensuring a good intermixture of the combustion gases, before they reach the boiler shell.

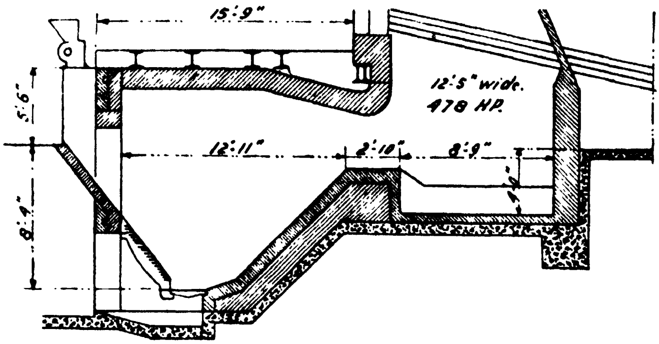


Fig. 289.—Flat Arch Step Grate Furnace.

Sharp corners should be avoided in the combustion chamber as the refractory brick will easily burn away at these points. Moreover, less resistance is offered to the flow of gas. A very good combustion, having 13·3 per cent. CO_2 present in the combustion gases, has been achieved.

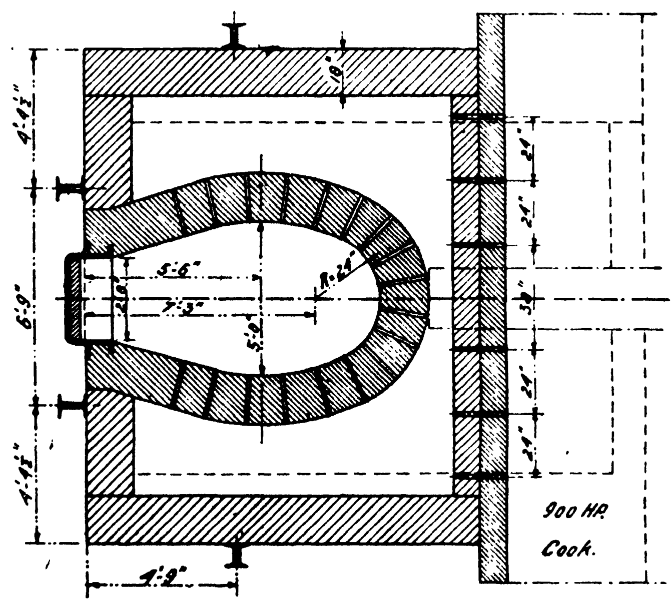


Fig. 290.—Horse-Shoe Bagasse Furnace.

A more recent construction of a step grate is the *Flat Arch Step Grate Furnace* (*Fig.* 289) as developed in Hawaii.[1] It aims at improving the combustion by having a sloping fire bridge, which can be raised by adding odd bricks, in case such an alteration is found better in practice. The flat arch has the advantage over the concave arch, in that the intermixture of the combustion

[1] See the paper of G. H. W. BARNHART, read before the Haw. Sug. Technol. Ass., 1927.

gases will be improved, as gases of lower specific gravity cannot so easily escape this intermixture. Moreover, equal distribution over the full boiler width is ensured. The flat arch is suspended on joists lying over the arch and the refractory bricks used are of the interlocking type. These flat arches have proved cheap in maintenance cost and the author knows instances where, over 10 years' operation, all repairs needed were made by the 10 per cent. of spare bricks originally supplied when the furnace was installed.

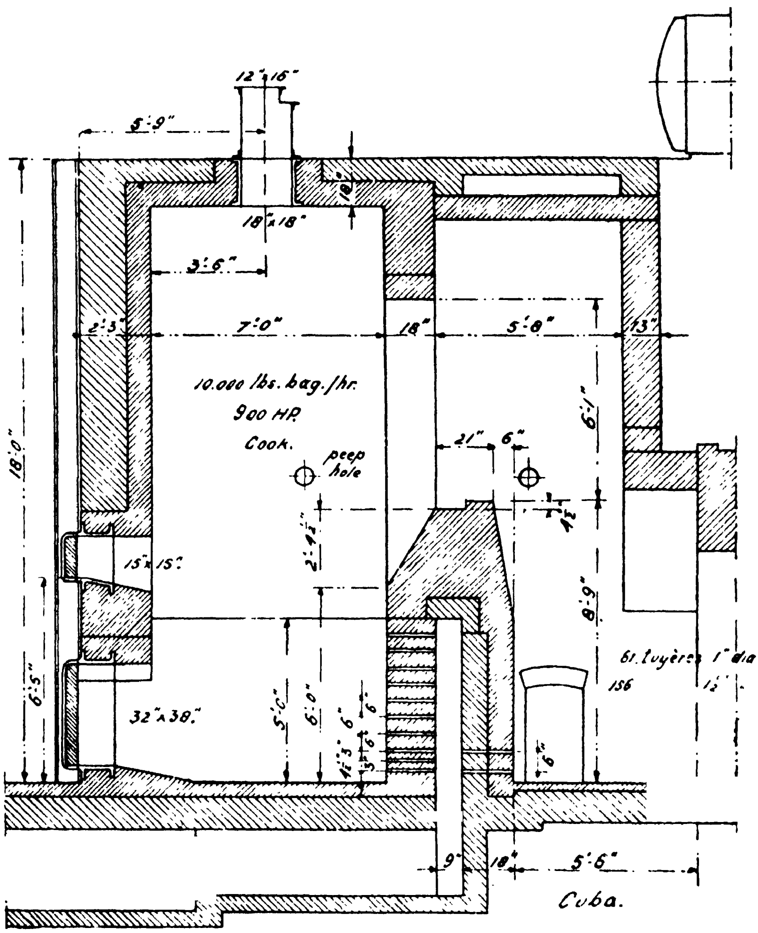


Fig. 291.—Vertical Section of Fig. 290.

A large combustion chamber is provided, and flying ashes will accumulate there, where these can be easily removed. Oil burners can be arranged in the back wall of the combustion chamber, when such is deemed necessary. The bagasse is automatically fed in this furnace over the full width and everything is arranged to avoid cause for choking.

Step grates are also furnished with hand-opérated pushers between the grates and favourable results of their application are reported.

In *Fig.* 290 is shown the sectional plan of a *Horse-Shoe Bagasse Furnace*, first fitted in Cuba some 40 years ago, and with which very favourable combustion results have been obtained, the CO_2 content normally lying between

13 and 14 per cent. on an average. Green bagasse is charged in at the top of the furnace and falls in a pile on the floor of the hearth. A cold or hot air blast (of about 0·75 in. water column) is blown from outside on the burning bagasse and a very high rate of combustion per square foot hearth surface has been obtained. In case the combustion is not completed in the furnace proper, secondary air is blown into the combustion chamber. While step grates will burn about 250 to 300 lbs. of bagasse per square foot grate area per hour, with 0·3 in. natural draught, this hearth will burn about 450 lbs. per sq. ft. per hour with a blast of 0·75 in. which may be increased to about 650 lbs. with a blast of 1·6 in.

The sectional elevation of this furnace is shown in *Fig.* 291 ; the top as well as the arches is of semi-circular shape. The bottom of the hearth is cooled by the air from the blast and the height of the horse-shoe is about 5 ft. The fire and ash doors of the furnace are lined with refractory brick on account of the high temperature prevailing. The ash doors have been supplied by the author made in cast steel, as they are very heavy and have to withstand rough handling when the fires are cleaned. The refractory brick is fastened by iron rods, which have to be well covered by refractory cement. As the ashes accumulate at the bottom of the hearth, they have to be removed every 12 to 24 hours. Care should be taken that when a water hose is used for damping the ashes, the water does not reach the ash door frame, as it would then easily crack.

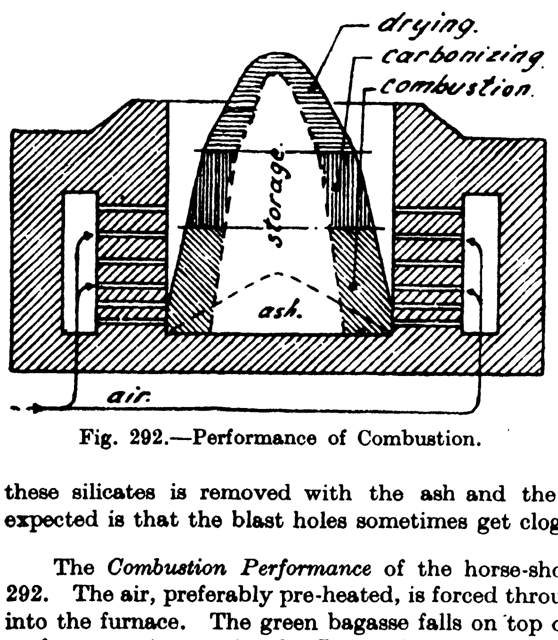


Fig. 292.—Performance of Combustion.

The bagasse contains silicates, forming a glazed substance at high temperature, which adheres to the furnace walls. Part of these silicates is removed with the ash and the only inconvenience to be expected is that the blast holes sometimes get clogged.

The *Combustion Performance* of the horse-shoe furnace is shown in *Fig.* 292. The air, preferably pre-heated, is forced through the blast holes or tuyeres into the furnace. The green bagasse falls on top of the pile, where the drying performance is completed. It will be obvious that the bagasse when falling through flames of high temperature will be partially dried. The second zone is the carbonizing zone, where dry distillation takes place and volatile matter is burnt. The lowest zone is the combustion zone with the highest temperature. As the bagasse is falling down the cone and the air is supplied at the circumference, the maximum combustion will occur here, the centre forming a reserve of carbonized bagasse, which will get burnt in case the supply is interrupted, thus equalizing any irregularities in the feed.

The blast pipes or *tuyeres, Fig.* 293, are made from cast iron, and are 18 in. long, thus the length of two refractory bricks. The height is 2½ in., so they can be conveniently located between the bricks. The hole is conical with the smallest diameter towards the hearth, and diameters vary between 1 and 1½ in. at the smallest end.

A *Flat Arch Grate Furnace* is shown in *Fig.* 294; the bagasse is fed in automatically, thus forming a cone on the grate, which slopes towards the tuyeres in the fire bridge; the combustion at this end will thus be more intense. The ashes will fall through the grates, a convenient arrangement, as the cleaning of a large furnace like this would cut out the heating performance for a short interval.

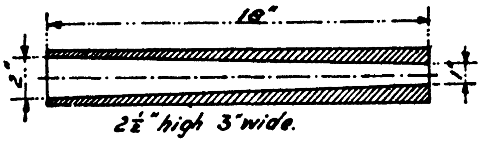


Fig. 293.—Tuyeres (Blast Pipes).

A provision is made for burning oil along the rear wall of the combustion chamber and flying ash will deposit at the bottom of it; the refractory tiles

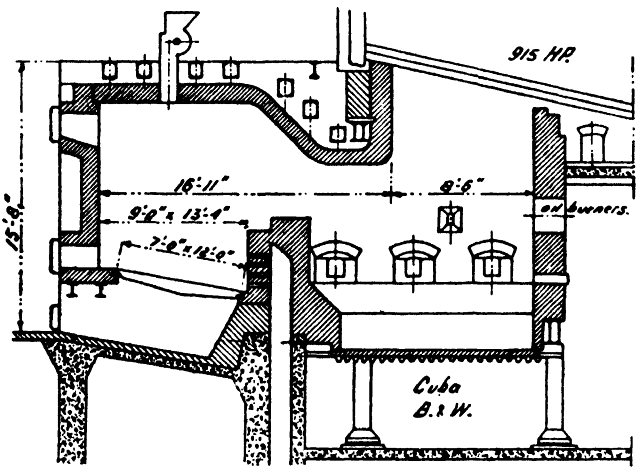


Fig. 294.—Flat Arch Grate Furnace.

being removable to allow for the extraction of the ash. Fuel oil burns at a higher temperature than bagasse and the refractory side walls should be about 2 ft. 6 in. away from the burner. The front wall or fire bridge should be at least 10 ft. distance from the tip of the burner, as the author has seen refractory walls burnt through when subject to the direct action of the flames. This rear oil burner arrangement is used extensively with bagasse furnaces.

The bagasse is fed automatically by means of hoppers having a counter-balanced flap, but it is better practice to have a *Rotary Bagasse Feeder*, as shown in *Fig.* 295, which will eliminate any unchecked entry of air into the furnace, at a spot where it is not always required. The feeder is composed of a drum, having five partitions, so as to seal the opening

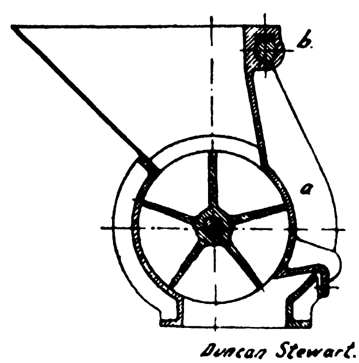


Fig. 295.—Rotary Bagasse Feeder.

completely when revolving. The speed and size of the feeder drum depends on the volume of bagasse to be fired per hour, and it should be remembered that excess capacity has no drawbacks.

Lest any obstruction should enter the feeder, the side *a* is mounted on the shaft *b* and free to revolve thereon, so that it will open and avoid a breakage. Moreover, the feeder shaft is driven by a chain sprocket having a spring releasing clutch, which will operate when too heavy a load is encountered.

As already explained, bagasse and fuel oil burn at different temperatures, hence both fuels must have different combustion chambers, in order to achieve perfect combustion in each case.

There are two types of *Oil Burners*, the one having the oil supplied by pressure from a pump, so as to cause atomization through the very small perforations of the burner tip; these perforations vary between 0·0156 and 0·0859 in. in diameter, so it will be obvious that the fuel oil has to be carefully strained before being pumped to this type of burner. The amount of oil depends upon the pressure applied, but the atomizing effect is also subject to this pressure and, therefore, this class of burner is used in those cases where there is no variation in the amount of oil to be fired.

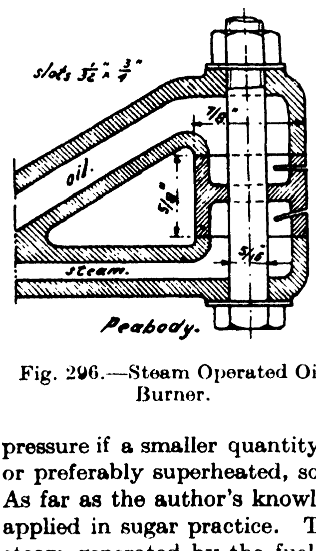


Fig. 296.—Steam Operated Oil Burner.

For sugar-house work, where great variation is required in the amount of oil used (since the expense for additional fuel has to be kept within the smallest limits) the *Steam Operated Oil Burner* is used to advantage. There are different types, but the essential features can be deduced from *Fig.* 296. The oil is pumped to the top of the burner tip, where it is sprayed through a slot, whereas the steam is injected from the lower part through an inclined slot, so that the steam spray will meet the oil one and a good atomizing action is thus obtained. The oil is pre-heated to about 150°F. or slightly below its flash point and supplied at a pressure of about 75 to 100 lbs./sq. in. or at lower pressure if a smaller quantity of oil is required. The live steam has to be dry or preferably superheated, so as not to have any dampening effect on the fire. As far as the author's knowledge goes, the use of compressed air has not been applied in sugar practice. The steam consumption is about 5 per cent. of the steam generated by the fuel oil alone, that is without reckoning the bagasse.

As the regulation of small quantities of fuel oil will involve a low rate of flow in the pipelines and thus a slow operation of the fuel oil pump, there are now oil burners on the market having a byepass pipeline for fuel oil, which can be regulated to the smallest limits.

The usual sizes of fuel oil pumps and oil heaters are as follows :—

For 2000 sq. ft. H.S., duplex pump 3 in. × 2 in. × 3 in.;
without heater connexions 1¼ in.

For 5000 sq. ft. H.S., duplex pump 3½ in. × 2¼ in. × 4 in.;
heater 15 sq. ft. H.S., connexions 1½ in.

For 10,000 sq. ft. H.S., duplex pump 4½ in. × 2¾ in. × 4 in.;
heater 25 sq. ft. H.S., connexions 2 in.

For 25,000 sq. ft. H.S., duplex pump 5¼ in. × 3½ in. × 5 in.;
heater 50 sq. ft. H.S., connexions 2½ in.

These pumps will supply all the oil needed for firing the boilers with fuel oil alone and an allowance has to be made when oil is to be considered only as supplementary fuel.

Fuel oil can be kept in steel tanks up to 80 ft. in dia. and 30 ft. high, provided with lightning conductors and surrounded by earth walls, about 4 ft. high, giving a capacity around the tank 50 per cent. in excess of that of the tank itself.

Concrete tanks can also be used, when made of a strong composition and lined with several coats of a 1 : 4 solution of sodium silicate—the last one of 1 : 2 strength. All fuel pipelines should be laid underground.

Oil burners have a steel casing around them for the admittance of the air, provided with adjustable blades or baffles to regulate the amount of air required. Induced draft is also applied to oil burners.

Cleaning Doors for medium temperatures can be made of cast iron, as shown in *Fig.* 297, having a protection plate of grey iron on the inside, well separated from the doors proper. The hinges on each side have through-going bolts of 1 in. to 1¼ in. dia. For keeping the double doors closed, two revolving clamps are provided. The clamps are held tightly between the door frame by locked nuts, so as to remain in any position desired and not obstruct the path of the doors, when being closed.

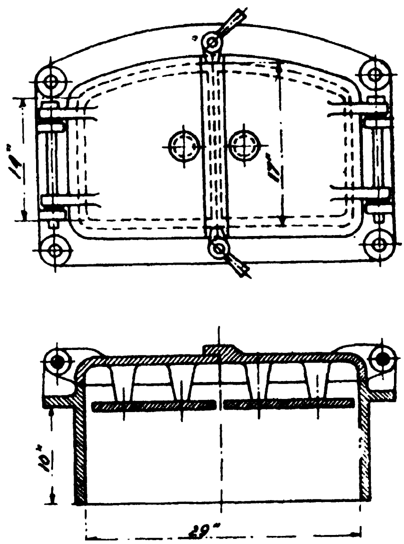


Fig. 297.—Cleaning Doors.

The door knobs are of substantial design. Sometimes handles are provided. The cast iron frame has to be well anchored into the masonry of the boiler setting.

5.—Types of Boilers in Use.

For cane sugar factories nearly every type of boiler has been employed at one time or other, but those designs which have been found particularly suitable for this industry can be divided into three large groups :—

(1) Return multitubular boilers.
(2) Multitubular boilers with bouilleurs.
(3) Water-tube boilers.

The multitubular and water-tube boilers are to be found in nearly all cane growing centres, whereas the boiler with bouilleurs is principally to be found in Java where over 800 are in use.

In *Fig.* 298 is shown a cross section of a *Multitubular Boiler*, having tubes of 4½ in. outside dia. and also threaded stays of 3¼ in. outside dia. An interesting feature of this European design is the bottom shell plate of heavier section, as it is subject to the action of the flames.

A vertical centre passage is provided between the two nests of tubes, about 11 in. wide, which will induce a better water circulation and allow a better inspection of the tubes.

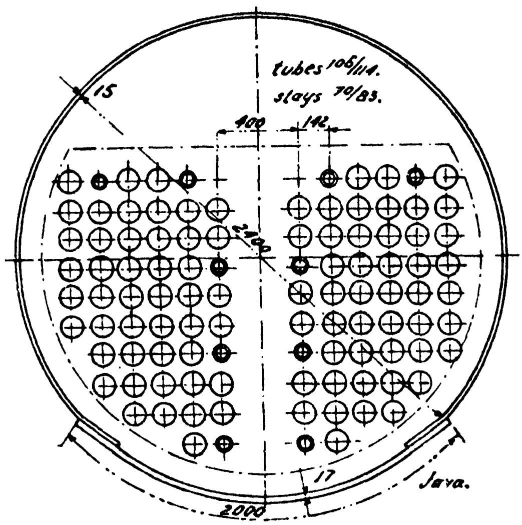


Fig. 298.—Multitubular Boiler.

The fronts are spherical except for the flat area where the tubes are grouped; this will eliminate the bracings necessary for flat fronts.

An American design of a multitubular boiler is shown in *Fig.* 299, having a braced flat front. A manhole, 15 in. × 11 in., is provided at the lower end, but the author would advise manholes 20 in. × 16 in., which will make access considerably easier.

The tubes are expanded in the front plates and the thicker the material the better will be the adhesion of the tubes. Five-eighths thickness is common practice in U.S.A. for sugar factory boilers, whereas European designers use end plates up to $1\frac{1}{16}$ in. thickness. The tubes should be of seamless steel with the ends annealed. The holes in the tube or front plates should not exceed the outside tube diameter by $\frac{1}{32}$ in., and the holes have to be chamfered before the tubes are inserted. After expanding the tubes the protruding portion of about $\frac{1}{8}$ in. is beaded and a good piece of workmanship thus achieved.

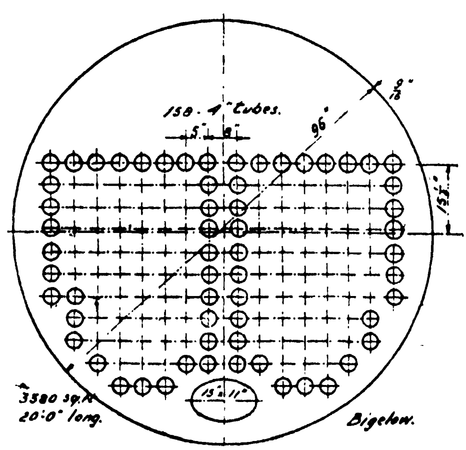


Fig. 299.
American Design of Multitubular Boiler.

With multitubular as well as with water-tube boilers, the flues will generally corrode just behind the tube plates, thus at the division line between the annealed and the non-annealed part of the tube, which might suggest that the tubes should be annealed over the full length or not be annealed at all. The author has repaired corroded tubes, by cutting off the corroded part and welding on a new end by electric or acetylene process. In some instances in remotely situated sugar factories, the author has seen these tubes welded in a common blacksmith's fire. Repaired tubes should be tested with a water pressure twice as great as the working pressure of the boilers, before being expanded in the boiler fronts or tubeplates.

The boiler tubes in *Fig.* 299 are 4 in. in outside dia., the thickness being generally No. 9 B.W.G. (0·148 in., 3·75 mm.). Threaded stay tubes are not used in this design.

The circular and longitudinal joints of a boiler are not subject to the same unit stresses, as can be easily deduced from the *Pressure Diagram* in *Fig.* 300. In the longitudinal direction the total force P_l will amount to :—

$$P_l = \frac{\pi \times d^2}{4} \times p$$

where d is the inside diameter of the boiler and p the working pressure in lbs./sq. in. The material which has to withstand this force has obviously an area of $\pi \times d \times t$, in which t is the thickness of the shell in inches. The average unit stress f of the material will thus amount to :—

$$f = \frac{\pi \div 4 \times d^2 \times p}{\pi \times d \times t} = \frac{d \times p}{4\,t} \quad \ldots\ldots\ldots\ldots \quad (85)$$

The longitudinal joint is subject to a force P_d amounting to :—

$$P_d = d \times l \times p$$

where l is the length of the boiler section reckoned in inches.

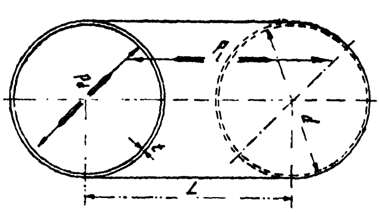


Fig. 300.—Pressure Diagram.

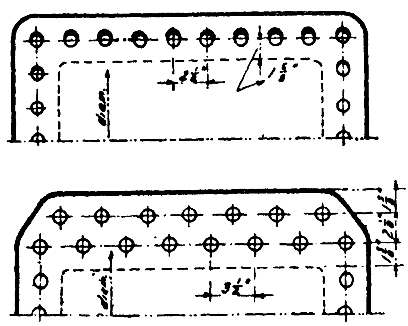


Fig. 301.—Patch Plate.

The two longitudinal strips of shell material have to withstand this force and the average unit stress f therefore will amount to :—

$$f = \frac{d \times l \times p}{2 \times l \times t} = \frac{d \times p}{2\,t} \quad \ldots\ldots\ldots\ldots \quad (86)$$

This shows clearly that the longitudinal joints of a boiler are subject to twice the unit stresses of the circular joints ; and longitudinal laps, therefore, are generally double riveted, whereas circular laps are single riveted.

Attention may be drawn to a repair done on a multitubular boiler shell, which had badly bulged and had been patched. The original *Patch Plate* is shown in *Fig.* 301 (top) with the length shortened in the drawing, having been riveted on the inside of the shell, but when inspected by the author, the longitudinal rivets were loose, as could be ascertained by hammering them, and when the patch had been removed, it revealed oval wear in the rivet holes as indicated by the heavy circles, and it was obvious that a mistake had been made in providing a single riveted lap joint, when the boiler shell had much heavier riveting on the longitudinal lap joints. A double riveted patch, as shown in the lower part of the figure, had since been applied with good results.

The strength of the different riveted joints can be found from the following table :—

Single riveted	lap joint	....	56 per cent.
Double ,,	,,	....	70 ,,
Triple ,,	,,	....	75 ,,
Double ,,	butt joint	....	80 ,,
Triple ,,	,,	....	85 ,,
Quadruple ,,	,,	....	94 ,,

The strength of any lap depends on the pitch of the rivets and their diameter, but the above values can be taken as a fair average of standard practice. It will moreover be obvious than an additional degree of safety can be obtained by using heavier plate material, but then the heat transmission will be less. The outer edge of the lap joints should not be facing the direction of the flames, as corrosion will then be increased.

The masonry of the boiler setting is subject to high temperatures and therefore may bulge or crack, and as a supporting medium for the heavy boiler it needs special provision. The *Suspension Arrangement* shown in *Fig.* 302, where the boiler weight is suspended clear of the masonry from overhead joists which are supported by individual columns, should therefore be preferred. The suspension bolts are $1\frac{1}{4}$ in. to 2 in. in diameter and the supporting columns have concrete foundations with a layer of refractory brick under the base plates. Foundation bolts are generally not used, but provision has to be made to ensure that the columns remain in place.

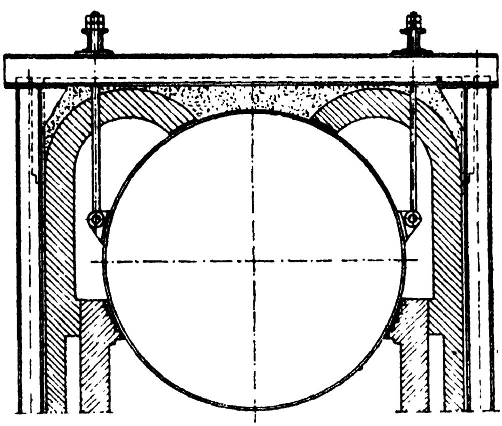


Fig. 302.—Suspension Arrangement.

The brickwork should be kept clear of the boiler structure, so that both can expand freely, and a packing of asbestos material is inserted between them, so as to close the joint.

Opinions differ amongst designing and operating engineers with regard to the use of refractory brick. With good installations all paths traversed by the combustion gases should be lined with refractory bricks as far as the entrance to the chimney. But this is a costly proceeding and good red bricks may be used instead in those channels where there is no direct flame.

Refractory mortar has to be of the same composition as the brick itself and should be supplied by the brick manufacturer ; the joints should be made as thin as possible, so as to get lasting construction. Fresh brickwork has to be dry before the boiler is started and it is best to heat it gradually for several days before getting the bciler to work. In those cases where the boiler setting is prone to absorb moisture during the off season, this preliminary heating is advisable and will help to keep the setting intact.

If cracks occur, they should be opened by a wedge and then filled up with a plastic mortar, generally refractory mortar mixed with asbestos. Anchorage by joists and tie-rods will sometimes stop further cracking and these should be applied with every arched boiler setting. Cracks will give rise to an unchecked entry of air which reduces the efficiency of the boiler.

During the off season the boilers should be emptied and the man and arm hole covers removed, so as to ventilate the interior. The inside, as far as feasible, should be painted with a mixture of graphite and fish oil or with some good anti-corrosive paint specially supplied by manufacturers for the purpose.

In case a tube bursts during crop time, a provisional repair by means of a tie-rod and two blind flanges can be applied. For 4 in. tubes the tie-rod should have 1 in. minimum diameter.

On the top of the boilers is placed insulating material like asbestos or a magnesia compound or even dry ash, but the latter has to be laid in such a way that it cannot drift away through joints, etc.

Water-tube boilers are generally suspended by heavy U-bolts, so as to avoid any riveted supports on the drums.

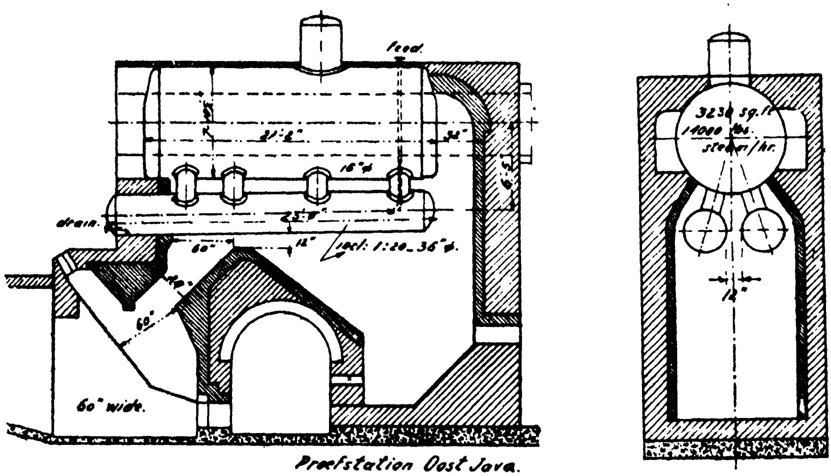


Fig. 303.—Multitubular Boiler with Bouilleurs.

The setting of a *Multitubular Boiler with Bouilleurs* (known also as an Elephant Boiler), *Fig.* 303, is widely used in Java. It is a boiler of an early type, but has special advantages in those countries where unskilled labour has to be employed. Maintenance costs are low and a long life is assured through the simple construction.

It is sometimes argued that the large quantity of water in this boiler will act as a heat accumulator and so provide for variations in load or steam consumption. This factor should however not be over-estimated, as the quantity of water present can only supply a small amount of sensible heat when the steam pressure drops, and it should be obvious that, as the latent heat necessary for steam production is about five times as great as the sensible heat, not very much can be expected when the furnace does not supply the additional heat required.

The two *bouilleurs* are connected, by four large dia. tubes each, to the multitubular boiler on top. Some manufacturers place these tubes at equal distances, whereas others prefer to have them concentrated at the front end as shown in *Fig.* 303, so as to improve the water circulation, which takes place in clockwise direction.

The *bouilleurs* slope towards the drain at the front end and the feed pipe passes through the rear tubes into the *bouilleurs*, being bent in the direction of the water circulation.

The combustion gases follow a three-fold path before entering the chimney flue.

The efficiency of this boiler being the quotient of the heat available in the steam produced, divided by the heat available in the fuel, lies normally between 55 and 65 per cent. and under favourable conditions 72 per cent. has been reached.[1] The average heating surface is between 2500 and 3500 sq. ft. and a spare unit is always provided for an emergency, or to allow for cleaning operations.

The thickness of the *bouilleur* shells is about ½ in. and a fairly good heat transmission is achieved.

The *Setting of a Multitubular Boiler* as used in Louisiana is shown in *Fig.* 304; here all heavy masonry is avoided by the fitting of a plate shell, filled with refractory brick, so as to reduce the radiation surface of the boiler setting.

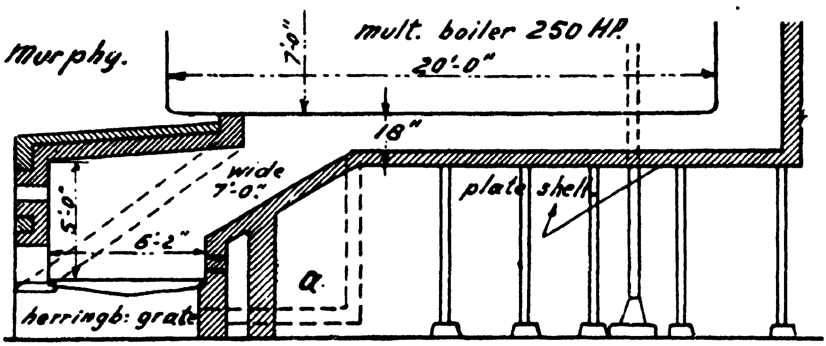


Fig. 304.—Setting of a Multitubular Boiler.

The grate is of the herring-bone type and hand firing is preferred, maintaining a thin layer of fuel, so that the air will pass freely under proper combustion at a normal chimney draught of about 0·3 in. The air slots in these grates comprise about 50 per cent. of the total grate area. As the boiler shell might cool the combustion gases before they are completely burnt, a more recent design has been evolved having a combustion chamber *a* as indicated by dotted lines, which is arranged between the furnace and the boiler. About 11 per cent. of CO_2 has been found in the chimney gases with the setting of *Fig.* 304, whereas 13 per cent. has been obtained with the setting of *Fig.* 303.

The *Straight Water Tube Boiler* has found many applications and was first introduced into sugar factories in the Western hemisphere. It will now be found in nearly all cane growing countries and a very good operating and maintenance performance has been obtained. The author has inspected boilers of this type, which had been installed 45 years ago, still having cast iron headers, and replacements had been confined to the water tubes and the ash and fire doors, the rest of the structure being in good operating condition.

It is sometimes argued that the explosion of a water-tube boiler will cause less damage than that of a multitubular boiler on account of the reduced quantity of water in the boiler, but it is safer to minimize this security.

[1] See *Het Archief*, 1927, pp. 427-511.

The first cost of a water-tube boiler is higher, and a careful calculation including all items has to be made when ordering.

There are many water-tube boilers of the straight tube type, and a predominant design in shown in *Fig.* 305. The boiler has 20 horizontal rows of tubes, 18 rows high, their outside diameter being 4 in., thickness No. 9 B.W.G., and length 20 ft. The author has in some instances replaced the lower rows, which corrode more quickly, by tubes with a thicker wall, e.g. No. 5 B.W.G. (0·220 in. or 5·5 mm.).

The furnace is of the horse-shoe type with flat arch, and oil burners are provided as a standby. A three-fold circulation of the combustion gases is secured by refractory baffles between the tube nests. These baffles are generally supported by cast iron plates and in case the profiled refractory bricks have suffered, repairs can be made with the very good high temperature cements now on the market. In existing tube nests such baffles of high temperature cement can be easily erected on wooden lattice work, inserted between the tubes. When the boiler is started, after the baffles are dry, the lattice work burns away, but the baffle, about 3 in. thick, will remain in position.

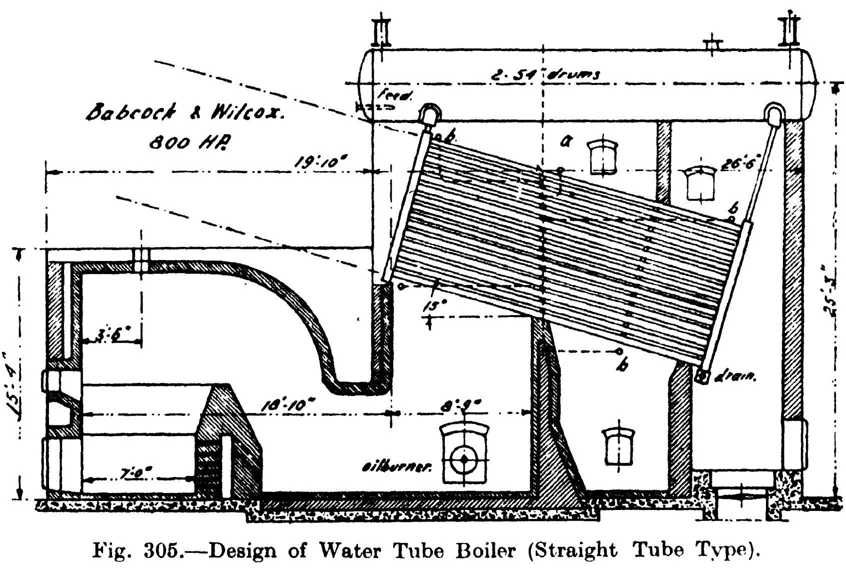


Fig. 305.—Design of Water Tube Boiler (Straight Tube Type).

With a draught of 0·38 in. above the hearth and 0·43 in. induced draft at the tuyeres, the two horse-shoe furnaces 7 ft. by 5 ft. have produced combustion gases with 14 per cent. CO_2. A furnace temperature behind the fire-arch of 2000 to 2100°F. (1100 to 1160°C.) has been measured and the flue gas temperature amounted to 440°F. (225°C.).

Efficiencies of water-tube boilers run as high as 75 per cent. or over, but after each crop careful inspection should be made as to the condition of the baffles, since a short circuit of the flue path will greatly reduce the boiler efficiency.

This boiler can be arranged with superheaters at the position *a*. For cleaning the tubes of the accumulated ash, the boiler setting is provided with long frames at the sides in the path of the combustion gases, having small openings spaced equal to the intervals of the water tube rows. These openings are protected each by a cast iron cover which can be easily lifted for the insertion of a steam hose. A better arrangement is to have fixed *Soot Blowers*, which will throw a number of steam jets between the pipe rows, vertically up or down. These steam jets are rotated during operation and thus an efficient cleansing is achieved. In the figure the location of these soot blower units is indicated at *b*.

The tubes are expanded in sectional headers of the sinuous type so as to allow for individual expansion. The headers are connected by nipples to the drums above and the rear headers also are connected by short nipples to the mud header, where the boiler is blown off. Each tube has a separate cover in the header, well secured by a bolt with cap nut and a bridge piece. These parts should be of reinforced construction and high-class material. The covers have machined joints and manganese packing is inserted between the cover and the seat on the header. Manufacturers supply special cutters for straightening these cover seats.

Another type of the straight water-tube design of recent construction which has been installed in cane sugar factories is the *Cross Drum Water Tube Boiler* (*Fig.* 306). The sectional headers are placed vertically and a very good

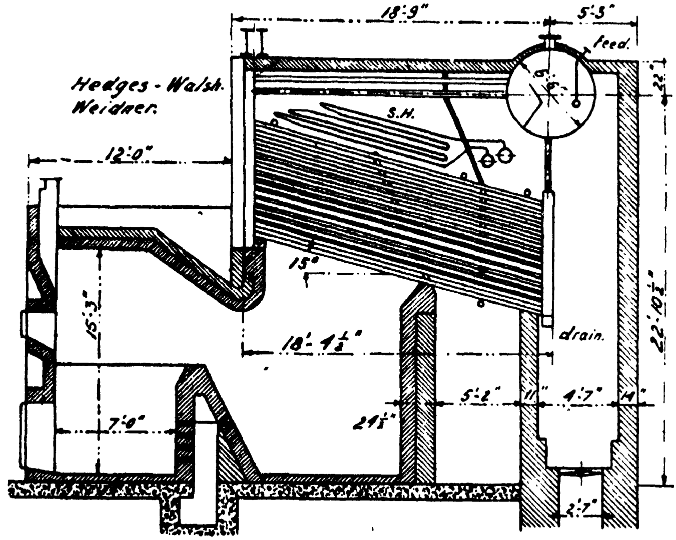
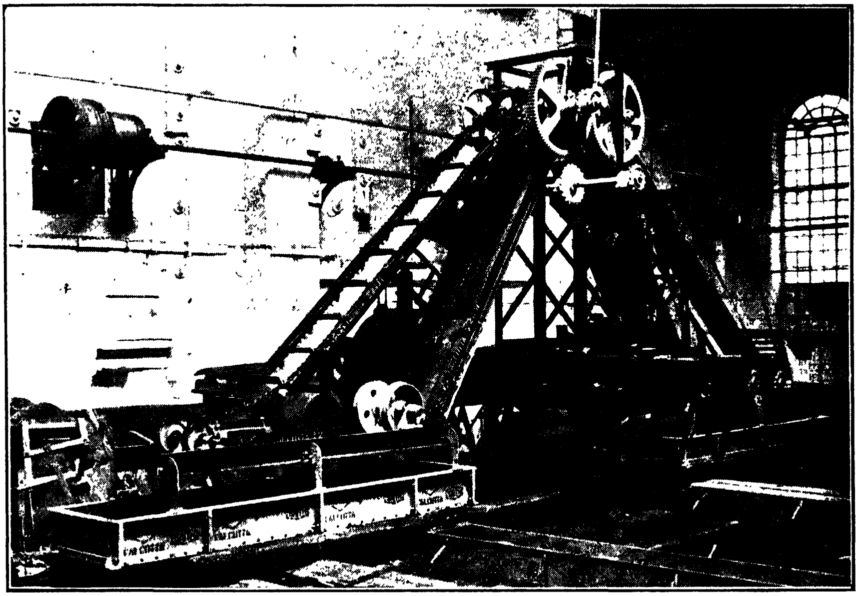


Fig. 306.—Cross Drum Water Tube Boiler.

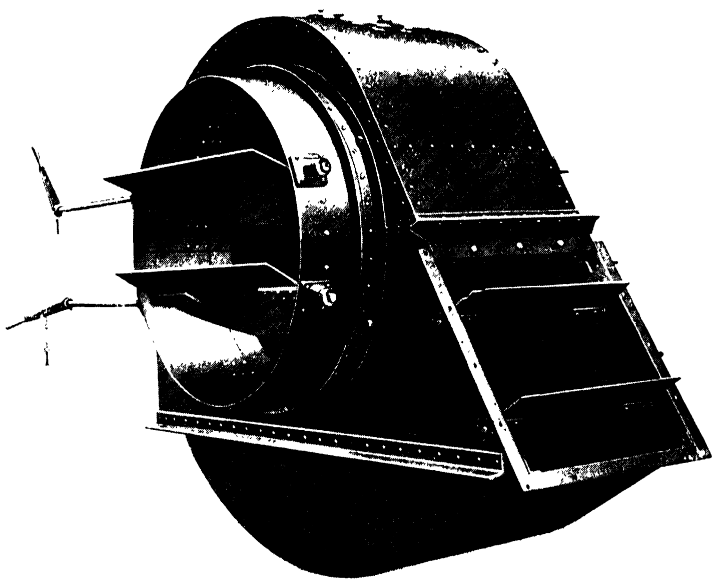
water circulation results. The rear headers are connected by long nipples to the drums and by short nipples to the mud header. The front headers are connected by horizontal tubes to the drum. A superheater, *S.H.*, is arranged on top of the tube nest. The feed is arranged in such a way that the coldest part of the boiler circulation will receive the water first and the sloping tubes release the steam to the top part of the boiler. Baffles and soot blowers are arranged similarly as with the design of *Fig.* 305.

A horse-shoe bagasse furnace has been provided, and rear oil burners can be used.

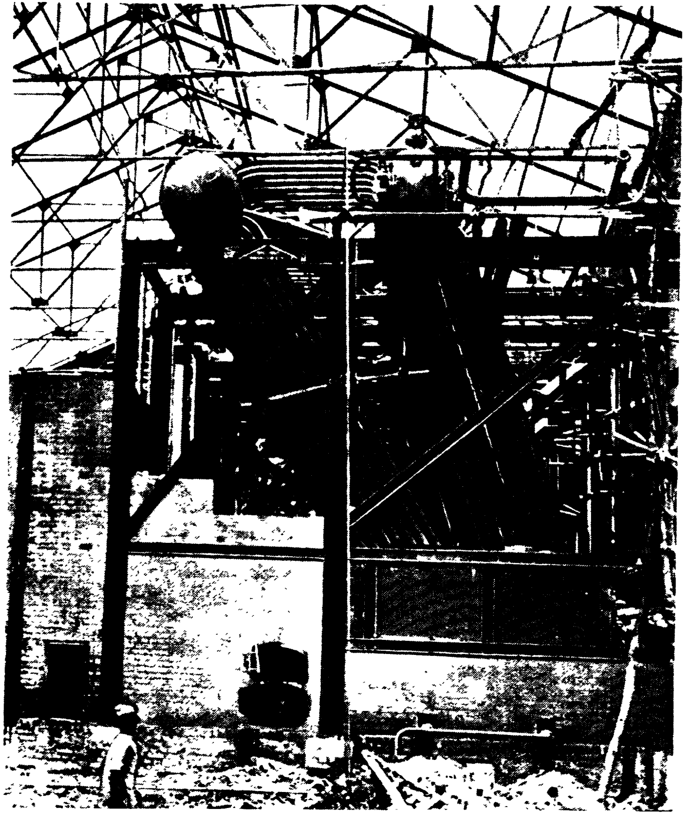
With all straight tube boilers, care has to be taken that the tubes can be replaced, generally from the front side, and only removable obstructions, like bagasse carriers, are allowed. Sometimes there are doors in the rear wall, whence the tubes can be removed, but sufficient space above the floor level has then to be available.



TRASH (CUSH-CUSH) ELEVATORS.
(A. & W. Smith & Co., Ltd.)



SINGLE INLET INDUCED DRAFT FAN WITH DAMPERS.
(Davidson & Co., Ltd.)



5,000 Sq. Ft. Water Tube Boiler in Course of Erection.
(*John Thompson Water Tube Boilers, Ltd.*)

Boiler in Course of Erection in British India.
(*John Thompson Water Tube Boilers, Ltd.*)

In *Fig.* 307 is shown an up-to-date design of a *Bent Water Tube Boiler,* having the tubes sloped at an angle of about 60° with the horizontal, thus ensuring a very efficient water circulation. These boilers have been adopted in the most modern factories in Australia, British India, Brazil, and other countries.

The boiler is fitted with step grates and a large combustion chamber, provided with secondary air supply and a three-fold path for the combustion gases will result in a good heat transmission and a high efficiency. Soot blowers are conveniently arranged to keep the tubes clear of ashes during operation.

The ashes from the step grates are discharged during operation into the ash pit by a patent ash dumping valve, which enables ash to be removed without interfering with normal combustion conditions and without cooling of the combustion chamber or reduction of the steam output.

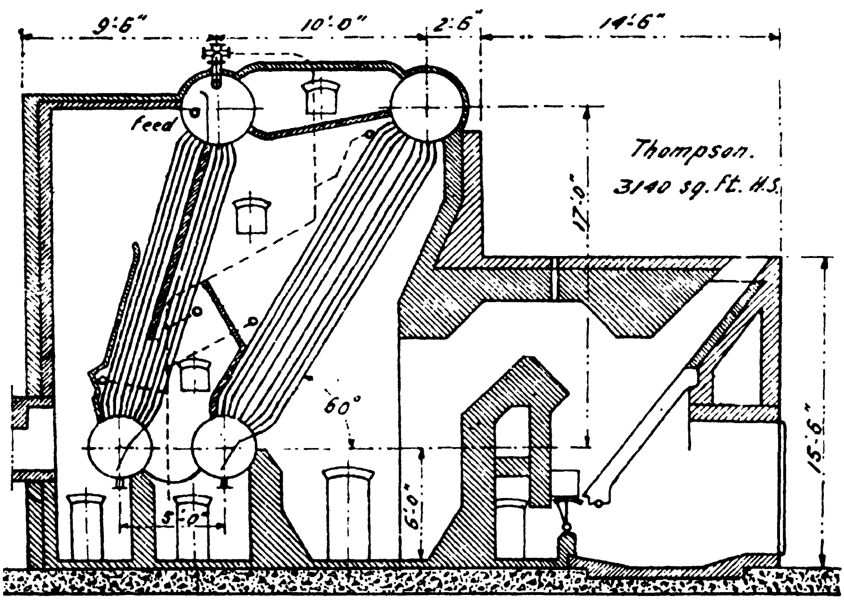


Fig. 307.—Bent Water Tube Boiler.

The heating surface in this type of boiler is formed by banks of steeply inclined tubes, each tube being entirely open and unrestricted, giving maximum freedom for circulation, and each tube is also only sufficiently bent to ensure correct entry into the circular drums, providing freedom for expansion and contraction during working conditions. These tubes owing to their steep angle and free circulation give little trouble ; the author has seen steep tube boilers that have had 28 years' operation before the centre tubes needed replacement ; the front rows of tubes, which require somewhat more frequent replacement, can be supplied with thicker walls.

The tubes are arranged with alternate wide and narrow pitch in order that any tube in the boiler can be replaced without disturbing adjacent tubes.

Should a water tube inadvertently fail during operation, the boiler has to be stopped, as otherwise it would empty in a short time and therefore has to be disconnected from the other boilers. After the boiler has sufficiently cooled off, the manholes on the drums are opened and mild steel thimbles can be inserted at both ends of the burst tube. When these thimbles were not available, the author has made use of hard wooden plugs, packed with a piece of canvas. The final repair can be made after the crop, when the boiler is cleaned and completely overhauled.

Another type of bent water-tube boiler of high efficiency is shown in *Fig.* 308 ; this has been installed in cane sugar factories in Hawaii. Louisiana, Cuba, etc., being equipped with a superheater, *S.H.* and a tubular air preheater, *A.H.*

The furnace is of the horse-shoe type and rear oil burners are provided. There are three upper and one lower drums and a three-fold path for the combustion gases is provided. These boilers rank among the highest in efficiency, and about 80 per cent. of the fuel heat has been recovered in the steam as a maximum.

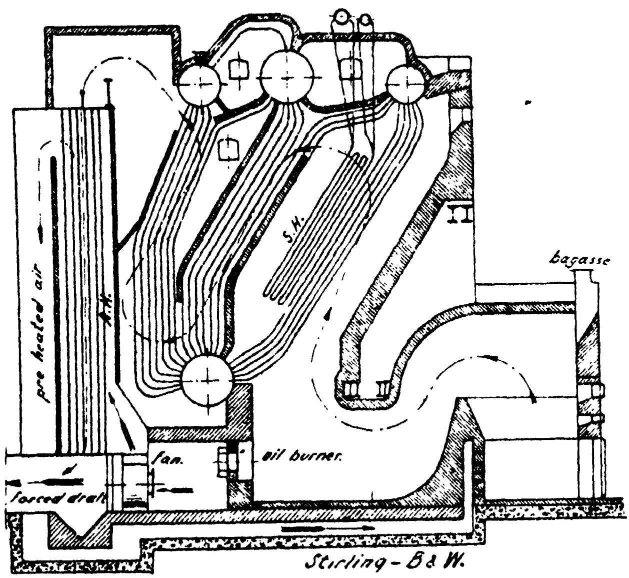


Fig. 308.—Bent Water Tube Boiler.

The feed is applied at the rear upper drum at the coolest spot of the boiler and will readily circulate down to the lower drum. The steam is also extracted from the rear upper drum, as in the front drum a considerable amount of water is carried along. All the upper drums are inter-connected by steam as well as by water connexions.

As the combustion gases encounter a high resistance, forced draught is applied and the air for the combustion is also forced by induced draught through the air pre-heater to the tuyeres of the horse-shoe furnace. The fans required for this service will consume about 2 per cent. of the heat energy produced.

The heating surface of the tubular air heaters is sometimes as large as that of the boiler itself. A heat transmission of about 2 B.Th.U. per sq. ft. heater surface and per degree F. in temperature difference is given, mean temperatures being considered.

A very recent design of the bent and vertical type of water-tube boiler, which has found application in several American sugar factories, is shown in *Fig.* 309. The design is very plain, and excellent water circulation is aimed at. The boiler is provided with a superheater, *S.H.*, and an economizer or feed-water heating principle is also embodied in the design.

The economizer as a separate unit has not found general application, as the condensed water from the heating bodies in tropical cane sugar factories can be kept at a temperature of about 203°F. (95°C.) for boiler feed; the original design in other industries considered heating cold feed-water by means of the combustion gases going to the smoke stack.

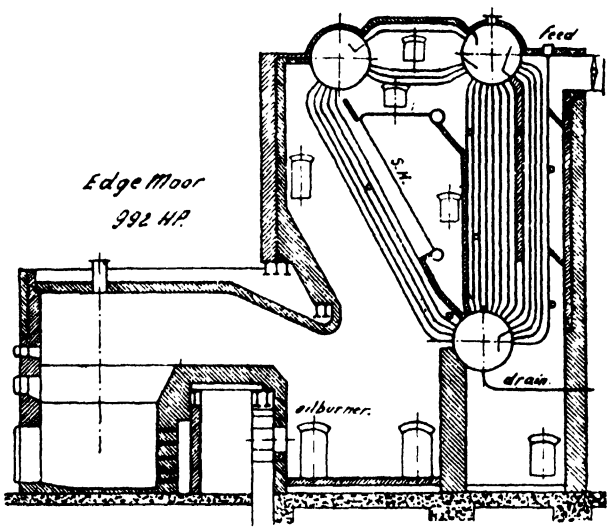


Fig. 309.—Bent and Vertical Water Tube Boiler.

The feed-water enters at the top of the last row of tubes and circulates three times in the last flue before participating in the regular boiler circulation. The rear rows of tubes can thus be fully considered as economizer units.

The superheater is located, as is usually the case, behind the first bank of tubes. Soot blowers are arranged at convenient places.

The bagasse is burnt in a horse-shoe furnace and oil burners with induced draught are provided underneath the fire arch. The oil burner tips should be removed from the combustion chamber when not in use, lest they be damaged by the high temperature ruling.

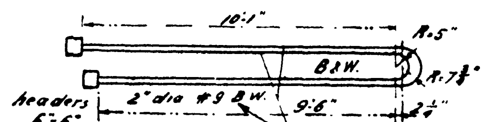


Fig. 310.—A Superheater Element.

A *Superheater Element* used in connexion with a water-tube boiler and measured by the author is shown in *Fig.* 310. Each element is composed of a U-tube 2 in. in diameter, and the ends are expanded in two 6 in. × 6 in. headers, which have tapped holes in continuation of the tube holes, so that a pipe expander can be inserted. The total heating surface of superheaters for a cane sugar factory amounts to about 30 per cent. of the boiler heating surface for a superheat of approximately 100°F.

As already mentioned, superheated steam is to be recommended in sugar factories for use in the prime movers.

Every boiler should be equipped with *Accessories*, both necessary and useful. The safety valves should be of the enclosed type so that the blow-off pressure cannot be varied by the operators. Moreover, there should be a steam pipe connexion beyond the roof of the boiler-house for the steam blown off. A low-water alarm is also a necessity, operated by the fusion of a melting plug, when the water has fallen below the safety level. The regulation of the dampers to the chimney passage should be arranged in such a way that the opening or closing can be easily controlled.

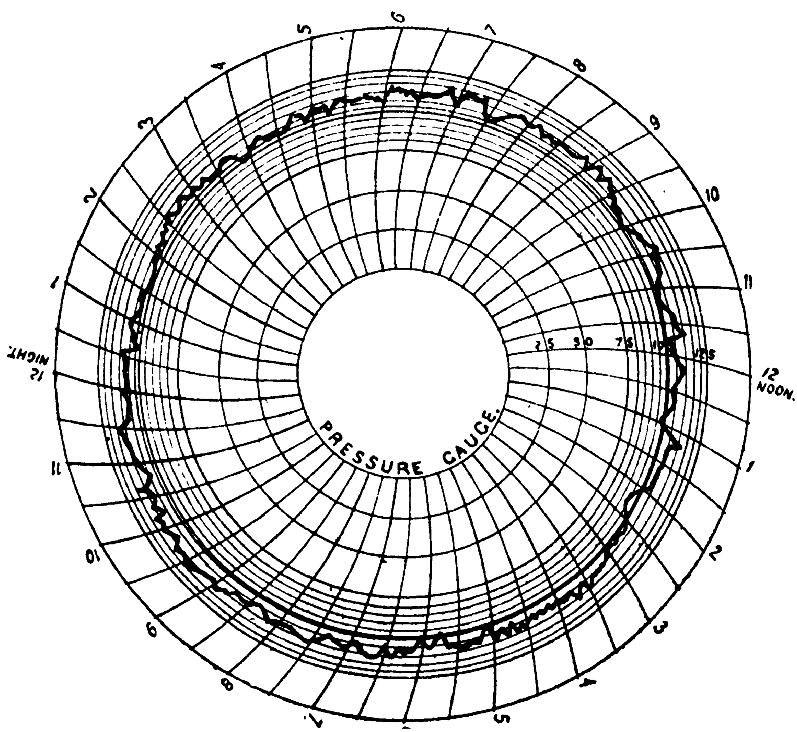


Fig. 311.—A Recorder Chart.

Water gauges have to be grouped at a spot visible to the firemen and have to be lit up at night time. The 8 in. to 10 in. pressure gauge on each boiler has to be tested after every crop and a general recording pressure gauge for the whole boiler battery should also be provided. In *Fig.* 311 is shown such a *Recorder Chart* from an efficiently operated Cuban sugar factory, this being an average one and not specially selected. The factory has only three horse-shoe furnaces, each for a water-tube boiler of about 8000 sq. ft. H.S., and it will be noted that there is practically no pressure drop during the cleaning of these furnaces, thus demonstrating the good training of the firing squad.

Other accessories comprise the steam dryers, and purifiers or tracifiers for retaining the moisture carried along. Return multitubular boilers are sometimes fitted with steam domes arranged on top of the boiler to ensure the supply of drier steam.

Automatic non-return valves should be attached to the steam line of each boiler, so that in case the pressure in such a boiler drops suddenly through a burst flue or water-tube, this valve will automatically close.

The blow-off valve should be arranged at the lowest level of the boiler and at the coolest spot. The construction of this valve has to ensure that its working will not be impaired by any scale and mud which is blown off with the water.

Oil separators for the extraction of any chance accumulation of oil in the boilers are sometimes used to advantage, but a better method is to have oil separators in the exhaust steam lines of the piston engines and a feed-water filter. Oil will cause acidity of the water in the boiler and a slight alkalinity is to be preferred, as it also assists in the coagulation of the oily components, which can be drawn off with the sludge. Any acidity has a corrosive action on boiler plates.

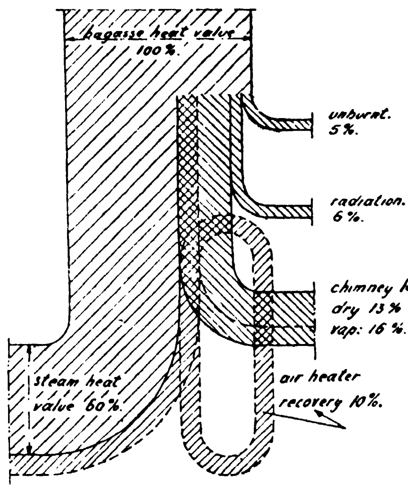


Fig. 312.—Boiler Heat Flow Diagram.

Draught and air gauges are also convenient accessories, and flue gas temperature recorders will add to a proper boiler control. Behind the damper of each boiler a connexion should be made for attaching an Orsat apparatus for flue gas analysis. Recording CO_2 meters of the chemically or mechanically operated type will be found in use in large sugar factories.

After the crop the boiler should be boiled out with a solution of soda to rid the inside surfaces of their oily scale.

Water-tubes of the straight and the bent tube types should be cleaned of scale by rotary tube cleaners, operated either by air, steam, or water pressure. Tubes of multitubular boilers have to be thoroughly scraped, but the hammering action of a special rotary tube cleaner will release the scale on the outside. There are decrustating solvents on the market which will remove the scale in the latter type of boiler on boiling the solvent in it. Care has to be taken to rinse the boiler afterwards with clean water.

As the chimney losses with bagasse furnaces are higher than for other fuels on account of the water vapours leaving the chimney which contain latent heat, any heat recovery from the chimney gases should be worth consideration. Feed-water heating is not as useful as air heating, as the temperature for air is about 30°C. (86°F.), whereas feed-water is already available at 95°C. (203°F.), and thus more heat can be extracted from the gases by heating the air necessary for combustion. *Air heaters* therefore have been embodied in the design of up-to-date boilers in cane sugar factories.[1]

The air temperature is raised to about 400°F. (200°C.) while the quantity used is also reduced on account of the better combustion with pre-heated air, so less excess air is required. But the furnace temperature will increase and the refractory lining has to withstand the higher temperature.

[1] See the articles of Prof. E. C. von Pritzelwitz van der Horst, Third International Congress, Java, 1931; and G. P. Ward, Fifth Conference, Cuban Sugar Technologists, 1931.

The two types of air heater mostly used in cane sugar factories are the *Tubular Type* as shown in *Fig.* 308, and the *Rotary Air Heater* as shown in *Fig.* 313. The tubular air heater has steel tubes, which is permissible since the dew-point of the combustion gases of bagasse lies around 140°F. (60°C.) and thus the danger from condensation is not marked. Six-inch cast iron tubes (such as previously used for the hot air heaters) were found to give only a partial heat transmission, but for other industries cast iron pre-heater tubes have already found application, having a smaller inside diameter.

Fans for forced and induced draught are required in case of velocities of air and gases running up to about 50 ft. per sec., whereas natural draft alone only gives velocities of about 15 ft./sec.

The *Heat Flow Diagram* of the average sugar factory boiler is shown in *Fig.* 312, where a 60 per cent. efficiency is assumed. By pre-heating the air, the efficiency can be raised about 10 per. cent. It should, however, not be overlooked that the additional efficiency is only required in those factories where there is a large power demand and where a shortage of bagasse fuel might occur. There is no object in saving bagasse, when it cannot be sold at a remunerative price or be used for other purposes. In some factories the transport or conveyance of excess bagasse has become a liability instead of an asset.

The rotary air heater in *Fig.* 313 is an ingenious design, the revolving part containing a large metallic grid surface which is heated by the combustion gases and cooled by an induced air current. The speed is about 4 r.p.m. and the cooling, through the air, will be about 15°F. per revolution.

6.—General Boiler Data.

So far as not mentioned already in this Chapter, the following data are given.

The *heating surface* of boilers, as an average, is taken :—

for raw sugar factories at	15 sq. ft.	per	ton of	cane /24 hrs.
for sulphitation ,,	17	,,	,,	,,
for carbonatation ,,	19	,,	,,	,,

The author knows instances where only 11 sq. ft. per ton of cane per 24 hours was available, such raw sugar factories having nevertheless a very good heat balance with about 10 per cent. fibre in cane.

Operating engineers differ in their views as to the number of furnaces required. In the Eastern hemisphere small units with step grates are preferred, whereas in the Western hemisphere large horse-shoe units are preferred. Personally, the author has found no inconvenience in operation when only three furnaces were provided, the size of the unit being for an average combustion of about 10,000 lbs. bagasse per hour.

The *Steam Pressure* employed in a cane sugar factory is generally 100 lbs./sq. in. More recent installations give 160 lbs. pressure and the maximum known to the author is 225 lbs./sq. in. High steam pressures are not required in sugar factories, as the power consumption is not excessive and the exhaust steam has to be used for heating purposes. The back pressure is generally below 10 lbs. but pressure evaporators are now used in a few instances requiring 25 lbs. back pressure and thus a higher live steam pressure.

The *Grate Area* is a very flexible figure. For step grates the proportion to the boiler heating surface varies between 1 to 50 up to 1 to 90, whereas for horse-shoe furnaces this proportion ranges from 1 to 100 up to 1 to 200.

The capacity of the *Combustion Chamber* also varies and recent designs in which favourable combustion results have been obtained are equipped with large combustion chambers. The amount of fuel burnt, i.e., the boiler rating, has to be taken into consideration. Normally 1 cub. ft. of combustion chamber volume is required for 3 to 15 sq. ft. boiler heating surface, the smallest figure for the highest combustion rates.

Boiler rating in U.S.A. is considered as one horse-power for 34 lbs. water evaporation from and at 212°F. per hour or about 10 sq. ft. heating surface. It is to be noted that this rating is a purely empirical one, as the steam supplied by a boiler can develop greatly varying amounts of power, depending upon the class of prime mover it has to supply. The evaporation figure per square foot of heating surface per hour is therefore a more practical figure.

Furnace Temperatures with cold draught lie between 1450 and 1900°F. and with hot air up to 2500°F. The temperature drops quickly, as may be noted from the following table :—

Furnace	1900°F.
1st path	1000°F.
2nd path	700°F.
3rd path	525°F.
Damper	425°F.

As the furnace temperature is judged by practical operators by the brightness of the incandescence, the following table may be useful :—

White	1300°C.—2400°F.
Bright Yellow	1200°C.—2200°F.
Dark Yellow	1100°C.—2000°F.
Cherry Red	900°C.—1650°F.
Dark Red	700°C.—1290°F.

The *Steam Production* of sugar factory boilers is generally 3 to 4 lbs. water evaporated per sq. ft. heating surface (14 to 20 kg./sq. metre) but in up-to-date boilers 7·5 lbs. has been obtained.

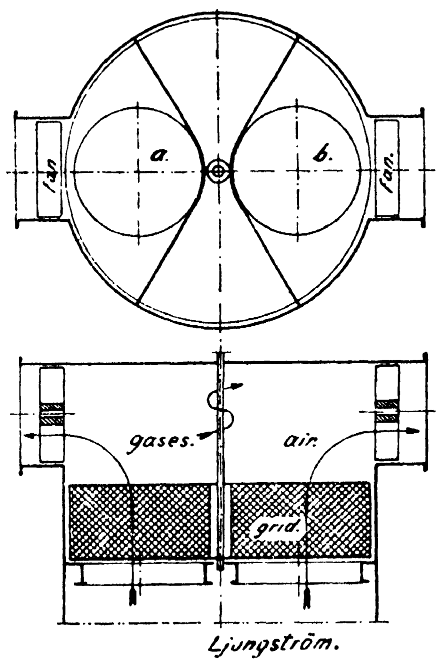


Fig. 313.—Rotary Air Heater.

One lb. of bagasse will produce between 2·4 and 3·2 lbs. steam, depending on the dryness of the bagasse and the efficiency of the boiler.

Refractory Bricks for the furnaces and boiler settings should be selected with great care, as the maintenance costs of unsatisfactory bricks are high. As furnace temperatures nowadays may reach 2500°F., bricks of 3200°F. fusion point, containing 42 to 45 per cent. alumina, are now obtainable at reasonable cost.

Good refractory brick should not spall under operating conditions. Moreover, slagging, i.e. the chemical union of the fuel components and the brick components, should not cause any disastrous lowering of the fusion point. Smooth refractory bricks will occasion less clinkering and the warpage under normal oven temperature should be less than 2 per cent.

Refractory brick has not always high insulating capacity and in recent boiler settings insulating brick has been applied between the refractory and the red brick. These insulating bricks are not to be considered as highly refractory.

Refractory brick expands under high temperatures and expansion joints of asbestos material should be applied at about every 8 ft. of wall length.[1]

With their rectangular shape, each brick makes a joint with the two lying underneath or on top. To increase the locking effect, rhombic-shaped refractory bricks are now available, giving a three-fold lock, and thus a longer life to the refractory walls.

Natural refractory stone is found in Crummersdorf, Germany, containing about 95 per cent. silex and having a fusion point of about 2960°F., being thus of very compact structure. The stones are hewn and therefore are expensive ; but are of lasting quality for high furnace temperature.

The fusion points of refractory bricks are measured by the Seger cones and are sometimes given according to the Seger cone scale. The sizes of fire bricks vary for different countries :—

Great Britain	9 in. × $4\frac{1}{2}$ in. × $2\frac{5}{8}$ in.
U.S.A.	9 in. × $4\frac{1}{2}$ in. × $2\frac{1}{2}$ in.
Germany	250 × 125 × 65 mm.
Holland	215 × 105 × 55 mm.

Different shapes, like keys and wedges, are made for forming arches, etc., and a sketch should accompany each inquiry, so as to indicate the exact sizes and shapes required. A 10 per cent. excess for breakage during transport in barrels should be allowed for.

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[1] See the article by G. W. CONNON, *Facts about Sugar*, 1929, page 1146

CHAPTER XIII.

SMOKE STACKS AND FEED WATER.

CHIMNEY AREA AND DRAUGHT — WIND PRESSURE AGAINST STACK — CHIMNEY DESIGNS — FEED-WATER — STEAM ACCUMULATORS.

The performance of combustion requires air, and to overcome the resistance of the fuel bed and the passages through the boiler, a draught has to be produced either by mechanical means or by the difference in specific weight of the combustion gases in the stack and the air surrounding it.

A higher chimney, therefore, will produce an increased draught with the same flue gas temperatures, and higher temperatures will have the same effects.

For economic boiler performance, the stack temperatures should be kept as low as possible, so as to reduce heat losses.

Draught considerations apart, the smoke stack serves the definite purpose of discharging the smoke and the combustion gases well above the surroundings so that these will not cause any nuisance in the vicinity.

1.—Chimney Area and Draught.

According to our assumption in the previous Chapter with a flue gas temperature of 520°F. (270°C.) and 5 lbs. combustion gases per lb. of bagasse fired with about 50 per cent. excess air, the average volume of these gases at atmospheric pressure can be derived from formula (84).

$$P \times V = R \times T$$

$$2116{\cdot}27\ V = 52 \times 980.$$

$P = 2116{\cdot}27$ lbs./sq. ft. $R = 52$ for average combustion gases of bagasse and $T = 520 + 460 = 980$°F.

The volume of the combustion gases thus results in $V = 24$ cub. ft. per lb. and 1 lb. of bagasse will thus produce about $5 \times 24 = 120$ cub. ft. of combustion gases at the above-mentioned temperature of 520°F.

As 1 lb. of bagasse produces about 2·5 lbs. steam and as an average of 4 lbs. steam is produced per sq. ft. heating surface of the boiler per hour, the following conversion table can be established :—

One lb. bagasse produces 2·5 lbs. steam.
One sq. ft. heating surface produces 4 lbs. steam/hr.
0·625 sq. ft. heating surface produces 2·5 lbs. steam/hr.
0·625 „ „ requires 1 lb. bagasse/hr.

The area of the flue gas passage to the chimney should thus be calculated by assuming a velocity of the combustion gases of 15 ft. per sec.—a practical figure, which will not cause too much friction ; and for each lb. of bagasse we require : $120 \div (15 \times 3600) = 0{\cdot}0022$ sq. ft. passage area.

Per sq. ft. heating surface 0·0036 sq. ft. flue area is thus required. In practice 4 sq. ft. flue area is the accepted standard per 100 h.p. boiler rating, being about 1000 sq. ft. It is obvious that the flue area should be calculated according to the bagasse burnt or the boiler rating and not to the heating surface of the boiler, as this is a flexible figure.

From our overall figure of 24 cub. ft. combustion gases per lb., it is easily found that the specific weight will amount to :—

$$S = 1 \div 24 = 0{\cdot}0417 \text{ lbs./cub. ft.}$$

The volume of the surrounding air of the smoke stack at a tropical day temperature of 30°C. (86°F.) at atmospheric pressure is also to be obtained from formula (84) : $2116{\cdot}27 \times V = 53{\cdot}33 \times 546$. $V = 13{\cdot}75$ cub. ft./lb. and the specific weight $S = 0{\cdot}0727$ lbs./cub. ft.

It will be obvious that the atmospheric pressure at levels well above the sea is less than 2116·27 lb./sq. ft. but cane sugar factories, as a rule, are situated in the low lands slightly above sea level and our assumption of atmospheric pressure thus holds good.

The difference in specific weights of the gases in the stack and the surrounding air will thus amount to : $0{\cdot}0727 - 0{\cdot}0417 = 0{\cdot}0310$ lbs./cub. ft. A stack of 100 ft. height will obviously produce a draught of 3·10 lbs./sq. ft. at the given temperatures.

The draught is measured in inches water column and as a cubic foot of water weighs 62·32 lbs., an inch water column will exert a pressure of $62{\cdot}32 \div 12 = 5{\cdot}19$ lbs./sq. ft.

A draught of 3·10 lbs./sq. ft. is thus equal to $3{\cdot}10 \div 5{\cdot}19 = 0{\cdot}6$ in. water column. For bagasse furnaces 0·6 in. to 0·8 in. water column is the practical draught at the foot of the chimney, and thus a chimney height from 100 to 135 ft. should be sufficient, but it should be recollected that the combustion gases will cool down in traversing the stack and, moreover, through excess of air, a temperature of 520°F. is not always available and lower stack temperatures will reduce the stack losses. With a mean chimney temperature of 350°F. (176°C.) a draught of 0·44 in. water column can be obtained per 100 ft. stack height, and normal chimney heights for medium and large size cane sugar factories are between 150 and 200 ft.

The area of the top opening of the chimney may be taken as an overall figure at 3·5 sq. ft. per 100 h.p. boiler rating (equivalent to about 1000 sq. ft. H.S.).

2.—Wind Pressure against Stack.

The chimney needs to have sufficient resistance against any prevailing wind pressure, as a collapsing chimney may cause great damage to adjacent buildings and equipment. Chimneys were formerly built square, as they could then be erected with common brick, but nowadays the round or octagonal type is extensively used.

The maximum wind velocity in some countries may not reach 60 miles per hour (100 km./hr.) which limit is usually classed as stormy weather, but in those countries where cyclones or typhoons prevail, maximum wind or hurricane velocities up to 150 m.p.h. (240 km./hr.) have been recorded.

The corresponding wind pressures, calculated according to formula (30) in Chapter II, are tabulated as follows :—

$V =$ 50 m.p.h.	....	$P =$ 8 lbs./sq. ft.
$V =$ 75 ,,	....	$P =$ 18 ,,
$V =$ 100 ,,	....	$P =$ 32 ,,
$V =$ 125 ,,	....	$P =$ 50 ,,
$V =$ 150 ,,	....	$P =$ 72 ,,

Generally, 100 to 120 miles wind velocity per hour is considered as a basic figure for chimney calculation, but for those countries affected by tornados, cyclones or hurricanes, a higher basic figure should be applied.

The wind pressure loses part of its force close to the ground through friction over obstacles like trees, etc., but it is safe to assume the full wind pressure acting on chimneys and other high structures of a cane sugar factory.

On curved surfaces the wind does not act on the full exposed projection and for semi-circular shapes, the total wind pressure may be taken as :—

$$W = A \times 0{\cdot}666 \times P \qquad (87)$$

where W = total wind pressure in lbs.
A = the exposed or projected area in sq. ft.
P = wind pressure in lbs./sq. ft.

This wind pressure W will act at the centre of gravity of the exposed area and in the case of a straight cylindrical chimney this centre will lie at half the chimney height. For chimneys of conical shape, the exposed area is a trapezoid and the centre of gravity will lie below the middle of the chimney height.

The wind pressure will thus cause a tipping or collapsing momentum of the value :—

$$M = W \times L \qquad (88)$$

where L is the height of the centre of gravity above the chimney base in ft. and M the tipping momentum in ft./lbs.

3.—Chimney Designs.

Four types of chimneys are used for cane sugar factories :—

(a) Brick chimneys.
(b) Steel chimneys supported by guys.
(c) Concrete chimneys.
(d) Self-supporting steel chimneys.

Brick chimneys require specially profiled bricks and have to be erected by expert chimney builders. The construction gives lasting qualities and where a rigid foundation is available, such chimneys can be used to advantage.

But in countries affected by earthquakes, the brick chimney should be avoided and the three other types are to be preferred.

From *Fig.* 314, showing a *Cross Section of a Brick Chimney,* it is evident that the wind pressure has a tendency to tumble the chimney over to the left and the right half circle gives no support at all, as it is assumed that the mortar of the joints will not show resistance against tensional stresses.

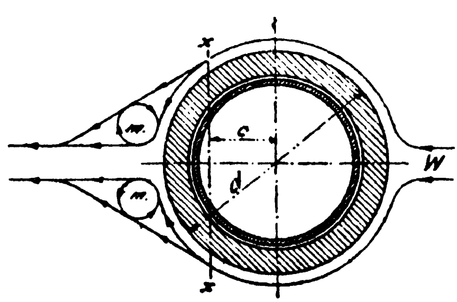

Fig. 314.—Cross Section of Brick Chimney.

The gravity axis x-x of the left hand semi-circular ring is thus the tilting axis, and for the stability of the chimney the following formula has to be adhered to :—

$$M = W \times L < P_w \times c \qquad (89)$$

P_w being the total chimney weight above the section to be considered and c the gravity distance from the chimney centre axis in feet.

A safety factor of 3 is the accepted standard for the maximum wind-pressure prevailing. The wind, moreover, causes whirls at m on the lee side of the chimney and the sucking effect will sometimes draw the smoke downwards, when it emerges from the top of the stack.

The safety of a brick chimney, therefore, is proportional to its weight.

A refractory lining is usually built inside the chimney to about half the chimney height with the object not only to give protection against excessive temperatures, but also as an insulating layer against heat losses, as an air chamber is formed between the inner and outer brick shell. The refractory lining is generally half a brick thick and it has to be well supported by the outer shell at intervals of 25 ft. The top of the refractory lining has to be capped, to prevent the entrance of rain water into the air chamber. The author knows an instance where damage was done through an uncapped refractory lining. But a refractory brick or two must be left out close to the top, to relieve the air chamber of gases produced by the heat in the stack.

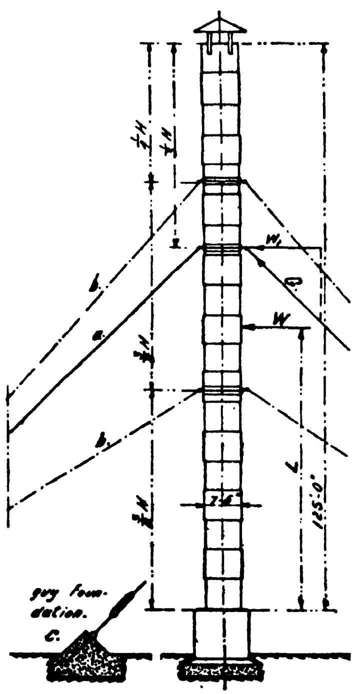

Fig. 315.—Supported Steel Stack.

For a 3000-ton sugar factory a brick chimney has been built with 12 ft. inside diameter at the top, 15 ft. at the bottom and a height of 200 ft. The wall thickness at the bottom was 3 ft. 6 in., and at the top 13 in., where a cast iron ring was provided as a protection. The flue entrance measured 130 sq. ft. for a rated boiler capacity of 3900 h.p.

A *Steel Stack* supported by cable guys is shown in *Fig.* 315, erected under the author's supervision for 800 h.p. rated boiler capacity. For smaller size factories and those places far from industrial centres, where stack builders cannot be engaged, this type has found a wide application.

The base plate is made of steel, this being preferable to a cast iron base plate, which might easily crack. A ladder is provided on the outside, as inside ladders are subject to heavy corrosion.

To prevent rain water entering, a hood is provided on the top, but most chimneys are without this attachment.

A $\frac{3}{8}$ in. material should be used at the lower end and $\frac{5}{16}$ in. up to the top. A saving in material is not always a paying proposition. Each belt is made in two halves for easy erection. Some engineers prefer bolted connexions, but these chimneys can likewise be riveted when compressed air is available. To renew corroded plates the chimney has to be dismantled as far as the place where the replacement has to be made. Riveting of chimney parts on the ground, with joints of angle iron, is only feasible when the necessary hoisting gear is at hand, but it should not be overlooked that booms of over 150 ft. height are clumsy pieces of equipment. For the erection of half belts the hoisting gear can be conveniently attached to the longitudinal bolt or rivet holes of the composite pieces.

Four guys of 1 in. dia. are attached at two-thirds of the chimney height, so as to take up the forces caused by the wind pressure. The tilting or wind force momentum obviously amounts to :—

$$M = H \times D \times 0{\cdot}666\, P \times H \div 2 = 0{\cdot}333\, H^2 \times D \times P.$$

The horizontal reaction at the locus of the guys W_1 will thus amount to :—

$$W_1 = 0{\cdot}333\, H^2 \times D \times P \div 0{\cdot}666\, H = 0{\cdot}5\, H \times D \times P.$$

where D is the stack diameter, H the height, both in feet, and P the wind pressure in lbs./sq. ft.

At a 45° inclination of the guys and assuming that in the most unfavourable case only one guy is acting on the windward side, the guy force G has the value in lbs. :—

$$G = \sqrt{2} \times 0{\cdot}5\, H \times D \times P \simeq 0{\cdot}7\, H \times D \times P \ldots. (90)$$

The guys should be well anchored and the author has used concrete foundations as shown with a 1½ in. tie-rod, turnbuckle and thimble to advantage. An anchorage on wooden poles is not always to be relied on.

Chimneys have collapsed ere now not on account of insufficient guy support, but by the sagging of the part below the guys. Where wind pressures are heavy, two sets of guys *b*, the top one at one-fourth of the height from the top and the other midway between the base and the former, will give additional safety.

The chimney of *Fig.* 315 weighs about 23 short tons and the socket is made of brick, so as to have the boiler flues above ground level. In case this is not practicable, underground flues have to be emptied of any ground-water before the crop is started.

The brick socket is laid on a heavy concrete foundation base of sufficient size to have a low specific soil bearing of between 8 and 16 lbs. per sq. ft. for soft soil. In clay soil pile driving is sometimes required.[1] Rock or hard sand strata will allow a higher soil bearing.

Lightning arresters should be placed on the chimneys of cane sugar factories and although they are not always found on steel stacks they are essential for brick and concrete chimneys, as lightning may easily destroy the chimney at the top.

In case of a high ground water level when laying a chimney foundation, a well of greater depth than the foundation pit should be dug close to the latter and be provided with a good pumping outfit. Boreholes with low head centrifugal pumps above ground level will also lower the ground water level.

In those countries where there are official regulations for buildings and steam appliances, the drawings of the chimney and the boilers should be submitted to the local authorities for approval, before the work is started.

A *Reinforced Concrete Chimney* built for a tropical cane sugar factory is shown in *Fig.* 316. This type combines light weight with great strength and has been supplied in many cases, but needs expert concrete workers to supervise the construction. The shape is generally of circular section, but octagonal concrete chimneys have been built as well, which allow additional reinforcement at the corners.

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1 See Chapter I, Formula (4).

The chimney of *Fig.* 316 has a refractory lining, well capped to prevent the entrance of rain water between the lining and the chimney shell. A wide base, as shown in *Fig.* 317, is provided, reinforced by a 5 in. pitch netting of ¾ in. square corrugated bars at about 3 in. from the base and a coarser netting 1 ft. 9 in. higher. The perpendicular reinforcing bars are well fastened by bents in this netting. The number of these reinforcing bars is indicated in the figure.

The resistance of a reinforced concrete chimney is not decided only by the chimney weight; the reinforcing bars will give great additional tensile strength and the combined moment of resistance or section modulus of the concrete area with reinforcing enters into the calculation. Bending performance therefore must be considered. Formula (89) will thus change into :—

$$M = W \times L < P_w \times c + \frac{R \times P}{12} \quad .. \ (91)$$

where R is the combined moment of resistance in cub. in. and p the combined allowable stress in lbs./sq. in.

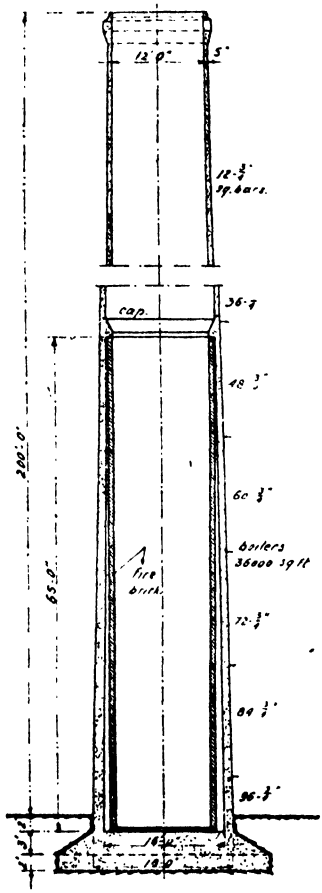

Fig. 316.—Chimney (reinforced concrete).

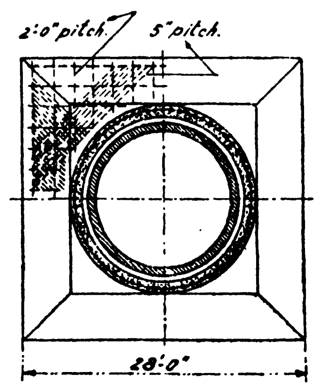

Fig. 317.—Chimney Base of Fig. 316.

At the opening for the flue at the stack bottom, additional bars and wall thickness have to be provided. Nevertheless the concrete chimney requires only light foundations and is resistant to cyclones and earthquakes.

The fourth type is the *Self-supporting Steel Chimney* as shown in *Fig.* 318. The plan view is shown in *Fig.* 319 and the anchorage in *Fig.* 320. This chimney is of American design, but the author does not know the maker; attention was called to its construction, as it had withstood a wind pressure of about 42 lbs./sq. ft. during a cyclone.[1] The lower belts are double riveted with ¾ in. rivets on the circular joints and the shaft is well reinforced by inside

[1] See also Plate 9.

and outside angle rings to prevent collapse by buckling. On top is a walking platform (not shown in the drawing) and a circular flat iron or rail for a trolley, on which a tackle can be attached for use when painting and repairing.

The base is of inverted paraboloidal shape, but it can be designed equally as a cone at reduced manufacturing cost.

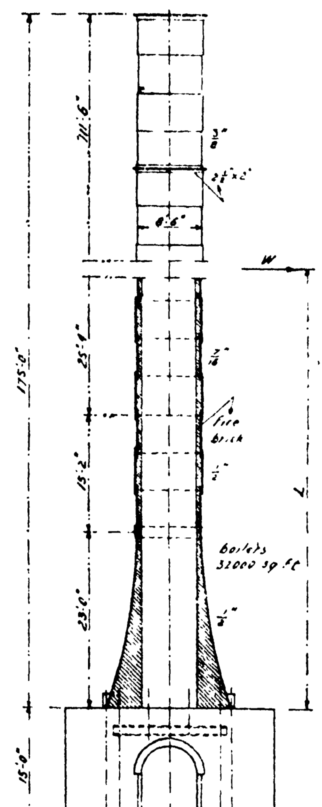

Fig. 318.—Self-supporting Steel Chimney.

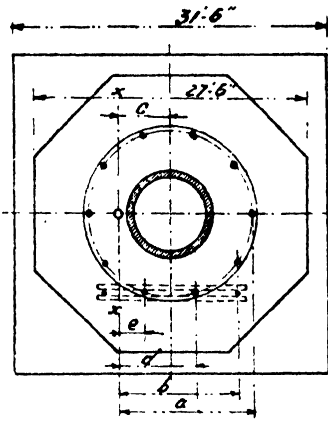

Fig. 319.—Plan of Fig. 318.

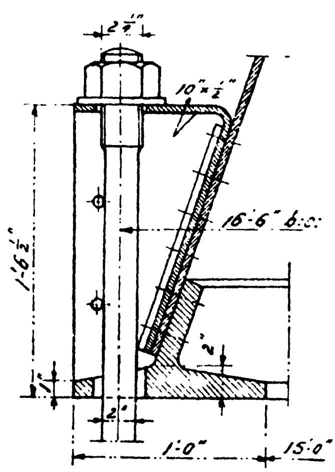

Fig. 320.—Anchorage of Fig. 318.

The circular section will have to satisfy formula (91), but the anchorage must be well dimensioned to take up the total tilting momentum. The weight of the chimney is about 44 tons, whereas the foundation weighs about 280 tons. A refractory lining is built up to about half the height and as it is laid with thin joints, it will help to maintain the circular shape when the chimney is under bending stress.

The tilting formula for the anchorage thus reads :—

$$M = W \times L < P_w \times c + P_f/10 \times a + P_f/5 \times (b+d+e)$$
$$< P_w \times c + P_f/10\,(a + 2b + 2d + 2e) \ldots . (92)$$

where P_f is the weight of the foundation block in lbs., and a, b, d, e in feet.

The ten anchor bolts obviously have each to support an average stress of one-tenth of the block weight.

From actual practice the following dimensions of chimneys can be cited :—

Capacity 1400 tons of cane	12 ft. 6 in. dia.,	height	165 ft.
„ 700 „	9 ft. 0 in.	„	150 ft.

The arrangement of the *Boiler Flues* is generally as shown in *Fig.* 321, although the actual shape of the flues of this installation was somewhat different. Sharp corners are avoided and in case of a chimney arrangement

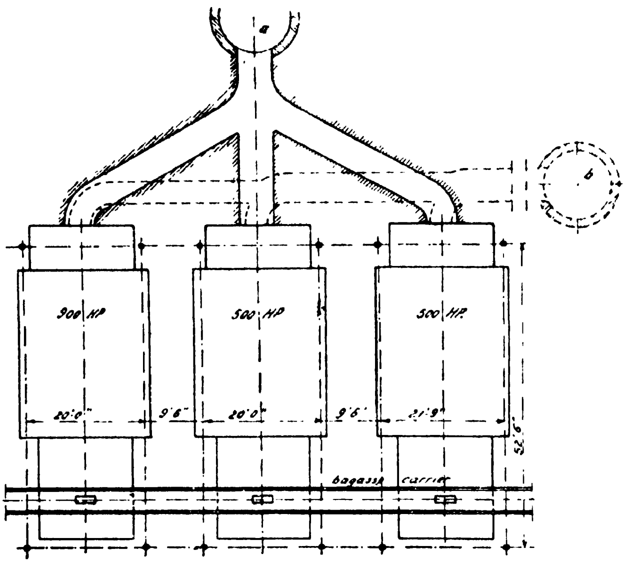

Fig. 321.—Arrangement of the Boiler Flues.

at *c*, equal draught should be aimed at for all three boilers. Sometimes the chimney is arranged at the end of the flue, as drawn at *b* and the flue section should gradually increase towards the stack. For large boiler plants two chimneys are sometimes provided, one at each end of the main flue ; this has given good operating results.

4.—Feed-Water.

For efficient boiler operation the feed-water should be supplied in a pure state and with a temperature as close as possible to that of the water within the boilers.

The feed-water pumps are either piston, plunger or centrifugal types. These pumps should be brass-lined to avoid corrosion of the moving parts.

Piston and plunger pumps should not pump into a closed feed line, as the pipe line will then burst before the pump is stopped ; in any case a spring-loaded safety valve should be fitted at the end of the line, the discharge flowing back to the source whence it came.

Steam-driven duplex pumps have a higher steam consumption than flywheel pumps. Centrifugal pumps for boiler feed are either electrically or turbine driven and provided with more than one impeller for the manometric head required. The author has furnished the electrically-driven type with motors at 3,600 r.p.m. for alternating current of 60 cycles. This high speed type is of a reduced size and the cost is lower.

In contrast to the piston or plunger pump, the centrifugal type can pump against a closed feed line and in each case the maximum pressure should be obtained from the pump curve at zero delivery. These pump curves or characteristics are supplied with every centrifugal pump. The maximum pressure should be about 25 per cent. above the working pressure of the boilers.

At the boiler front a non-return valve has to be fitted, so that the water within the boiler never flows back into the feed line under any circumstances.

The feed-water pump should have at least 50 per cent. more capacity than the normal steam production of the boilers. And where one pump only is sufficient for the whole battery of boilers, a spare pump of the same size must still be provided for emergency purposes when a pump breakdown occurs.

In cane sugar factories nearly all the steam produced by the boilers is condensed in the calandrias or coils of the heating bodies, so a pure condensate can be obtained, sometimes carrying a small amount of lubricating oil, when reciprocating steam engines are used as prime movers. This oil should be eliminated by a feed-water filter and the boiler water be kept slightly alkaline.

The condensation water has a temperature close to the boiling point and feed pumps have to be placed low, so that a steady flow by gravity is assured to the pump intake. The suction lines have to be of ample size, and a water velocity of about 3 ft./sec. should not be exceeded, as the pump suction will readily fail with hot water. With centrifugal pumps any failure of the suction might give rise to heavy water hammer when the suction is taken up again. The pressure lines can be dimensioned for a water velocity of from 6 to 10 ft. per second.

A *Closed Condensation Water System* is diagrammatically shown in *Fig.* 322.[1] There are two sources of condensate which are used for boiler feed, the one being the condensate of the exhaust steam of the prime movers and the other from the live steam used in the heating bodies. Of the assumed vacuum pans, two are fed by exhaust and one by reduced live steam, and the condensate is discharged by gravity through the main condensate line into a closed tank, which is covered by insulating material. Operating engineers as a rule prefer separate pipelines from each source of condensate, but a main line can be used when in each branch a non-return valve is arranged; and moreover a discharge *d* to the ditch and a sampling tube *s* with a small size valve, so as to control each branch in respect to sugar, which might get into the condensate through a leaking coil, etc.

Where live steam is used, care has to be taken that it cannot enter the closed condensate system. When the pressure is only 10 to 15 lbs. above the exhaust pressure, a syphon, having the seal length *x*, can be inserted. Another safe method is to use a steam trap, which will shut off any entry of live steam.

Those supplies of condensate which have not sufficient gravity head need to be pumped into the closed tank by a pump *y*.

To replace the losses of hot condensate which will always occur, a make-up supply of condensate from vapours of atmospheric pressure, or slightly below, has to be pumped into the closed tank by a pump *z*. A fresh water connexion can also be used for this make up, when hot condensate is not at hand.

[1] See the article of G. W. CONNON, Haw. Sug. Tech. Ass., 1928.

The closed tank is connected on top with the exhaust steam line ; and with an exhaust pressure of say 8 lbs. a feed water temperature slightly below 234°F. will be maintained. The make up condensate or fresh water should be sprayed in through a perforated pipe when entering the closed tank, so that the exhaust steam will heat this water.

A water level gauge has to be provided, as well as a thermometer and a manometer ; and at *a* an overflow of ample size (6 to 8 in.) is arranged, having a 26 ft. syphon. As for each lb. pressure 2·31 ft. seal length is required (34 ft. for the atm. pressure of 14·7 lbs.), the seal will be broken at about 11 lbs./sq. in. pressure within the tank. The overflow discharges the excess water when the pressure has reached 11 lbs. and the steam will escape through the roof at *f*. Provision has to be made for this syphon to be re-sealed with water, when the pressure has dropped below 11 lbs. and the water level in the tank is not of sufficient height to perform this duty. A float-operated discharge valve is sometimes also provided for excess water, instead of the hand-operated one.

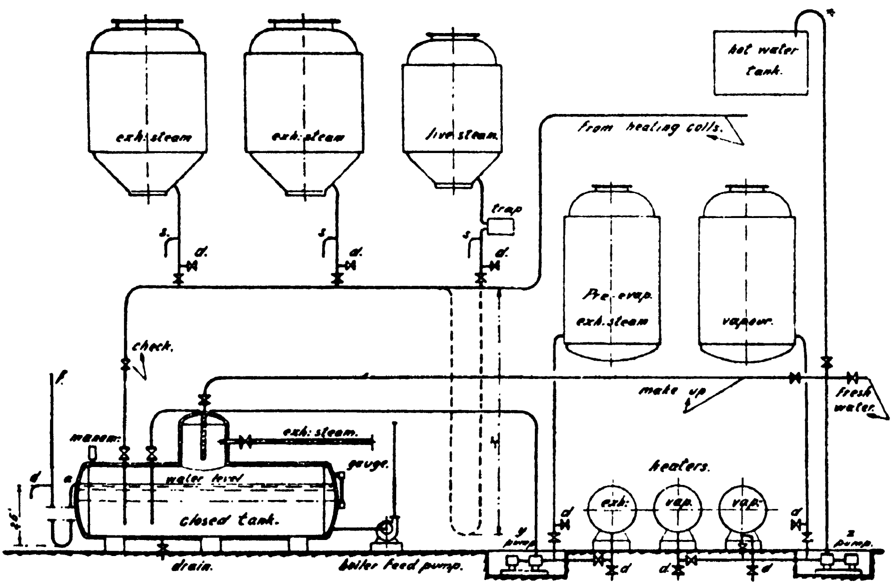

Fig. 322.—Closed Condensation Water System.

Often a coil or perforated pipe is laid at the bottom of the tank, so as to be able to heat fresh water, e.g., when the tank has been cleaned after sugar has entered it.

Favourable operating results have been obtained with this closed condensate system and the boiler feed-water is practically de-aerated.

The open condensate system is built on similar lines, save that the receiving tank is not closed. Heat losses in this type are considerably higher and the feed-water temperature will not exceed 205°F. (96°C.) and is generally less, when the condensates are not properly separated. Condensates of lower temperature from the last evaporating bodies, the first juice heaters, etc., should be stored separately for imbibition, dilution and cleaning purposes.

Open condensate or feed water systems can be conveniently fitted with a V-notch water meter for measuring the feed-water, and a control of the steam production is thus obtained. For the closed system closed water meters are used, as well as recording Venturi meters.

Where water of a certain hardness has to be used, as is the case with nearly all well water, it will pay the operator to have a water softener installed for this purpose. This not only refers to boiler feed-water, but also to wash-water for the centrifugals and the like.

Feed-water is sometimes softened by the lime-soda process, which needs sedimentation tanks of a certain size and sodium aluminates are also added.[1]

The hardness is due principally to compounds of lime (calcium) and magnesia, which will settle out as a sludge when heated in the boilers or form boiler scale, which reduces greatly the boiler efficiency when not removed regularly, and through its insulating capacity may be the cause of overheating of the boiler material.

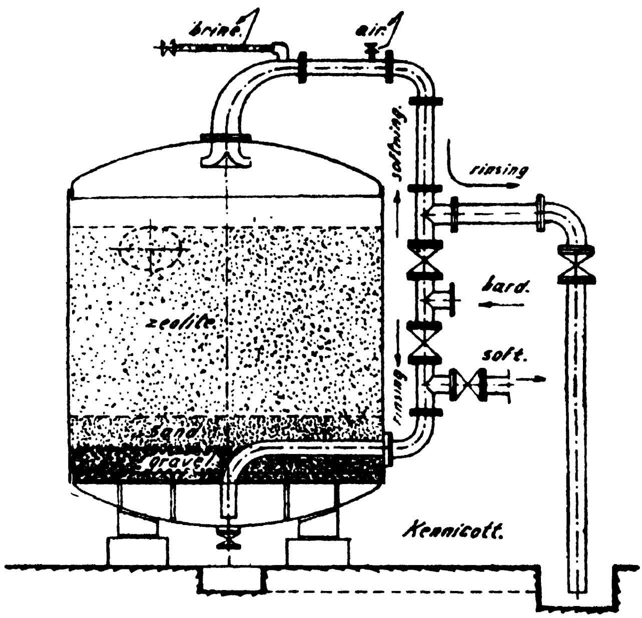

Fig. 323.—Water Softener.

As far back as 1850 a British chemist, J. T. WAY, made experiments with certain types of soil (anhydrous silicates of complex chemical composition called *zeolites*) giving favourable results as a water-softening medium. These zeolites can be easily reactivated by a solution of common salt, and manufacturers now produce zeolites of a porous structure especially adapted for water softening.

A water softener of recent design is shown in *Fig.* 323.[2] The hard water enters at the top as indicated by arrows and the soft water is withdrawn at the bottom, after having traversed the bed of zeolites. For reactivation a 10 per cent. mixture of common salt is introduced at the top and circulates the same way. After reactivation is complete, the zeolites which do not dissolve are rinsed by water flowing in the opposite direction and the apparatus is then ready for service again.

1 See *Int. Sugar Jl.*, 1935, page 366. 2 See *The Engineer*, 1935, page 264.

Generally, two softeners are installed, one for softening while the other is reactivating. The softening of water having 10 grains hardness per gallon will require about 10 minutes, and as the zeolites occupy about 2·5 times the volume of the water that is traversed, every 1000 gals. of water to be softened per hour will thus require a net tank volume of :—

$$1000 \times 10 \times 2{\cdot}5 \div 60 \simeq 420 \text{ gals.} \simeq 67 \text{ cub. ft.}$$

For less hardness, the capacity is increased, and when overloaded the performance will be less efficient and zero hardness not be obtained. Softened water can be stored for any length of time and the apparatus is generally designed for low pressure, it being a riveted or welded construction.

Sludge in the water will fill the pores of the zeolites and may make it useless, so water with suspended impurities should be filtered over a sand bed before being pumped into the softener. A pressure gauge will indicate the good mechanical working of the softener.

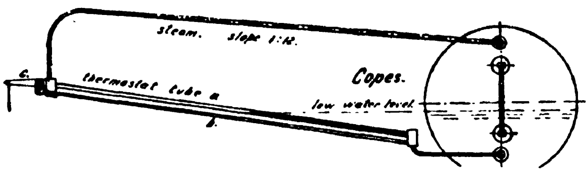

Fig. 324.—Feed-Water Regulators.

For regulation of the boiler feed, automatic *Feed-Water Regulators* are used to advantage and a widely used type is shown in *Fig.* 324. An inclined thermostat tube *a*, made of special material having a large linear expansion coefficient, is at one end firmly attached to a rod *b*. The inclination of the thermostat tube is arranged in such a way that the range of the tube filling is within the lowest and highest water levels in the boiler.

At the lowest point a connexion is made with the water space of the boiler, whereas the upper part is connected with the steam space in such a way that no air pockets can be formed. The water in the thermostat tube will cool off rapidly and when the water level falls, the steam will heat and thus expand this tube. This expansion is transferred by a bell crank *c* to a regulating valve in the boiler feed line and automatic control of the boiler feed is thus obtained.

5.—Steam Accumulators.

The steam consumption of a cane sugar factory is not always at a constant level and with high consumption the steam pressure generally drops, whereas with reduced steam outlay the pressure will rise. From steam flow charts of a raw sugar factory which the author has, the following maximum fluctuations are taken :—

Prime movers : Consumption 70 per cent. of total steam produced.
  Fluctuations : plus 20 per cent. to minus 20 per cent., being plus 14 per cent. to minus 14 per cent. of total steam.

Boiling-house : Consumption 30 per cent. of total steam produced.
  Fluctuations : plus 65 per cent. to minus 55 per cent., being plus 19·5 per cent. to minus 16·5 per cent. of total steam.

Total combined fluctuations in steam consumption : from plus 25 per cent. to minus 25 per cent. of the average.

In *Fig.* 325 is given an extract from such a *Steam Consumption Chart* and it will be obvious that the boilers have to cover the peak loads, but the boiler and its settings do not always have sufficient accumulated heat to cover these peaks efficiently and pressure fluctuations will be the inevitable consequence, if not prevented by the intervention of the operators.

A logical step, therefore, has been to look for an accumulator of heat and as steam cannot be stored on account of its volume, a *Steam Accumulator* is really a large tank, filled with hot water. This large volume could not possibly be stored within the boiler and therefore a separate holding unit has been designed.

Originally developed for the cellulose industry, it now has been installed in several tropical sugar factories and refineries.

The water in the steam accumulator is heated by steam and its optimum temperature will be that corresponding with the steam pressure in the boilers. Superheated steam will be readily saturated as soon as it comes in contact with the water of the accumulator. It will be apparent that the capacity of the accumulator will increase and its required volume decrease with a greater heat drop, which is equivalent to a greater pressure drop and the accumulator therefore will supply steam at a considerably reduced pressure than that prevailing in the boilers.

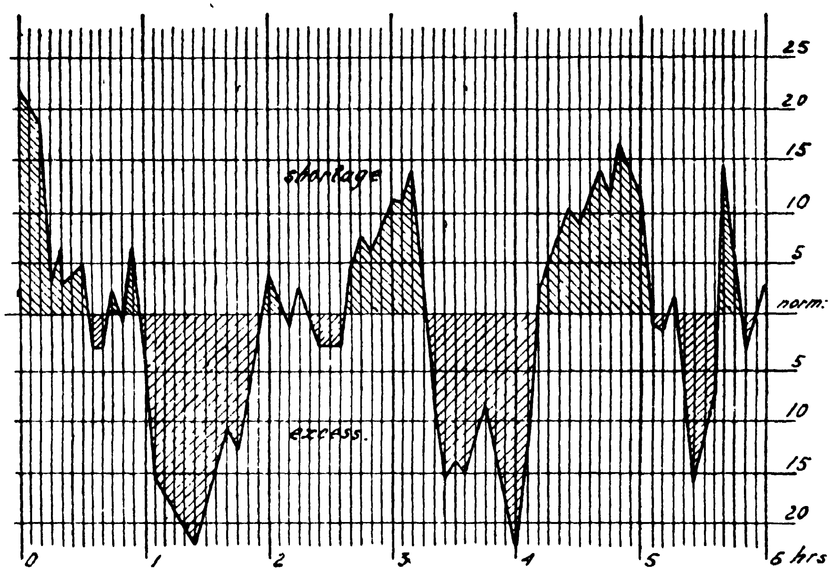

Fig. 325.—Steam Consumption Chart.

For average conditions in cane sugar factories, the author assumes a steam pressure of 10 lbs./sq. in. (exhaust steam) for the heaters and evaporators and 30 lbs./sq. in. for the vacuum pans.

If S be the sensible or liquid heat per lb. of water, thus S_{10} corresponding to a steam pressure of 10 lbs./sq. in.; S_{100} corresponding to 100 lbs., etc.; and L the latent heat of the steam, thus L_{10} for 10 lbs. pressure, etc., and allowing 1 cub. ft. $=$ 62·43 lbs. of water, the following amounts of steam Q in lbs. can be produced from each cubic foot of water in the steam accumulator :—

$$Q = (S_y - S_x) \div L_x \times 62{\cdot}43 \ldots\ldots\ldots\ldots\ldots\ldots (93)$$

Thus for 140 lbs. pressure $S_y = 333$ B.Th.U. per lb.
for 30 lbs. „ $S_x = 243$ „ „
for 30 lbs. „ $L_x = 928$ „ „

and Q therefore $= 90 \div 928 \times 62{\cdot}43 \sim 6$ lbs. steam per cubic foot of water in the accumulator.

In *Fig.* 326 a *Steam Accumulator Production Chart* is compiled for charging steam (boiler) pressures from 0 to 225 lbs./sq. in. gauge pressure and for discharging steam of 10 and 30 lbs./sq. in.

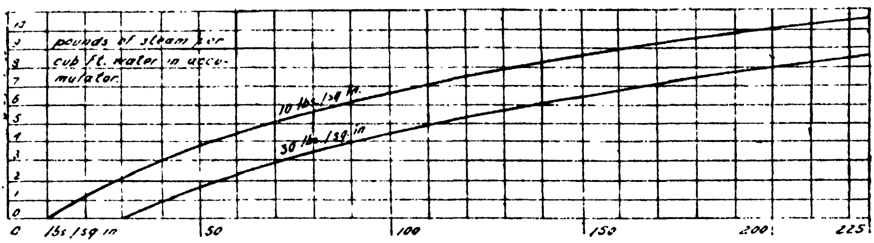

Fig. 326.—Steam Accumulator Production Chart.

The *Scheme of a Steam Accumulator Installation* for a cane sugar factory is given in *Fig.* 327. The boilers supply live steam at 125 lbs. pressure to the turbo-alternator and the other prime movers, which produce exhaust steam at 8 lbs./sq. in. For the vacuum pans a reduced steam pressure of 30 lbs./sq. in. is required.

When the boiler pressure is above 115 lbs./sq. in., the valve 1 is opened and live steam flows to the nozzles of the steam accumulator, which are

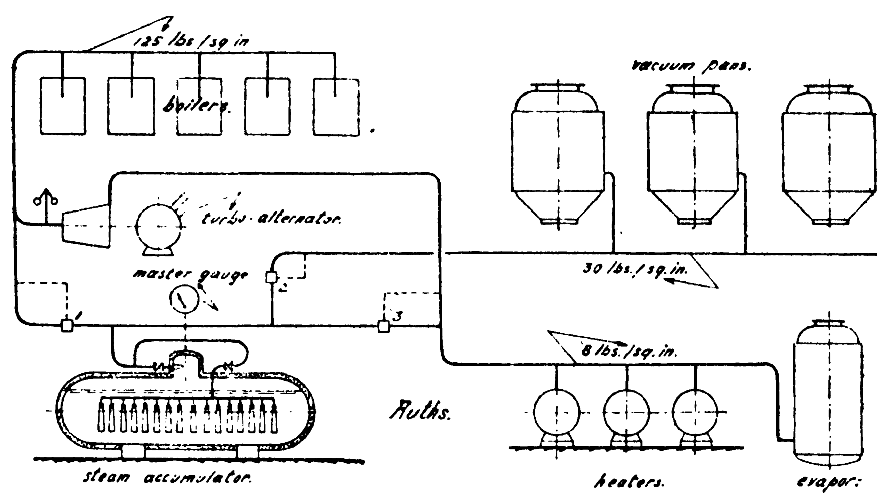

Fig. 327.—Scheme of a Steam Accumulator Installation.

surrounded by conical or bell-shaped circulation pipes for noiseless operation and good absorption of heat. The automatic overflow valve 2 maintains a pressure of about 30 lbs. in the vacuum pan branch ; the valve will close when 32 lbs. pressure has been reached.

As soon as the pressure of the exhaust steam drops below 6 lbs. the valve 3 opens and will close when 10 lbs. has been reached.

When the boiler pressure drops below 115 lbs., valve 1 will close and the accumulator now supplies valves 2 and 3 as may be required. In the dome of the accumulator is located the nozzle-shaped outlet, which will allow a maximum discharge within the capacity of the accumulator, while preventing water being forced along through priming. Moreover, non-return valves will prevent steam entering the accumulator through the discharge outlet or water escaping through the steam charging pipe.

There is very little, if any, water required for the make-up, as the high pressure steam has about 2 to 3 per cent. more heat value per lb. than the low pressure steam, and moreover radiation may cover the difference. This radiation amounts to about 0·2 B.Th.U. per sq. ft./hr. per degree F. temperature difference, the insulating material being about 4 in. thick.

The accumulator is mounted on a fixed support at one end and a roller support at the other for free expansion, and it can be placed on the ground level, or on any platform, or even on top of a flat roof of sufficient bearing strength.

For pan work, where a difference in steam pressure, which means a difference in temperature, may cause inconvenience, the steam accumulator has considerable advantages.

The automatic overflow valves are of high-class mechanical construction, operated by oil under pressure. The master gauge, moreover, will indicate when the accumulator is charging or discharging.[1]

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[1] See also the article of J. LEWIS RENTON, Haw. Sug. Tech. Assoc., 1928.

CHAPTER XIV.

BAGASSE EQUIPMENT.

INTERMEDIATE CARRIERS—BAGASSE CARRIERS—BAGASSE PRESSES.

1.—Intermediate Carriers.

The bagasse has to be conveyed from the discharge roller of a previous mill to the feed roller of the next one, and distances between mills have to suit the centres of the mill driving gear, so some form or other of *intermediate carrier* is required as the conveying agent. Three different designs are used, viz. :—

(1) Apron carriers.
(2) Drag carriers.
(3) Sliding carriers.

The first type, the *Apron Intermediate Bagasse Carrier,* is shown in *Fig.* 328, as used in Java. The discharge roller of the right hand mill delivers the bagasse on a sloping chute, having a minimum inclination of 45° with the horizontal, thus descending on to the carrier apron, which is composed of two strands of malleable iron pintle chain, to which wooden slats are attached between. As bagasse has been brought to an advanced state of fineness in our present-day milling plants, a considerable amount of fine trash may fall through the gaps between the slats, and for this reason double-beaded or corrugated overlapping steel slats, similar to those shown in *Fig.* 121, are nowadays preferred.

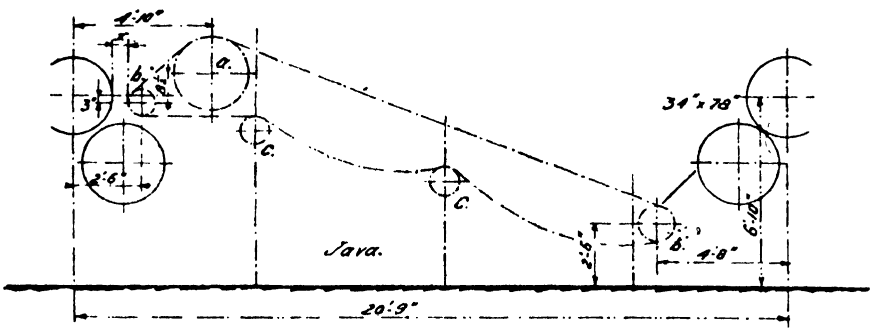


Fig. 328.—Apron Intermediate Bagasse Carrier.

The driving shaft *a* is the upper one, *b* shows the reversing sprockets and *c* the idler pulleys, which carry the returning apron. For short centre distances, these idlers are omitted, as is common practice in America, but the chains will then suffer more wear.

The part of the carrier between *a* and *b* at the left side will push the cane towards the mill feed and thus assist in the proper feeding performance. The distance *x* should be of sufficient size (8 to 12 in. or more, according to the size of the mill) as otherwise choking will ensue. The carrier speed is preferably 7 to 10 per cent. more than the peripheral speed of the mill rollers, so a pushing action on the feed side will result. It is obvious that with a carrier speed less than the peripheral roller speed, the bagasse will not be pushed but be dragged into the mill.

The *Chain Support* of this type of carrier is shown in *Fig.* 329; the chain guiding angles are arranged outside the carrier wall at *u*, so that the chains will not be soiled by the bagasse or dirt and thus will have a longer life, as wear will be less. Moreover, with this design the chains can be properly lubricated. This is a very good arrangement for guiding the conveying chains. The latter have right and left hand attachments for the left and right hand chains respectively, and the ultimate tensile strength is about 9600 lbs. for each strand.

Wooden beams *w* give support to the wooden slats. The slats are made from a tough hard wood, which should not be brittle. The guide beams *w* are preferably made of hard wood, having a high resin content as this will give long wear and offer less friction.

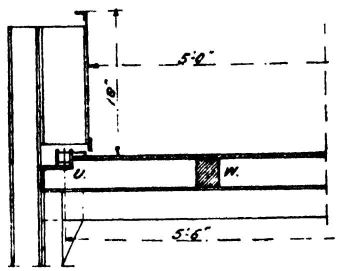


Fig. 329.—Chain Support.

The *Drag Type Intermediate Carrier* is shown in *Fig.* 330, as measured by the author from an existing installation. Two shafts *f* and *g* carry each three sprockets, whereas two 3 in. pipes connect both. The driving shaft *g* can be lifted out when the driving chain is detached, and the bearings of shaft *f* are sometimes spring-loaded. The conveying chains are of the roller or pintle type, about 3 in. pitch, having each about 28,000 lbs. ultimate strength. At each fourth link an attachment is provided, on which the scrapers of $2\frac{1}{2}$ in. × $2\frac{1}{2}$ in. × $\frac{1}{4}$ in. angle iron are bolted. These scrapers should be divided (cut) in the middle, so as to offset the effect of any unequal chain stretch in the three strands.

The carrier chute is flush inside and water-tight, and the bagasse will be pushed towards the next mill, assuring an even feed. The clearance of the angle flights should be calculated at a specific weight of the bagasse of about 15 lbs. per cub. ft., so as not to exert too heavy a pressure on the chute bottom. Three sets of guide sprockets hold the chain down, while one set supports the return chain.

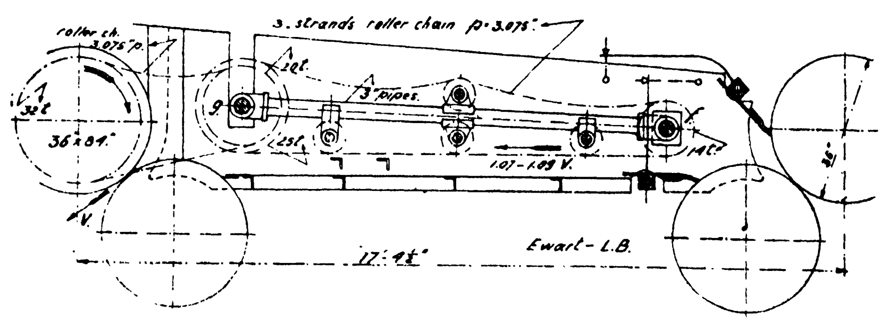


Fig. 330.—Drag Type Intermediate Carrier.

As these carriers are water-tight, they are fit receptacles for imbibition and maceration. Moreover, there is more head room underneath the carrier, thus making this part of the mills more accessible. But the chains will wear more rapidly than is the case in *Fig.* 329.

Instead of angle flights, sometimes a comb construction or rake is used.

The arrangement of the mill scrapers of the previous mill can be observed from *Fig.* 330.

An *Improved Apron Carrier* of recent design is shown in *Fig.* 331. This is of the pushing type and its downward slope forms a tangent to the front roller periphery. As the shaft *a* is the main driving shaft (which is a necessity as the extension of shaft *b* will not clear the mill cheeks), the downward slope or pushing part of the carrier will bend inwards under heavy load. An intermediate chain transmission of double-stranded roller chain, $1\frac{3}{4}$ in. pitch, is therefore arranged between the shafts *a* and *b*, underneath the carrier apron of overlapping steel slats, and *b* may thus be considered as a driving shaft also.

The carrier is driven from a chain wheel on the front roller of the charging mill. Moreover, the return strands of chain are supported at *d*, which will add to the life of the chains. These supports are connected by a plate, so as to form a drip pan and the space underneath the carrier will thus remain clean.

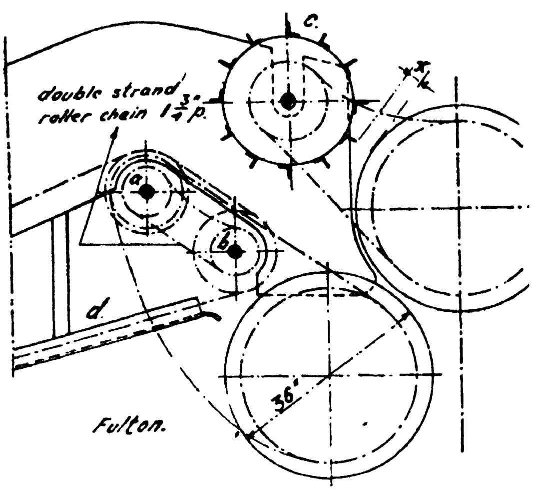


Fig. 331.—Improved Apron Carrier with Floating Forced Feed Roller.

Heavy imbibition or maceration, as well as the finely disintegrated bagasse of our modern milling plants, can cause difficulties in the feeding of the mill, and in Java, therefore, *feed rollers* are widely used, these being cast iron rollers of ridged construction, which can swing around the top roller centre. The feed roller is driven by a chain drive or a double set of gears from the feeding top roller, so as to maintain the proper direction of rotation.

In *Fig.* 331 is shown a *Floating Forced Feed Roller*, this being a drum of sheet steel 24 in. in diameter, with twelve angle irons attached and driven from the top roller. The distance x has to be kept as small as possible and as the angle irons may carry along some trash or bagasse, the cast iron ridged roller just mentioned serves the purpose of obviating this difficulty.

The power consumption of bagasse carriers is high and although the author has no exact data on hand as to this point, due to the fact that all these intermediate carriers are driven from the mill rollers and not independently, breakages have shown the heavy stresses to which these carriers for short centre distances are subject.

Chain manufacturers, therefore, have developed special alloys of malleable cast iron, with case-hardened steel or bronze bushings and pins of hard non-corrosive alloys, sometimes heat-treated, thus ensuring a longer life to the chains of the intermediate carriers.

For overlapping steel slats, roller chains are used, having 2·97 in. pitch, the rollers having $1\frac{9}{16}$ in. dia. and 15,000 to 18,000 lbs. ultimate chain strength.

As the trash or bagasse may pack between the links of the roller chain, a *Double Sprocket Chain* for intermediate carriers as shown in *Fig.* 332 is now on the market, made by different manufacturers. The material is the same as used for the above-mentioned roller chain, and every effort has been made to obtain a long wearing capacity for this hard service. The ultimate strength of the double sprocket chain is about 25,000 lbs. and three strands should be used for 7 ft. wide carriers and heavy grinding. Since the engaging links are open on the outside, the bagasse cannot pack between them.

The weight of roller chain is about 6 lbs. per ft., whereas the double sprocket chain weighs 11½ lbs. per ft.

Even with overlapping slats it is not always possible to prevent the trash accumulating between the ascending and the returning aprons, and one slat therefore sometimes has a rectangular opening, covered by a hinged lid, which opens automatically on the return journey.

The intermediate carrier is driven by the mill it has to feed and a friction clutch is arranged in the driving mechanism to allow one to stop the carrier at will independently of the mill. There are several types of friction clutch in use for this operation, viz. :—

(*a*) Cone clutches.
(*b*) Multiple disc clutches.
(*c*) Expanding ring clutches.
(*d*) Coil clutches.

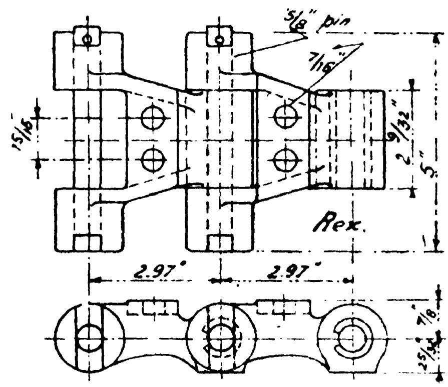


Fig. 332.—Double Sprocket Chain.

The *Cone Clutch* is of early design and although it is reliable, the axial forces for engaging are heavy, and for disengaging a considerable force is sometimes required as the cones may stick together.

A better clutch, therefore, is the *multiple disc clutch*, extensively used in automobile drives, composed of friction discs and arranged in such a way that the steel discs are provided with lugs, which enter into recesses or grooves on the steel hub pieces on the driven shaft, and the friction discs arranged between every two hub discs are similarly attached to the driving drum. The friction discs are made sometimes of "Ferodo" material, and it will be apparent that under slight compression the plates will pack together and develop heavy friction. As soon as the pressure is released the clutch is disengaged.

The *Expanding Ring Clutch* has a cast iron or cast steel ring rotating inside the driving drum. As the ring is split, the engaging device is designed in such a way that it will expand the ring and thus effect a firm grip on the peripheral surfaces of ring and drum. This clutch is made for operation in one direction of rotation and the force to engage it is small.

A design also used to advantage for intermediate carriers is the *Coil Clutch*, shown in *Fig.* 333, originally designed for steel mill work. The driving chain wheel *c* is keyed on a bronze bushed disc piece *b*, to which is attached the steel coil *e*, having an increased section towards *b*. This coil is bored out about 0·01 in. larger than the drum diameter *d* on which it rotates. The last turn of the coil has a bell crank lever f operated by the disc g. Thus, when the last turn is tightened on the drum, the whole coil will be tightened as a rope around a winch, according to the formula :—

$$F_c = P \times p \times e^{\mu\alpha} \text{ .... } (94)$$

where F_c = force acting on drum periphery in lbs.

P = Pressure with which the disc g is pressed against the bell crank lever f in lbs.

p = Bell crank ratio, normally = 5.

e = Base of natural logarithm = 2·718282.

μ = Frictional coefficient, normally = 0·12.

α = Coil winding angle, each loop = 2π.

$\mu\alpha$ being an exponential coefficient, the formula has to be computed with the use of logarithms.

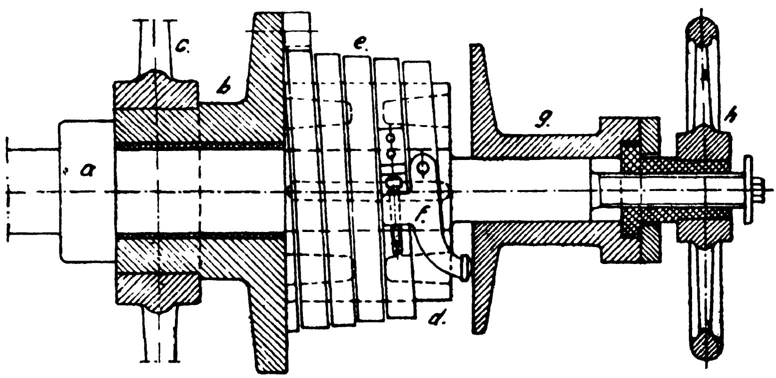


Fig. 333.—Coil Clutch.

The force F_c will amount to about 125 times the pressure on the disc g for the coil clutch shown, and the clutch is readily released when the handwheel and screw arrangement h is loosened. A very high gripping effect with little effort is thus achieved. The drum and the coil have to be lubricated for wear and the clutch must be operated in only one direction of rotation, that which tightens the coil.

The operation of the carrier clutches is generally done by a hand wheel, having spokes on the periphery, but some designers have the clutch worked from the mill platform by a chain drive. As the handwheel rotates when the clutch is engaged, there are now designs where an intermediate shaft is fitted, which can be rotated by hand either from the mill platform or the mill floor for engaging the pressure disc g of *Fig.* 333, without participating in the rotation of the carrier shaft. This design is quite practical. Incidentally, the collar a is to prevent the clutch sliding on the shaft.

To avoid the drawbacks of the moving parts of the intermediate carriers, a very ingenious design without moving parts has been developed in Hawaii, where the bagasse blanket moves along in a chute, without the assistance of an apron or drag conveying appliance. The receiving end of such an *Intermediate Sliding Carrier* is shown in *Fig.* 334. The mill discharges the bagasse between the scrapers m and n, and thus it will be pushed along the inclined chute *o*. The spring-loaded plate *p*, revolvably mounted on the scraper tip *m* maintains the blanket in a compressed state. The chute *o* is extended a little over midway between the two mills and the bagasse falls down a sloping chute towards the next mill.

No figures are available as to the power consumption of this sliding carrier as compared with the apron or drag type appliance, but the total absence of chains, slats, sprockets and the driving mechanism is a great advantage as it eliminates many sources of breakdowns. The author has seen this carrier in practical operation but the present inverted V shape of the chute has the disadvantage that the bagasse falls in heaps towards the next mill so that proper mill feed or good maceration is difficult to obtain, and this may be the reason why this unique design has not been used more extensively.

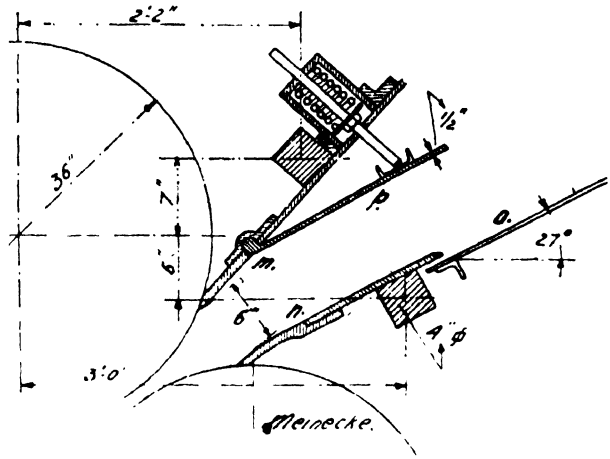


Fig. 334.—Intermediate Sliding Carrier.

Of the three types mentioned, the apron type thus holds the predominant position, as it gives an even feed, well adapted for applying maceration and pushes the bagasse gently into the next mill. Some special types of maceration carrier have been referred to in Chapter X.

2.—Bagasse Carriers.

The bagasse carriers serve the purpose of transporting the bagasse, as it emerges from the last mill, to the boilers and the bagasse store. An even feed is not required and a simple drag carrier design is quite suitable for the purpose. An endless chain moves in a trough (see *Fig.* 336) having walls about 12 in. high. The chain has attachments at regular intervals of about 30 in. on which are bolted wooden or steel slats about 6 in. high, vertical to the chute bottom, thus performing a scraping action on it. The slats are about $\frac{1}{8}$ in. clear of the chute bottom, so as not to offer additional friction. In Java the wooden slats are replaced by flat irons having prongs riveted to them to form a rake. The rakes are guided at both ends, but for fine bagasse the rake carrier is not as efficient as the scraper slat carrier.

The carrier chute is generally 48 in. wide or more, depending upon the size of the mill and the grinding capacity. The two chains, of malleable iron provided with rollers, have normally 6 in. pitch and about 34,000 lbs. ultimate strength. For very long carriers and heavy capacities, all-steel roller chains are used. The chain speed is kept at about 100 ft. per minute, thus about three times as fast as the average peripheral mill speed.

For the ascending part of the carrier, the single deck type is used, i.e., the return chain is not laid in a trough, but only supported on idler pulleys or guide rails.

Over the boilers, the double deck carrier is used in many instances to provide transport in both directions for the bagasse to and from the place of storage. The bagasse from the mills is brought to the upper deck and is discharged by one or more openings to the lower deck, where a slide is provided above each hopper or feeder on top of the bagasse furnaces.

In *Fig.* 335 is shown a *Bagasse Storage Carrier*, being the extension of the carrier over the boilers, and designed by the author for a 1500-ton cane sugar factory. About 750 tons of surplus bagasse can be stored on a floor space of about 2,600 sq. ft., but the piling is not done completely by the carrier for this capacity and manual labour is required to fill the space above the outlet of the discharging shoots *a*. The upper deck has a bottom height of 22 ft. 6 in. above column bases, whereas the lower deck is embedded in the concrete floor in such a way that it can be conveniently covered by wooden boards. These latter have to be removed at one end, when the surplus bagasse has to be returned to the boilers and the carrier is charged by manual labour.

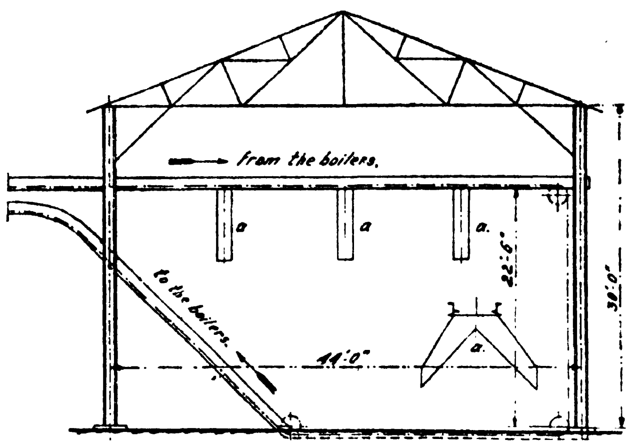


Fig. 335.—Bagasse Storage Carrier.

In the upper deck three double shoots *a* are provided, the first two fitted with slides, to allow them to be closed.

A *Carrier Slide* for a bagasse carrier over the boilers above the feed hoppers as designed by the author is shown in *Fig.* 336. An economic construction has been sought and therefore no use is made of the common rake and bevel gear construction.

The slide is 12 in. wide and covers the whole width of the carrier. The carrier chains, having 6 in. pitch and 34,000 lbs. ultimate strength, are supported on 4 in. × 3 in. angle irons riveted to the carrier walls, and provided with flat iron roller guides, 2 in. × $\frac{3}{8}$ in., attached by countersunk bolts and renewable when worn down.

The slide is composed of two parts, which open at the middle of the carrier width by means of a set of counter levers mounted on a shaft, which can be operated from the firing floor. The slides can be set at any required opening, the first ones, in the direction the bagasse flows, only partially opened, and the last ones completely so. The slides have riveted strips underneath, so that the bagasse cannot cover the slide guides. Three-inch angles and flat iron bars, 2 in. × ½ in., compose the slide-operating mechanism.

This carrier has a length of about 150 ft. and is driven by a 25 h.p. motor, although normal power consumption amounts only to about 15 h.p.

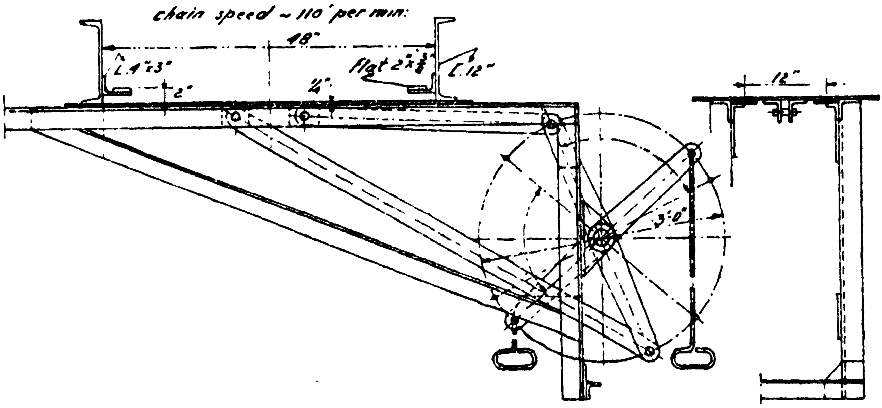


Fig. 336.—Carrier Slide.

For heavy capacities and long carriers, all-steel roller chains are used, generally having 75,000 to 80,000 lbs. ultimate strength. Such a chain as supplied by the author is shown in *Fig.* 337. Each fifth link has an attachment *c*, being a hinged malleable fitting, to which the 6 in. wooden scraper slats are attached. The carrier has a length of 258 ft. and has to transport

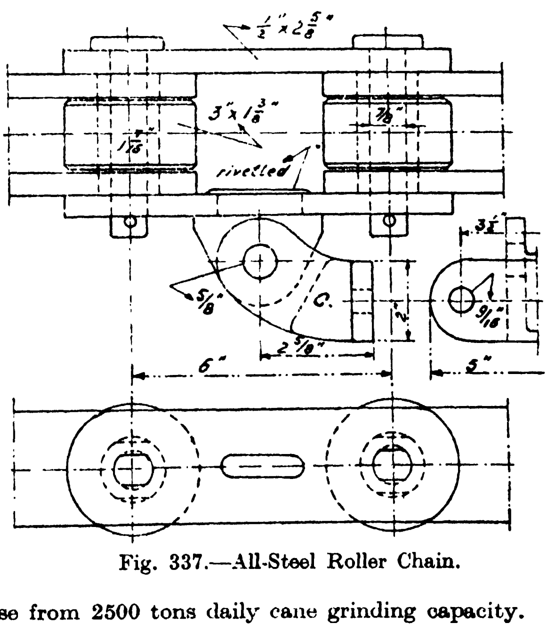


Fig. 337.—All-Steel Roller Chain.

all the bagasse from 2500 tons daily cane grinding capacity. The rollers and bushings are similarly made as shown in *Fig.* 120. The pins are constructed of nickel steel and drilled with a central oil hole, but the oil chamber construction shown in *Fig.* 122 is to be preferred.

3.—Bagasse Presses.

Where bagasse has to be shipped, stored or be used as fuel for locomotives, it should be baled, and this applies in particular to those countries where dry cane leaves are used as fuel for the boilers. These leaves easily catch fire, but when baled with green bagasse this danger is reduced and the fuel value of the bagasse increased.

The equipment used for baling is called a *Bagasse Baling Press,* although it could equally well be called a bagasse stamper. It is used in several cane-growing countries and the essentials of the different designs are embodied in *Fig.* 338. The baler can be erected in a horizontal position as drawn or on a slanting foundation, having an inclination of 45° with the horizontal and the discharge end at the bottom.

The driving pulleys revolve at about 400 r.p.m., which speed is reduced through a double gearing to about 25 r.p.m. on the main crank shaft. The crank action is generally achieved by means of two gear wheels, between which the connecting rod is attached to the crank pin. As the compression of the bagasse only takes place at the end of the outgoing stroke, it will be obvious that the gear teeth will not experience equal wear and therefore the crankpin can be changed at three or six different positions on the crank wheels, so as to distribute the wear on the teeth.

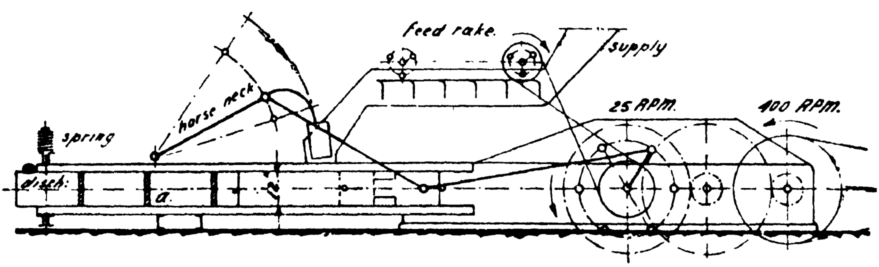
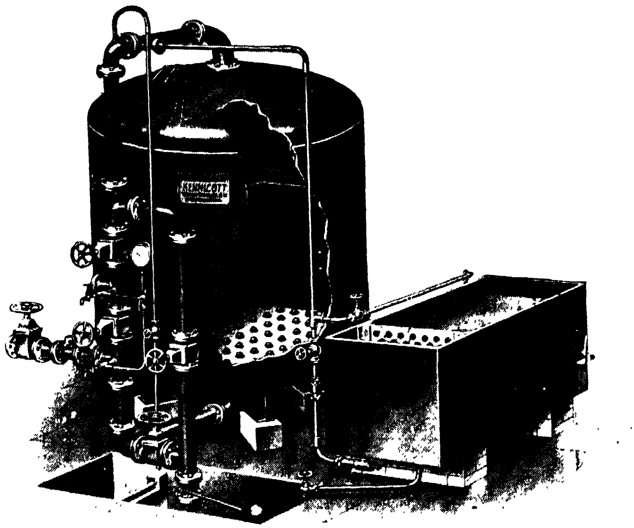


Fig. 338.—Bagasse Baling Press.

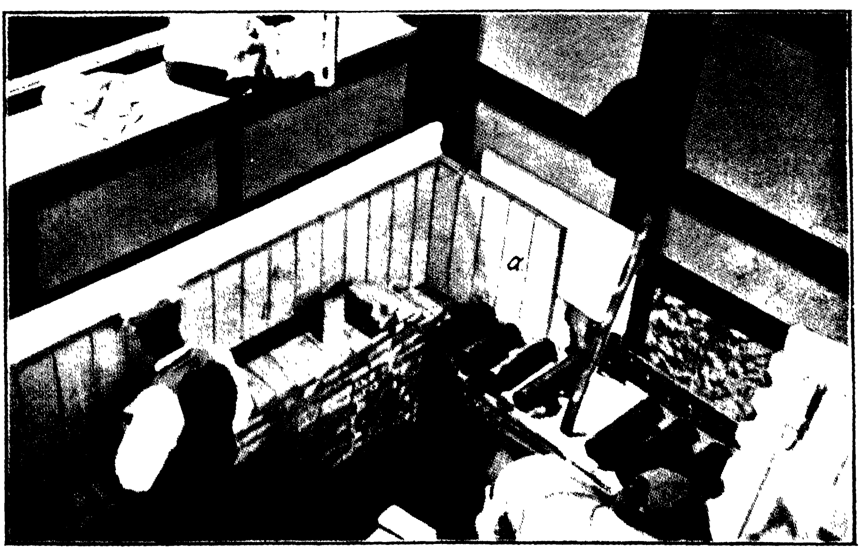
The crosshead is mounted on rollers, to avoid increased friction by trash falling on the guides, and is connected to a stamper, which compresses the fed bagasse. For even feeding a feed rake is provided in the supply trough of the bagasse, mounted on two shafts, one of which is operated from the crank shaft by belt drive. Each of these two shafts has a three-throw crank on which are attached the three sets of rakes. With the return stroke, the rakes run clear of the bagasse.

Driven from the crosshead, the horseneck (of a type extensively used in agricultural trashing machines) pushes the bagasse down into the press chamber, before it is compressed by the stamper. The compressed bagasse is pushed forward between heavy guides, which are kept tightened by a spring arrangement at the discharge end. Heavier spring pressure will cause increased compression of the bagasse. Between the layers of bagasse wooden boards or steel plates are inserted at the end of the stamper stroke, so as to give the required separation. These boards have two grooves in which galvanized iron wires (No. 10 B.W.G., 0·128 in. dia.) can be inserted for tightening the bales.

The shafts can be mounted in ball or roller bearings and the capacity reaches six to nine tons compressed bagasse per hour. Green as well as sun-dried bagasse can be used, although bagasse baled moist may start spontaneous

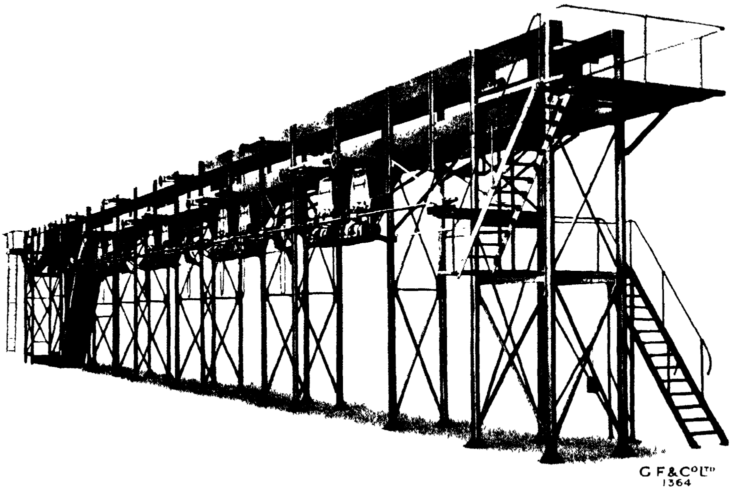


KENNICOTT KENZELITE BASE EXCHANGE WATER SOFTENER.
(*Kennicott Water Softeners, Ltd.*)

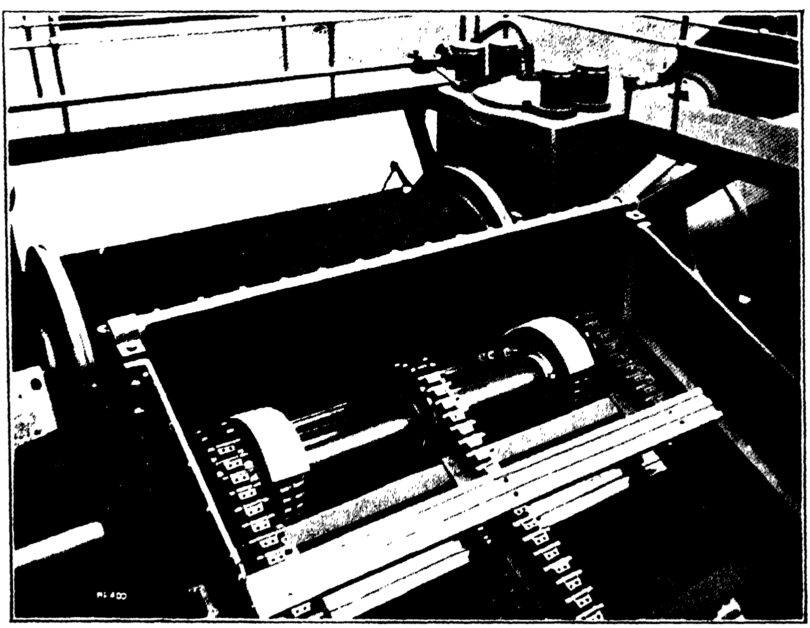


SUPEREX AND MAGNESIA COMBINATION INSULATION (*a*) BEING APPLIED TO FURNACE WALLS.
(*Johns-Manville Int. Corp.*)

PLATES 58 & 59.



Return Bagasse Carrier for Eight Furnaces.
(*Geo. Fletcher & Co., Ltd.*)



Driving End of Intermediate Apron Carrier.
(*Duncan Stewart & Co., Ltd.*)

combustion. The author has seen bales which were fouled inside and a loss of 10 to 15 per cent. was caused thereby.

The power consumption varies according to the size of the bales and the compression achieved, viz. :—

100 bales	of 50 lbs.	each per hour	......	10 h.p.
75 „	225 lbs.	„ „	......	25 h.p.
50 „	130 lbs.	„ „	......	15 h.p.

Overloading will tend to break the press and the crank drive is generally the weakest part of the system.

The size and the weight of the bales also varies, and the moisture content of the bagasse will greatly increase the weight. Recorded average sizes are :—

12 in. × 12 in. × 24 in.	From 50 to 90 lbs.
18 in. × 22 in. × 30 in.	225 lbs.

The specific weight of cane leaves, which have to be shredded before they can be baled, is about 20 lbs. per cub. ft. when compressed. Compressed bagasse will vary between 30 and 45 lbs. per cub. ft.

When green bagasse is used, the bales should be piled in checker board fashion, so that good ventilation will dry them to about 20 per cent. moisture and the bales will lose about 45 per cent. on weight. It is important to ship bales dry, when freight has to be paid for by weight.

The author has watched the operation of bagasse-baling presses and concludes that only one man is required for the main task, although it is convenient to have a man on each side for the wire binding. The transportation of the bales and the piling may require from two to four men.

Baled bagasse is burnt on locomotives, but as bagasse, especially when green, must be deemed low grade and voluminous fuel, the grate area for a bagasse-burning locomotive should be about 15 per cent. larger than for a coal-fired one.

In Hawaii it was found that bagasse-fired locomotives could only be used on level tracks, not on gradients.

A more compressed bagasse fuel will therefore have advantages for locomotive work and the *bagasse briquetting press* has been designed for this purpose. As these machines are expensive and the excess bagasse of a sugar factory generally remains within narrow limits, a co-operative scheme of several neighbouring factories should be organized for economic production. The excess bagasse should be transported sun-dried and baled to the briquetting press.

Green bagasse cannot be used for briquetting and the Java Experimental Station, as well as Queensland, have supplied us with information in regard to this matter.[1] The optimum moisture content is sun-dry with about 7 to 8 per cent. water. For sun drying, a floor space of 6000 sq. ft. is sufficient for drying up to 2·5 tons of bagasse per day.

A mixture with molasses in the proportion of one part molasses to four parts bagasse will give a good binding material and better briquettes, which will not disintegrate when piled.

The necessary pressure is at or above 15,000 lbs./sq. in. approx. and a heavy power consumption is thus required.

1 See *Het Archief*, 1933, part III, pp. 1–28.

In *Fig.* 339 such a *Briquetting Press* is shown, driven by an electric motor and requiring 70 h.p. for normal operation. The press has to be started at a speed of about 25 r.p.m. and the electric motor has to have a starting speed corresponding to this number of revolutions, which must be gradually brought up to 70 r.p.m. of the crank shaft. Two flywheels supply the necessary inertia for the compressing performance.

In some instances the briquetting press is directly driven by a steam engine with oppositely arranged cylinders ; this is also a useful arrangement for the varying speeds required.

The bagasse is fed to the feeding hopper *b*, whereas at *a* a worm-driven spindle allows for adjustment of the compresssion moulds. These moulds can be cooled, when excess heat is developed.

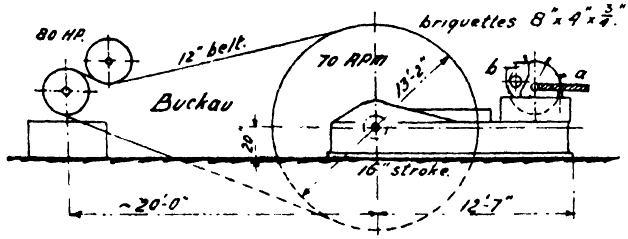


Fig. 339.—Bagasse Briquetting Press.

The size of the briquettes is generally 3½ in. × 9 in. and 4 in. × 8 in., the thickness being about ¾ in. The specific weight when piled is from 40 to 60 lbs. per cub. ft., which is well in excess of the baled bagasse and thus less space is required and the grate area also can be of smaller proportions. When fired the briquettes will keep their compressed form until almost completely burnt. The stamp and moulds are casé-hardened and subject to wear, so spares have to be kept at the factory. The normal capacity of such a press is from 5 to 10 tons per hour.

Though the cane is weighed, as well as the extracted juice, the weighing of the bagasse itself has not received special attention even in those factories where the imbibition water is not measured or weighed. *Weighing* has been attempted by intermittent weighing machines, but favourable reports are not available so far. For complete control, weighing is nevertheless desirable and a continuous weigher in combination with a belt conveyor is now on the market.[1] Operating data are not yet published, but the development should be watched with interest.

[1] See *I.S.J.*, 1932, p. 318.

CHAPTER XV.

PIPING.

STEAM, JUICE, WATER AND VAPOUR PIPES AND FITTINGS.

The pipe lines in a sugar factory do not always receive the attention they need and any improper lay-out of pipe lines will not only reduce the efficiency of the factory by undue friction, but may become dangerous where hot liquids or steam are conveyed ; and, besides, unnecessary expense may be involved.

The author knows instances where the boiler plant had satisfied all official and insurance requirements, whereas the steam lines were of "home-made" construction, and comprised a web-like arrangement not in keeping with the steam generators and steam consumers they had to connect. In most installations it will pay to have the pipe lines laid out by a competent engineer, and made by manufacturers who specialize in this kind of work, as the author could prove from many instances in his experience.

Pipe lines should in any event be laid as straight and short as possible with long radius bends, so that pipe line hammer and vibration may be avoided and efficiency and safety during the operation of the factory be increased. Steam lines and hot liquid lines should be tested by hydrostatic pressure of twice the working pressure. Cold liquid lines will be effective when tested at 1½ times the working pressure.

1.—Construction of Pipes and Fittings.

For cane sugar factories nearly all types of material are used for piping—cast iron, steel, wrought iron, copper and brass, while special alloys like molybdene steel, etc., have also been employed. Underground service requires non-corrosive material, and concrete and asbestos-cement pipes are used for sewers, while asphalted cast iron or glazed ceramic piping is used for sanitary purposes.

Water lines laid underground are generally of cast iron for pressures up to 500 lbs./sq. in. Faucet and spigot joints are used for this kind of service, as flanged pipes will prove objectionable in many cases since they are exposed to corrosion of the bolts, due to soil humidity or chemical action. Galvanized iron pipes are sometimes used for small water supplies.

Pressure and high temperature lines for steam or hot liquids should be made of steel or wrought iron, the latter having a better resistance against corrosion and thus giving a longer life.

Cast iron is used for acid juice lines, CO_2 and SO_2 lines in carbonatation or sulphitation houses, injection water lines and vapour lines. Attempts have been made to make the latter from aluminium, but this is not yet established practice.

Brass pipes are used for small size piping, where corrosion or adherence of gummy matter is to be feared. Copper is used for the heating coils on account of its excellent heat transmission, but as the material is soft, copper pipelines are easily buckled. Due to its ductibility copper can be worked into nearly any shape.

Pipes and fittings are connected either by flanges or screwed connexions, and as there are no universal standards, the flange or thread standard required should be mentioned when ordering. Flanged connexions can be easily taken apart.

In *Fig.* 340 is shown an *Expansion Flange* of outstanding design. The steel pipe (seamless steel pipes are made up to 11 in. dia.) is enlarged by a tube expander within the taper bore of the flange, so the material will be pressed into the two grooves. The flanges are made of forged steel and the flange connexion generally is stronger than the tube proper.

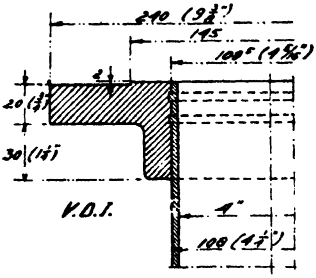


Fig. 340.—Expansion Flange.

These flanges are used in pipes from ⅜ in. up to 16 in. diameter. For heavy pressures spigot and faucet flanges are used, so that the packing will not be blown out; but they have the disadvantage that intermediate parts cannot be taken out so easily. The flange and pipe materials are firmly pressed together, so that no steam or liquid will enter or exude between them at the face of the joint.

Screwed Flanges as shown in *Fig.* 341 are extensively used in American practice. For sugar-house work cast iron flanges are mostly used, whereas for higher pressures and temperatures, malleable iron or forged steel is applied. Screwed flanges are available from 2 in. to 24 in. normal pipe diameter, and the cast iron ones are for 125 lbs. steam pressure up to 12 in. and for 100 lbs. beyond that limit.

As the pipe material is undercut by the thread, a heavier wall thickness is required. Moreover, the thread is tapered, and leakage between the pipe and the flange sometimes occurs at *b*. Through dangerous V-notch stresses, the screwed flanges may develop a break at *a*, especially when this part gets corroded.

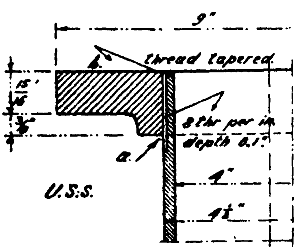


Fig. 341.—Threaded Flange.

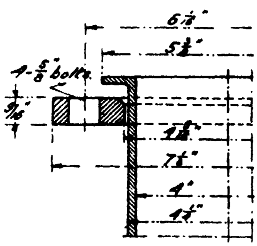


Fig. 342.—Lap Flange.

A *Lap Flange* is shown in *Fig.* 342, this connexion being used for pipes from 2 in. to 24 in. bore, especially for the larger sizes. The lap has to be done in a smithy fire when made at the sugar factory. Copper piping is well fitted for this kind of connexion. As the flange does not need a hub, small flange diameters result.

The *Welded Flange* is coming into favour, as many sugar factories will have a welding outfit. In *Fig.* 343 is shown such a flange, having a chamfered pipe end set at 45° so as to obtain a larger adhesion surface for the welding material. Welded piping requires only flanges in those cases where the pipe-line has to be taken apart. The flanges shown are put on the market by manufacturers with from 2 in. to 18 in. normal pipe diameter.

For heavy pressures the double-welded lap flange is used, ranging from 6 in. to 24 in. bore; the pipe end nearest the flange being widened and the pipe taken in, so that there will be an inside and an outside weld. It is obvious that with this construction the inside surface of the pipe will not remain flush.

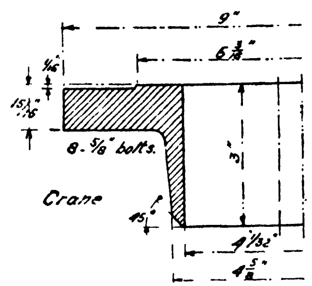


Fig. 343.—Welded Flange.

Another method is to employ slip-over welded flanges which also have a double weld and a smooth inside surface. For copper tubes these slip-over flanges are sealed with hard solder.

Riveted pipes in the form of straight or spirally riveted tubes are still used in many sugar factories, but manufacturers of sheet and plate work will now supply welded pipes up to any desired diameter.

Riveted flanges are used on large diameter pipes, and the flanges are generally made of heavy angle iron. But a better construction is to use cast steel riveted flanges, which have more resistance against buckling and will make a better joint, a matter of importance for vapour lines under vacuum. These cast steel flanges can also be welded.

A cast iron *Faucet and Spigot Joint* is shown in *Fig.* 344; the annular space is packed with hemp string and then filled with molten lead, which has to be caulked. There are now paste-like materials on the market for these joints, which do not require caulking. The construction of *Fig.* 344 will stand 250 lbs. non-chock water pressure and the net pipe length is 16 ft. The tubes are made according to the modern centrifugal casting method, where the moulds are rotated at high speed, when the metal is poured in and thus the slag will float on the inside and a sound homogenous casting is obtained. The tubes are coated in an asphalt bath before shipment. Cooling should be done gradually after casting, so as not to cause heat stresses.

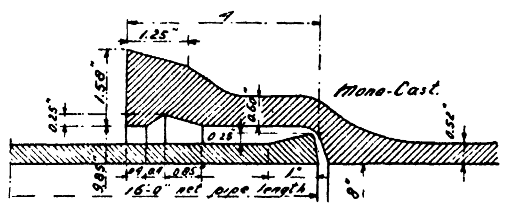


Fig. 344.—Faucet and Spigot Joint.

Branch connexions can be made by split saddle pieces, bolted around the tube, or by tees, crosses or branches, having faucet and spigot joints.

2.—Pipe Data and Miscellanea.

The pipe lines should give a steady flow of the steam or liquid they convey and abrupt changes in the direction of travel should be avoided wherever possible.

As the velocity of the flow has a direct bearing on the forces of inertia caused by it, this velocity should be kept within reasonable and safe limits. Moreover, the velocity of flow affects the pipe friction, which may destroy mechanical power or require this to overcome the friction.

Pipe friction depends not only on the velocity of flow, but on other factors, such as abrupt changes, smoothness of the inside of the pipe, adherence of scale or slimy matter, and viscosity of the conveyed material, which factors have been partially ascertained by practical tests, but many are not fully known in respect to the pipe-conveyed materials present in cane sugar factories.

From many actual installations, the author has compiled the following average *Velocities of Flow*, to form a good basis for well-designed pipe lines, and the more obstructions there exist, the lower the velocities of flow should be. For future extensions it should be unnecessary to mention that maximum velocities should not be used in a newly equipped factory.

Vapours below atmospheric pressure	120 to 200 ft./sec.
Saturated live steam	80 to 120 ft./sec.
Superheated live steam	130 to 250 ft./sec.
Exhaust steam	100 to 150 ft./sec.
Vacuum air	40 to 60 ft./sec.
Compressed air	60 to 100 ft./sec.
SO_2	60 to 100 ft./sec.
CO_2	40 to 100 ft./sec.
Water and thin-juice suction	3 to 4 ft./sec.
,, ,, ,, discharge	4 to 8 ft./sec.
Low pressure centrifugal pumps	6 to 10 ft./sec.
Syrup	2 to 4 ft./sec.
Massecuite	1 to 2 ft./sec.
Fuel oil	1 to 2 ft./sec.

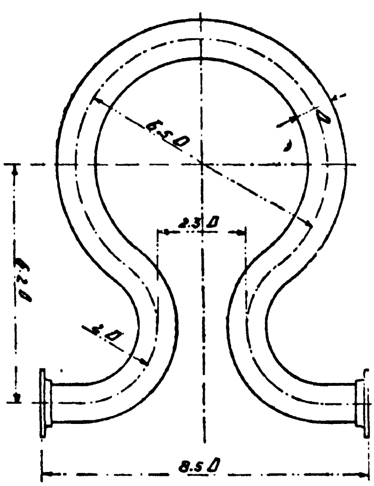


Fig. 345.—Expansion Bend.

The different specific volumes can be found from engineering handbooks, so far as not mentioned in this book.

As pipe lines at temperatures higher than the surrounding air will expand, provision has to be made for this and the *Expansion Bend* as shown in *Fig.* 345 is frequently used. These bends are also made of corrugated piping, which has less rigidity than the flush type.

These expansion bends are used in the main steam line and the boilers should be connected to this line by inverted U bends, so that a swivelling movement will be possible under expansion.

There are also expansion joints of the *Sleeve Type* with a stuffing box. These have to be secured by bolts, as the pressure inside the pipe line may throw the component parts asunder. Balanced expansion joints with sleeves are now on the market, but they are expensive.

The *Bellows Type* is also used, composed of two discs, hammered to a spherical shape of about twice the pipe diameter and riveted together by a hoop on the outside periphery. These joints can be made at the factory, but they should not be used for high steam pressures, as the force resulting from the steam pressure on the larger diameter will be considerable. A set of swivel joints can also be used for expansion purposes.

It should be recollected that all unbalanced expansion joints have to withstand an axial pressure equal to the product of steam pressure and pipe area and, therefore, the ends of a steam line have to be well anchored, as otherwise the expansion will not be taken up by the expansion joint. The intermediate sections, therefore, are mounted on roller supports, so as to allow for this expansion.

The degree of expansion depends on the temperature difference and the pipe material; the pipe thickness and diameter have no influence on it. An *Expansion Diagram* is shown in *Fig.* 346 for different pipe materials and temperature differences. As the expansion is practically proportionate to the rise in temperature, straight lines are to be found in the diagram. The increase in pipe length is given in inches per 100 ft. pipe length and the temperature rise in °F.

Flanged pipes have *packing gaskets* between the pipe joints, which are of asbestos material or metal for steam lines and rubber with canvas for water and juice lines of not too high a temperature. Manufacturers now put very excellent materials on the market for this purpose. On taking a pipe line apart the gaskets are generally destroyed, as the packing material will have adhered to the flange metal. The flange surface, therefore, should not be very rough and should be covered with tallow and graphite before the packing gasket is inserted. This will ensure that the packing does not stick to the flange surfaces. Screwed fittings are put together with white lead or manufactured pastes.

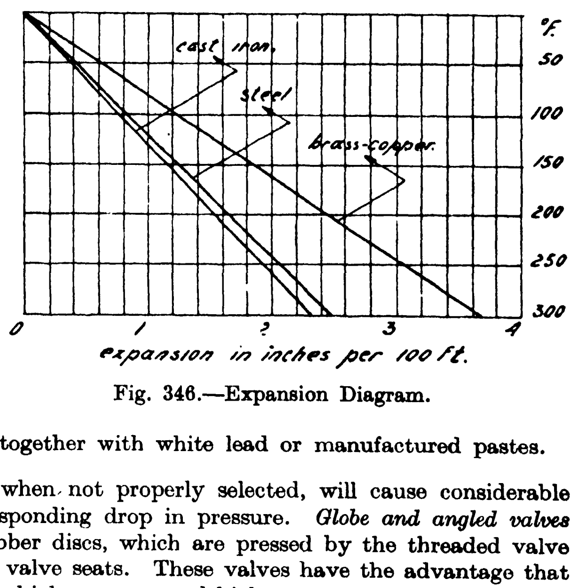


Fig. 346.—Expansion Diagram.

Valves and fittings, when not properly selected, will cause considerable pipe friction and a corresponding drop in pressure. *Globe and angled valves* have metallic or hard rubber discs, which are pressed by the threaded valve spindle on the removable valve seats. These valves have the advantage that they will keep tight under high pressure and high temperature. Cast steel or alloy steel is to be preferred for pressures over 100 to 125 lbs. and for superheated steam. The valve seats are made of brass for saturated steam and of nickel alloy for superheated steam. Impurities carried along with the steam will have a grinding effect on the valve seats. Leaky valves should be re-ground when metallic discs are employed.

The *Gate or Full Way Valve* does not necessitate a change in the direction of flow as does the globe valve, and thus gives less resistance, which is of interest for high rates of flow. These valves do not always keep tight for high pressures, when the single wedge type is used, but for water, juice, exhaust and vapour lines they are to be preferred. The wedge type gate valve may stick under high temperatures, and for this latter purpose double parallel disc valves are now made to overcome this inconvenience.

For larger size valves, above 16 in. and pressures above 20 lbs. per sq. in. or vacuum, a pressure equalizing pipe of 1 in. to 2 in. in dia. should be embodied in the valve construction, as otherwise the opening and closing of the valve will be difficult and heavy wear on the valve seat or wedge will result.

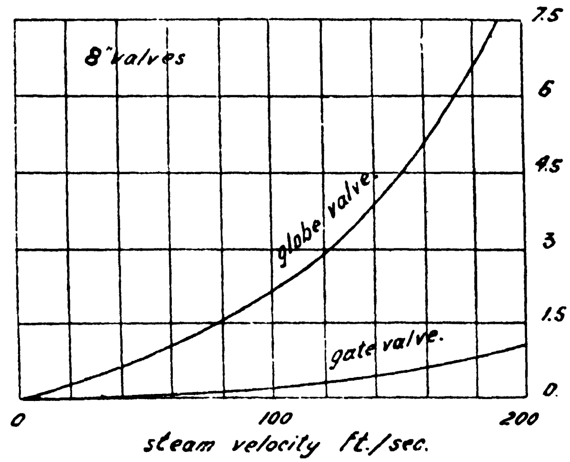


Fig. 347.—Pressure Drop Diagram.

Cocks are made of brass or cast iron for the larger sizes. The lubricated type is to be preferred for sizes over 3 in. For high temperatures cocks are not feasible, as they will easily stick. A cock will give a quick opening, but the chock effect may be considerable, so it should not be used on main lines but only on branches.

With all valves and cocks it is convenient to be able to ascertain readily whether they are open or closed. Valves with outside thread or a long spindle construction are to be preferred for all principal valves. Cocks are marked with a groove on the top of the plug for this purpose. A disc indicator with pointer is often fitted, marked "open" "closed."

In *Fig.* 347 is shown a *Pressure Drop Diagram* for two 8 in. valves, one being a globular and the other a full-way gate valve; and it will be seen that the gate valve is superior for high velocities of flow. Similar curves could be made for short elbows and tees and it indicates clearly that these accessories should not be used for velocities of the conveyed matter inside the pipe line beyond 60 ft. per second when steam or vapours are concerned, and not over 6 ft./sec. when referred to water or juice. Moreover, many a vibrating pipe line is due to abrupt changes of flow at too high a velocity.

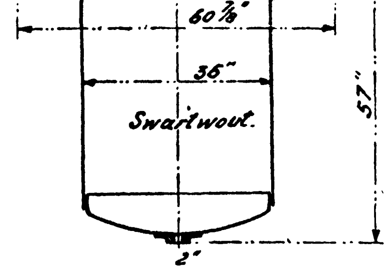


Fig. 348.—Steam Separator.

All steam lines will experience condensation from heat losses and should therefore be drained at all convenient places. In *Fig.* 348 is shown a *Steam Separator* destined for this purpose; the steam flow is twisted by an ingenious impeller *x*, which will throw the condensed water towards the walls of the vessel, and the discharge is fitted with a short internal pipe to draw off steam from the dry zone. It is obvious that the rate of flow will be considerably reduced inside the vessel. The drain is connected to a steam trap, or a water level gauge may be attached to the vessel, so that it can be drained by a valve, when water has accumulated.

For air lines to the vacuum pumps it is also convenient to have a dirt collector of simple design, as rust from the inside walls of the pipe lines, which have long vertical stretches going to the condensers, will do damage to the pump valves or piston courses, when it gets loose.

Insulation should be applied to all steam lines, but of course vapour lines going to the condensers need not be insulated. Hot liquid lines should also be covered, so as to protect them against heat losses.

Present-day insulating materials generally contain 85 per cent. magnesia and should have good adhering qualities. For steam lines canvas strips are wound around the insulation to prevent it falling off. Large bodies are sometimes covered with small mesh wire gauze before the insulating material is

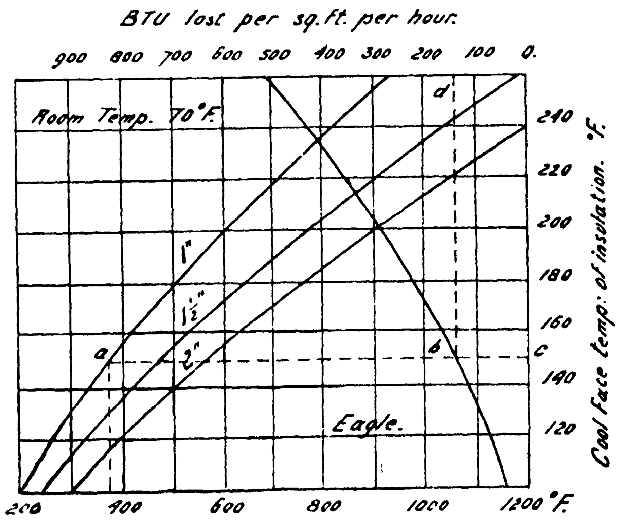


Fig. 349.—Heat Loss Diagram.

applied, so as to give proper adherence. The thickness of good insulating material does not need to be over 1 to 2 in., so should not prove an expensive item. In several tropical countries rotan-straw is used as insulating material, being cheap, and is neatly applied by native labourers.

Uncovered steam lines may lose as much as 10 per cent. of the total heat contained in the steam, and as steam generally has a much higher heat content and temperature, besides higher velocities of flow, heat radiation losses will here be more considerable than with hot liquids. This applies as well for exhaust steam and heating vapours.

A *Heat Loss Diagram* is shown in *Fig.* 349 where the losses are within reasonable limits under proper insulation. At 125 lbs. steam pressure, saturated steam has a temperature of 353°F. By applying this temperature at the bottom line of the diagram, a vertical is drawn until *a*, where it intersects the 1 in. insulation curve. The outside temperature of the cool face of the insulation will be found, by drawing the horizontal *a* - *c*, at about 150°F.; and by drawing the vertical $b - d$ at the intersection of the heat loss curve, a loss of approximately 140 B.Th.U. per sq. ft. exposed surface per hour will be the result—a reasonable figure.

Finally, in *Fig.* 350 is shown a *Cast Iron Vapour Line* as designed and supplied by the author for an existing sugar factory of 1500 tons maximum grinding capacity, connecting the four vacuum pans to the central condenser. The flow line principle is adhered to and the good results achieved can be easily ascertained by placing a vacuum gauge on the condenser and comparing the readings with the vacuum gauges on the pans.

Cast iron vapour lines are very resistant to corrosion, but their heavy weight requires specially strong roof trusses or the use of special pipe supports from the platforms.

As erection at a considerable height is not always an easy matter, steel or wrought iron plate, having a reduced weight, has been used also for this kind of pipe work. These steel pipe lines after each crop have to be painted with a good kind of non-corrosive paint on the inside, after having been well scraped.

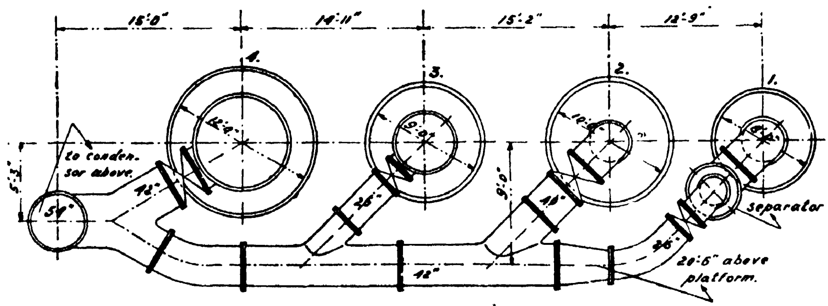


Fig. 350.—Cast Iron Vapour Line.

The calculation of the thickness of these vapour ducts is still an empirical one, as a principal factor, the true cylindrical shape, is not always adhered to in common practice ; $\frac{5}{16}$ in. to $\frac{3}{8}$ in., therefore, is accepted as a safe thickness, but it is as important as well to have reinforcing hoops of angle or T section at intervals of about $1\frac{1}{2}$ times the diameter, especially when the vapour pipes are laid horizontally and thus the bridge stresses might cause flattening of the tube section.

The author has seen thin steel tubes collapse under vacuum, thus a saving in material merely resulted in heavy expense from interruption of the grinding season and a costly replacement and repair.

The flanges of these vapour ducts under vacuum are preferably made of cast steel, riveted or welded to the pipe material and drilled for $\frac{3}{4}$ in. bolts at about 4 in. pitch. Flanges of angle iron should have a bolt pitch of about 3 in.

The packing of the joints consists of a mixture of red lead with linseed oil, hammered or ground to a thick paste, so that it will stick to the flange when applied for vertical joints. Asbestos cord is laid on the inside and outside of the bolts, to prevent the red lead being drawn in. The packing material pressed out should be removed from the inside as well as from the outside. When the joints are hardened, the vapour duct should be put under vacuum and tested at the joints by means of a candle. The flame will draw inwards, wherever there is a small leak. Cast iron vapour ducts should be painted with a thick body paint, so as to fill eventual pores.

The vapour valves of large diameter are built as gate or full-way valves, or sometimes they are constructed as angle valves, having a hard rubber seal ring in the valve disc.

An interesting recent design of a *Straight-way Toggle Vapour Valve* of British design is shown in *Fig.* 351. The opened valve does not offer any obstruction to the vapour flow and a good sealing performance is obtained by the toggle levers. The disc has a hard rubber seal ring, and when closed

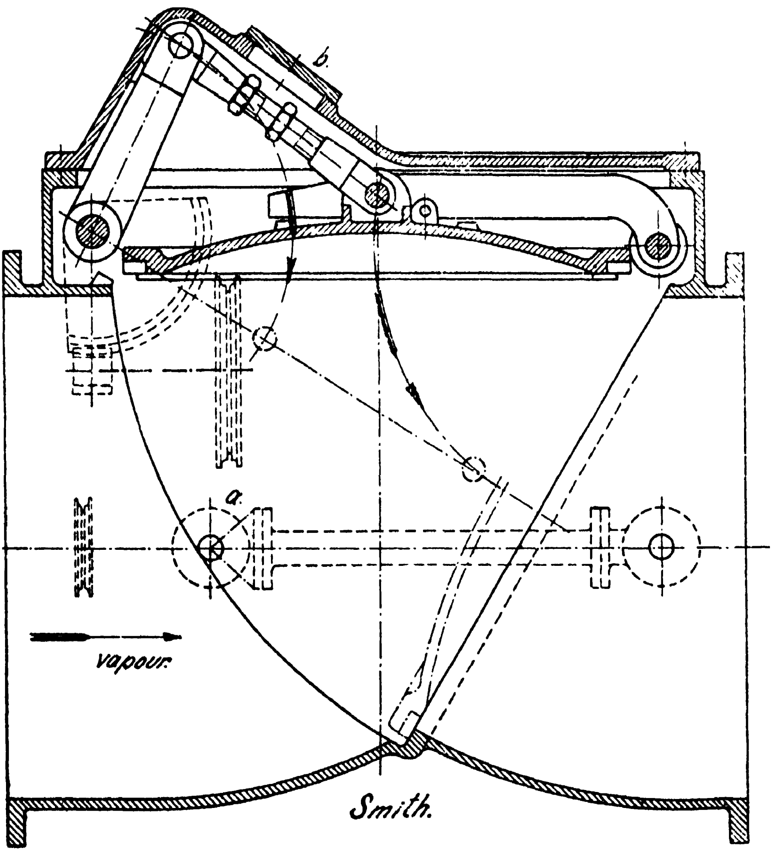


Fig. 351.—Straight-way Toggle Vapour Valve.

the pressure is acting on top of the disc, so will assist in the closing performance. Before attempting to open the valve, the pressure equalizing valve *a* has to be opened, to obtain nearly equal pressure on both sides of the disc.

The toggle lever can be easily adjusted by removing the cover of the hand-hole *b*, and it will be noted that the valve occupies a very reduced space and is of less weight than angle or gate valves of the same diameter.

CHAPTER XVI.

JUICE MEASURING AND WEIGHING EQUIPMENT.

VOLUMETRIC MEASUREMENT — INTERMITTENT JUICE WEIGHING SCALES — AUTOMATIC JUICE WEIGHING SCALES.

For the proper control of the sugar manufacturing process, accurate account has to be kept of the amount of sucrose entering the factory. The volume or weight of the juice is thus measured and the sucrose contents determined in the laboratory.

Measurement by volume will be less exact than by weight, as the volume is affected by the presence of scums and air or frothing through slight fermentation of the components of the juice. Moreover, the tank material will expand at a rate different to that of the liquid contained therein, which fact will make measuring less accurate for liquid temperatures above the surrounding air temperature.

Dirt, which is present in all mill juices, affects both the volumetric and the weighing method, and good sampling of the juice for proper sucrose determination is equally essential for both.

The measuring and weighing apparatus can be used also for other liquids than the juice itself; i.e., for imbibition water and diluted filter-press mud and, given special care to the tare-weighing, for measuring semi-liquids like molasses.

Measuring by venturi-meters is not sufficiently accurate for factory control data, although it has the advantage of continuous measuring.

1.—Volumetric Measurement.

Volumetric measuring is done in cylindrical tanks about 4 to 5 ft. in diameter and 6 to 8 ft. high, the average being between 1000 and 1500 gals. capacity. Three tanks are required for every installation, one for filling, one for measuring and the third for discharging and eventual cleaning. For large size factories, a double set of tanks should be available, two being filled at one time; this to avoid making these tanks too large, since a large diameter will affect the accuracy of the measurement.

The formation of scum or froth is due to the aeration of the juice when entering the tank; this will cause the juice level to drop when it has been standing for a certain time. Fermentation can be stopped by a reagent like formaldehyde and thus the correction (which may be up to about 3 per cent. of the tank volume) can be fixed.

To avoid aeration, it is advisable to run the charging pipes down close to the tank bottom and so reduce the tendency to scum formation or foaming. A small make-up or trickle feed juice line of ¾ to 1 in. is convenient so as to fill the tank up to the overflow level before discharging.

All measuring tanks should discharge freely into a receiving tank placed underneath. A single discharge pipe line connected to all the measuring tanks by branches is not advisable, as a leaky discharge plug or valve may then remain unobserved.

The *Overflow* generally is of the V-notch type as shown in *Fig*. 352, and the free section of the overflow discharge always has to be larger than the charging pipe area. The notch plate should be made of $\frac{1}{8}$ in. copper plate, as slimy matter will not adhere to it and thus the measuring is not impaired. The brass spout, bolted to the outside of the tank, should discharge into a gutter, which returns the overflowing juice to the unmeasured juice tank.

The discharge valves or plugs are sometimes made completely of brass, but a hard rubber joint will prove preferable for resisting grit and may keep tight for a longer time.

Measuring and weighing tanks should be flush inside and welded tanks are now used to advantage. The bottoms should be of conical section, having an inclination of about 45° with the horizontal, so that complete and quick discharge is ensured.

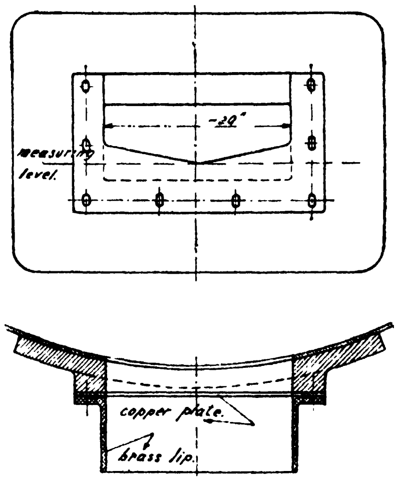


Fig. 352.—V-notch Overflow.

Quick opening gate valves are also convenient for the discharging as well as for the charging performance, and in case the capacity of the measuring tanks is small, the filling cycle can be speeded up by the installation of a supply tank with a large charging pipe to the measuring tanks.

Before the commencement of the crop or in case of doubt the measuring tanks should be rated with water of the same temperature as the juice to be measured. The water is weighed in advance on a good weighing platform.

To keep count of the number of measuring tanks filled, a counting board with wooden pins is generally used. Sometimes a recording instrument, connected to a float in the tanks, will show the number of cycles of each tank, and this should be a check on the counting board.

The tanks should be cleaned at regular intervals with hot water by means of a hose and nozzle or by a perforated ring-shaped pipe, attached to the inside top of the tank.

2.—Intermittent Juice Weighing Scales.

While the measuring tanks require a correction coefficient of up to 3 per cent., the juice weighing scales will give the true weight with an allowance of only from 0·1 to 0·25 per cent. and 20 to 30 weighings per hour can be achieved with non-automatic scales.

Some processes require the weighing of the hot juice, but it has been found that there is no practical difference with either hot or cold method regarding the accuracy of the results obtained.

An arrangement of twin *Juice Weighing Scales*, as widely used in tropical America, is shown in *Fig*. 353. Each tank is filled and weighed separately and the only precaution to be taken with this design is that the discharge plug or valve should be of large size, to allow one tank to be quickly emptied before the other is filled.

On the weighing beam a registering apparatus *d* is attached, in which a ticket can be inserted on which the weight is stamped by depressing a small lever. The net weight is the one weighed and the balance of the tare weight

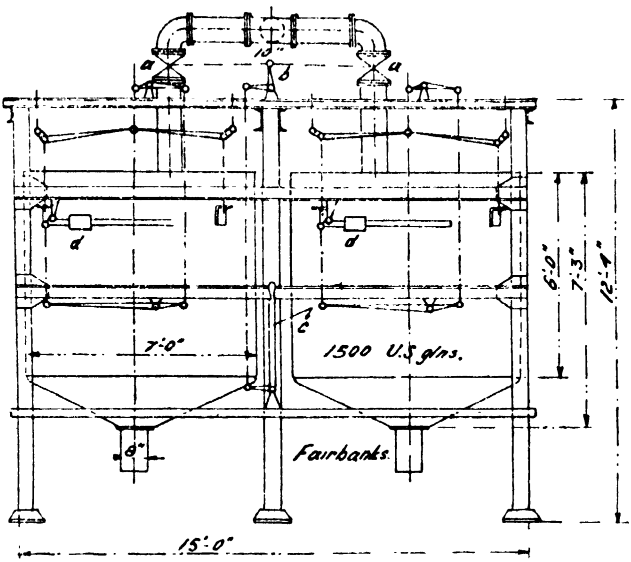


Fig. 353.—Arrangement of Twin Juice Weighing Scales.

should be checked at regular intervals. Any slime or dirt adhering to the tank walls has to be removed by hot water applied through a hose and nozzle.

The quick-opening gate valves *a* are worked by the bell crank *b* and the lever *c* at the operator's platform.

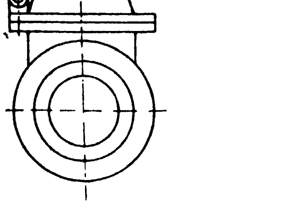


Fig. 354.—Quick Opening Gate Valve.

In *Fig.* 354 such a *Quick Opening Gate Valve* is shown, and in the author's experience these valves will render reliable service when properly cared for.

3.—Automatic Juice Weighing Scales.

Considerable attention has been paid to the design of automatic juice weighing scales, for which an attendant is not required. In the early days of their introduction, erroneous results led to their use being restricted, aggravated as it was by the high cost of upkeep; but nowadays reliable and efficient designs are offered by different manufacturers and a few (of which the author has data) are discussed below.

An error allowance of 0·1 to 0·2 per cent. of the true weight is advised by the manufacturers, and in respect to speed of operation, 40 to 50 weighings per hour can be performed.

An interesting *Revolving Automatic Juice Scale* is diagrammatically shown in *Fig.* 355. Two cylindrical tanks *b-c* and *a-d* are mounted on a shaft *z*, which is supported at both ends in ball bearings *v*, as shown in *Fig.* 356. Both tanks have a division wall in the middle of the tank length, so as to form four compartments *a*, *b*, *c*, and *d*. Of these compartments two (*a* and *b*) are interconnected and hermetically closed, one being filled with water, which serves as the counterweight of the juice to be weighed in the compartments *c* or *d*.

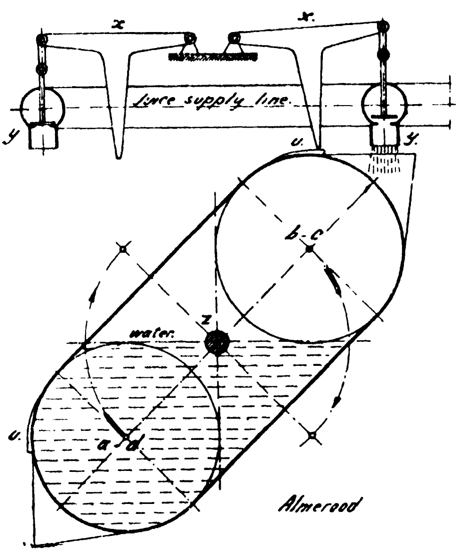


Fig. 355.—Revolving Automatic Juice Scale.

The operation of the scale now is as follows: With the counterweight compartment in its lowest position as drawn in *Fig.* 355, the opposite tank operates a double bell crank *x* by means of the cam *u*, thus opening the right hand charging valve *y* whereupon the compartment *c* is filled. When sufficient juice has been charged to cause equilibrium with the water in *a*, the tank *b-c* will move downwards and the right hand valve *y* closes immediately. The water from *a* flows at once into the compartment *b* as soon as the tanks start moving, thus increasing the acceleration.

To avoid shocks, an oil brake is provided to retard the acceleration when the farthest positions are reached, and the cam *u* on compartment *d* in the meantime has opened the left hand charging valve *y*.

It will be obvious, that the efficient working of this juice scale depends on the rapid closing of the valves *y*, which are especially designed for the purpose, and great accuracy in weighing, with a correction of only 0·1 per cent. of the true weight, is claimed. Maintenance is also very low, as the number of moving parts is very small.

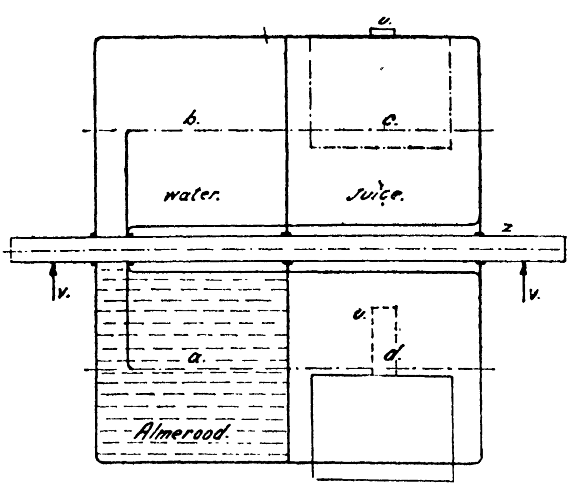


Fig. 356.—Revolving Automatic Juice Scale.

The juice supply line has to be connected to a supply tank, as a direct pressure line from the pumps might cause trouble, as both valves *y* are closed for a short interval of time during each cycle. The compartments *a* and *b* should be smaller than the compartments c and d respectively to avoid spilling of the weighed juice. The discharge outlets are hence so arranged that the juice compartments empty completely, when in their lowest position.

Another advantage of this scale is that it is not affected by accumulated slime, scum or dirt in the juice-weighing compartments. Assuming that W = the weight of the water or the net weight of the juice to be measured, W_1 = the weight of the dirt, etc., in compartment c and W_2 = the weight of the dirt, etc., in compartment d, then for a complete cycle of weighing c and d, the two juice weights obtained are :

$$\begin{array}{ll} \text{by filling } c: & W_c = W + W_2 - W_1 \\ \text{by filling } d: & W_d = W + W_1 - W_2 \\ \hline \text{Juice weight}: & W_c + W_d = 2W \end{array}$$

which shows that the dirt had no influence whatever on the exact weight. It is assumed, of course, that this dirt adheres to the walls of the compartments.

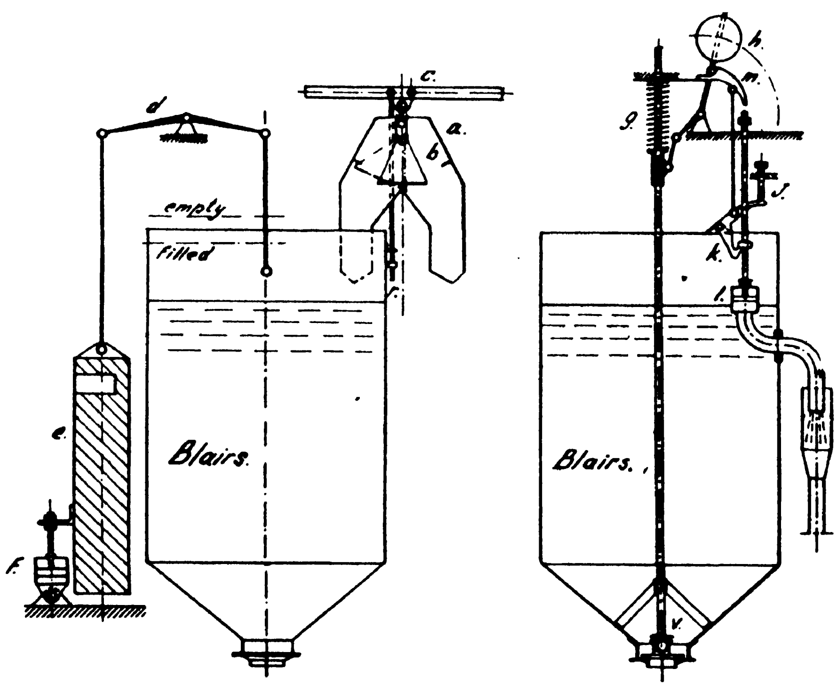
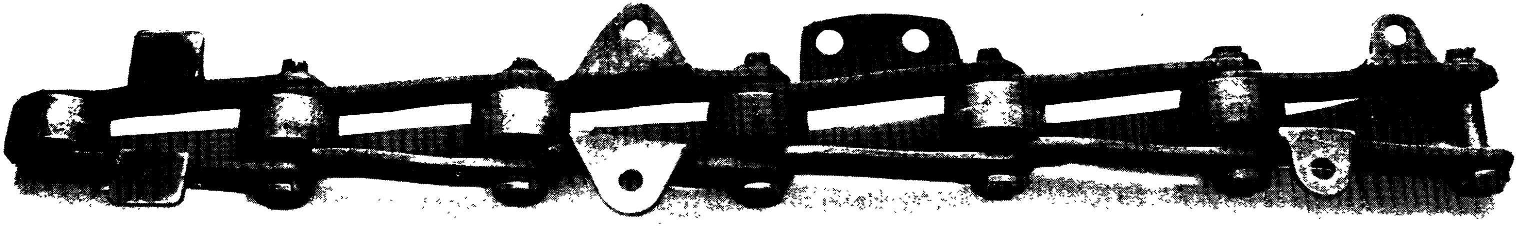


Fig. 357.—Equal Lever Automatic Juice Scale.

This tare-weighing is not considered of equal importance for every cane growing country and the author has found in Cuba, that on a 10,000 lb. juice weighing scale, similar to the one shown in *Fig.* 353, the adhering dirt did not exceed a few pounds on each weighing, thus being well within the accuracy of the factory control. Regular cleaning every six hours was strictly adhered to.

In Java this inconvenience apparently is of greater importance and there tare-weighing scales have found a wide field of application.

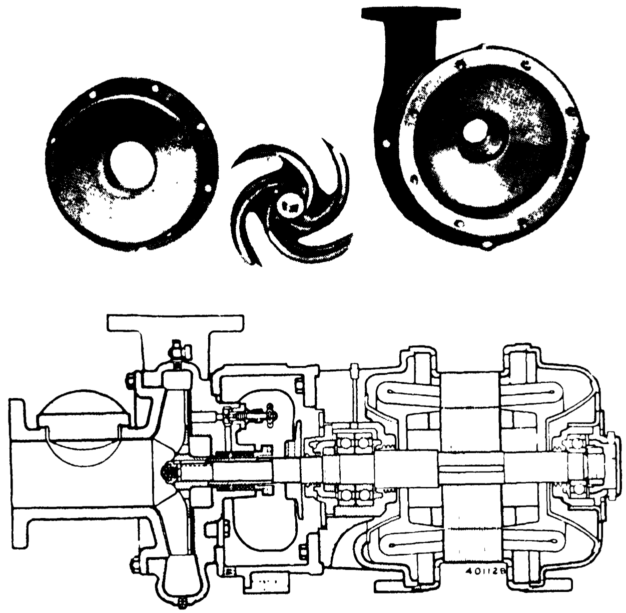
An ingenious patented design is the *Equal Lever Automatic Juice Scale* as shown in *Fig.* 357. Through the hopper a the juice is intermittently charged into each of the two weighing tanks and as soon as equilibrium with the weight e has been reached and thus the tank descends, the rod r will tumble over the



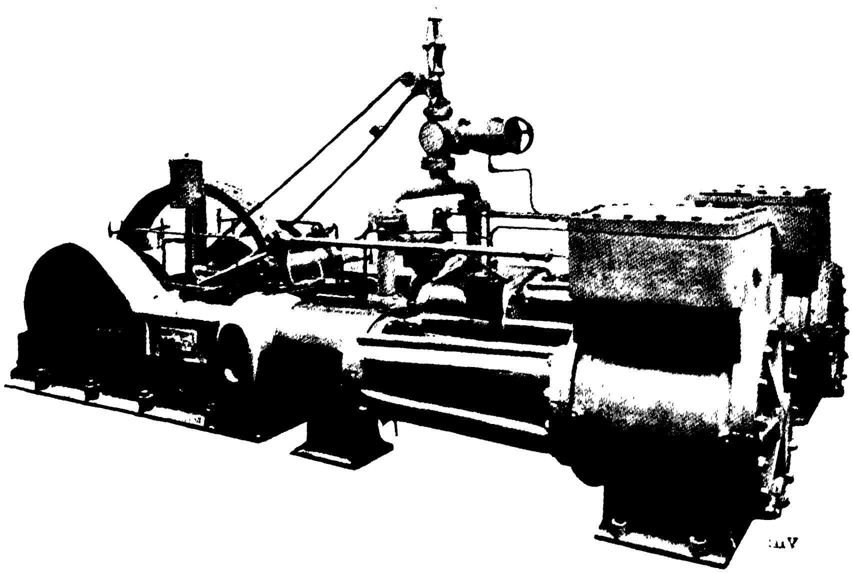
Malleable Iron Roller Chain.
(*Ewart Chain Belt Co., Ltd.*)

Intermediate Carrier Chain.
(*Ewart Chain Belt Co., Ltd.*)

PLATES 61 & 62.



OPEN IMPELLER CENTRIFUGAL PUMP FOR LIME-MILK.
(Ingersoll-Rand Co., Ltd.)

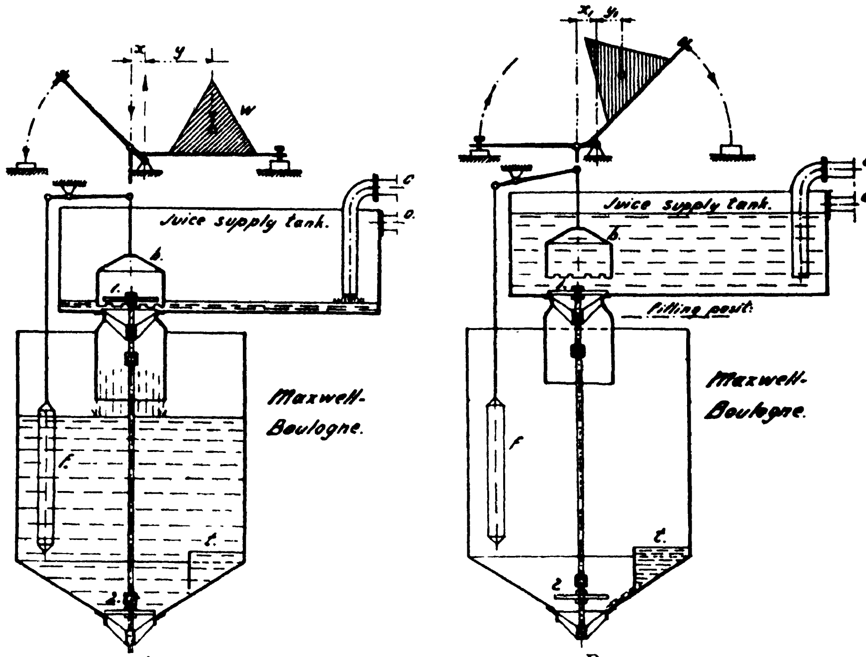


HORIZONTAL DUPLEX STEAM-DRIVEN CO_2 PUMP.
(Worthington-Simpson, Ltd.)

double lever *c*, which is provided with a tube partially filled with mercury. This double lever is connected to a baffle *b* in the hopper or a kindred device, so as to divert the juice charge to the other tank.

The weight *e* has a gap to provide additional weight for the tare, and a dashpot *f* is arranged for smooth non-shock operation.

For starting the movement an excess force is required, and an excess amount of juice is required in the tank, so as to initiate the tilting performance, and an excess above the true weight for which the scale has been rated has thus been charged into the tank. This excess of juice is released by the *surplus valve l* (shown at right side), until the tank starts to rise again. When the true equilibrium has been reached, a lever *k* touches a set screw *J*, releasing the surplus



A B

Fig. 358—.Automatic Tare-weighing Juice Scale.

valve rod, which closes immediately. At the same time a rod, connected to the lever *k*, pulls the arrester *m* down and the hammer weight *h* will tumble over and open the discharge valve *v*. When the tank is emptying, the dashpot will allow the tank to rise until the discharge has been completed. By now the spring *g* will be sufficiently compressed to raise the hammer weight *h* into its vertical position and thus the discharge valve is closed. When the tank is filled again and the surplus valve rod descends, it will be hooked again by the lever *k* and the cycle thus repeats itself.

Another reliable and ingenious design is the *Automatic Tare-weighing Juice Scale*, diagrammatically shown in *Fig.* 358. This scale has been invented in Java, where it is now widely used for juice, imbibition water, syrup, diluted filter-mud and molasses, and it is built for capacities of from 10 to 80 tons of liquid per hour.

Fig. A represents the filling position, the juice from the supply tank having been charged into the juice weighing tank through the large valve 1.

The float f has been raised and by means of a double lever the bell or shield b has lowered over the valve 1 and the serrated lower edge of the former leaves only a small clearance for the juice passage, thus diminishing the juice flow into the tank below. Taking W_j as the weight of the juice in the tank, plus the proper weight of the tank and accessories gravitating on the weighing beam, the following equation is in force at this moment : —

$$W_j \times x \quad < \quad W \times y$$

W being the gravity force of the counterweight, and x and y the prevailing weighing beam levers. As soon as equilibrium has been reached the weighing beam will tilt, the counterweight will rise and the tank descend. At this stage the valve 1 closes and on further descent of the tank, i.e., after valve 1 has been closed, the valve 2 on the tank bottom will be opened and the tank thus be discharged. Both valves 1 and 2 are joined by a flexible rod connexion.

The weigh beam now has come into the position shown in *Fig. B* and the equation of equilibrium reads :—

$$W_j \times x_, \quad > \quad W \times y_,$$

Of the juice a part is retained in a compartment t in the tank, having a small opening at the bottom and this juice will escape through the still open valve 2, until the true equilibrium :—

$$W_t \times x_, = W \times y_,$$

has been reached, where Wt is the weight of the tare-juice in the compartment t plus the dead weight of the tank and connected accessories, the valves 1 and 2 not gravitating on the tank bottom.

The weight W will tilt the weigh beam back into the position of *Fig. A*, whereupon the tank rises and valve 2 closes before valve 1 opens for a new charge.

The weight of one charge should be carefully ascertained on a weighing platform and afterwards the consecutive weighings will be exactly equal to this one.

The design of an automatic scale embodies special features not to be found in hand-operated scales. A new type, the *Automatic Mechanical Weigher*, shown diagrammatically in *Fig.* 359, is in fact a duplex scale, as shown in *Fig.* 353, but having its operation carried out automatically and mechanically. Such an automatic juice weigher has been put in practical operation in Hawaii, with results reported to be good.

The scale is of the reduction lever type and the juice *plus* the dead weight of the tank and accessories, is compensated by the weights m on the weighing beam, which can be adjusted by the poise w. It is therefore adjusted to a fixed weight and quick closing of the charging valve c is thus required.

When tare weighing is required, the scale can be equipped with an electrically operated compensation device and the effect of scum, dirt or adhering residue to the tank walls will not have an unfavourable influence on the true weight. There is a doubt as to the efficiency of tare weighing for raw juice, but it has to be done in the case of fuel oil or molasses.

The *modus operandi* of the automatic weigher now is as follows :—

Both tanks are operated alternately, but the working of one tank is just the same as that of the other. The juice is supplied in a receiving tank a, the excess being released by an overflow b and the tank is of sufficient capacity to fill both weighing tanks h. These latter tanks are filled by the valve c, protected by a shield d for regulating the juice supply at a pre-determined rate. A pipe e extends into the weighing tank, having sideways discharge, so as to reduce the inertia forces on the weighing levers.

A float is provided in the supply tank, that will break the electric circuit to the operating motor, in case there is not sufficient juice for a complete weighing.

Another float f in this same supply tank operates a compensation device g for difference in hydrostatic head in the supply tank for equalizing the effect of the inertia of the juice stream in the weighing tank, and the juice flow through the valve c.

The weighing tanks h bear upon levers j, of triangular compounded type, the reaction being transferred by reducing levers k to the weighing beam l.

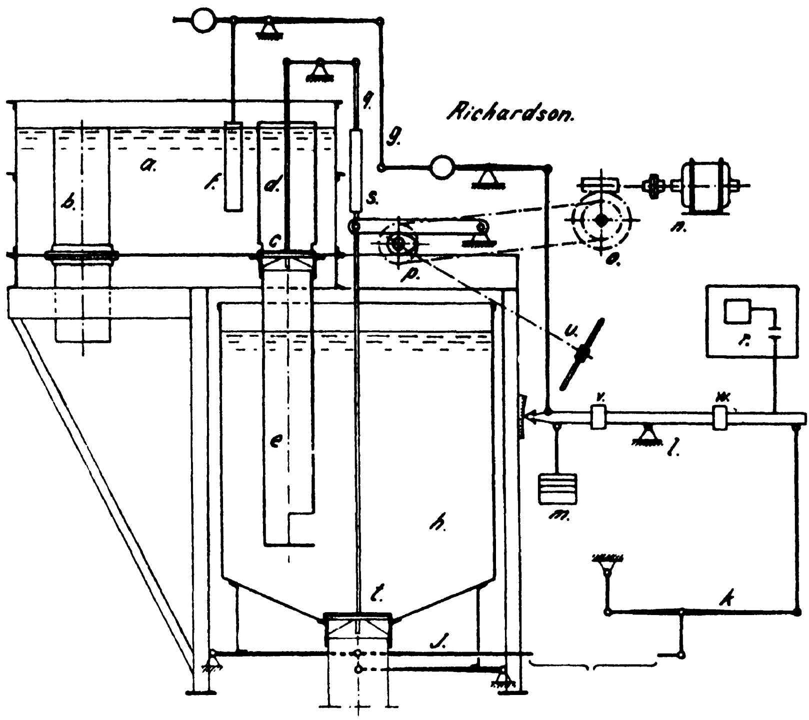


Fig. 359.—Automatic Mechanical Weigher.

The minimum level switch energizes the motor n, which drives by a worm and chain drive o the camshaft p, mounted lengthwise over the weighing tanks and which carries 4 cams, two for operating the discharge valves and two for the inlet valves. The corresponding cams are set 180° apart for alternate operation of the valves. When the motor has rotated the cam shaft half a revolution, the valve c is opened and the tank below will be filled. The motor stops automatically at this position.

As soon as the juice weight makes balance with the weight m on the weighing beam, the latter tilts over and closes a mercury switch r, which operates the electro-magnetic trigger s on the valve rod q and the valve c closes instantaneously. This same trigger is operated mechanically, in case the electric operation fails.

Through the closing of the inlet valve, another mercury switch is operated, which energizes the motor again for another half revolution of the camshaft and the discharge valve *t* is opened at the same time as the charging valve of the companion tank.

A test switch is provided for opening the motor circuit, so it can be readily ascertained at the pointer of the weighing beam, whether the full tank makes true balance or not.

When the tank is empty, the juice weight has to be taken off at *m* and the poise *w* is for rectifying the true empty balance. In case correction is required for dirt or scum the poise *v* has to be set.

If the electric current fails, the whole scale can be operated by hand by means of the hand wheel *u*, which drives, through a bevel gear, the camshaft *p*.

This automatic weigher is guaranteed for a tolerance of 0·25 per cent. of the true weight. Official tests with water showed only 0·02 per cent. difference with the true weight during 40 weighings of each tank.

An hourly capacity of 180 short tons of juice is stated by the manufacturers.

CHAPTER XVII.

LIME AND SULPHUR STATIONS.

LIME KILNS — MILK-OF-LIME PREPARATION — SULPHUR OVENS.

A necessary reagent for all cane sugar factories is lime, and while raw sugar factories, which use a relative small quantity, buy quicklime in barrels, carbonatation and sulphitation factories generally produce their requirements by burning limestone or lime rock in kilns. Sometimes, the lime kilns are worked under a co-operative scheme, so as to supply from one source all the lime required by several adjacent factories.

Sulphur is a bleaching and coagulating agent for the production of plantation white sugar and it gives great brilliancy to juice treated with it. Sulphur is applied in the form of sulphur dioxide (SO_2) and the pure sulphur or brimstone has to be burnt to produce this oxidation of S.

The chemistry, so far as is required for the dimensioning of the apparatus and its explanation, as well as the construction and operating details, are given in the consecutive sub-sections below.

1.—Chemistry of Lime and Sulphur.

Lime is found as lime rock or limestone generally close to the earth's surface and some cane growing countries have it available in nearly pure form. Impurities such as adhering organic soil are not as prejudicial as are combined or associated impurities. Where possible the purity of limestone should be about 97 to 98 per cent. and a magnesia content of over 1 per cent. will render it of inferior quality for sugar manufacture.

The chemical formula of limestone is $CaCO_3$ and the atomic weights are as follows :—

$$\begin{aligned} Ca &= 40 \\ C &= 12 \\ O_3 = 3 \times 16 &= 48 \\ \hline CaCO_3 &= 100 \end{aligned}$$

In the lime kiln the $CaCO_3$ will dissociate at a calcination temperature of a little below 1832°F. (1000°C.) into CO_2 (carbonic acid) and CaO (calcium monoxide or quicklime) in the following proportions :—

$$\begin{aligned} CO_2 &= 12 + 2 \times 16 = 44 \\ CaO &= 40 + 16 \qquad\quad = 56 \end{aligned}$$

thus pure limestone will furnish by complete dissociation 44 per cent. CO_2 and 56 per cent. CaO by weight.

The kiln may sometimes develop too high a temperature through excessive fuel, and actual temperatures may rise to over 1350°C. (2460°F.) which will result in overburning, making the quicklime inert and thus unsuitable for the slaking performance. An even size of lime rock for charging the kiln is thus of paramount importance, as all pieces will thereby require approximately the same temperature to calcine. With uneven sizes the larger pieces may remain unburnt in the centre and the small pieces be overburnt.

The old kilns had separate furnaces in which any fuel could be burnt, the ash having no influence on the burnt limestone. In the Khern or Belgian kilns, coke is charged intermittently with lime rock, so the ash of the coke remains for the greatest part in the quicklime, but this will not prejudice the sugar making process. For each 1000 lbs. of lime rock, about 100 lbs. of foundry coke are required. The amount of heat needed to burn each lb. of limestone is about 1250 B.Th.U. (700 cal. per kg. limestone).

The weight of the lime rock in relation to space occupied depends on the size, and the author has fixed it for 4 to 6 in. sizes at about 80 lbs. per cub. ft., whereas foundry coke weighs 30-32 lbs. per cub. ft. under the practical operating conditions of the kilns.

For the Khern as well as for the separately fired kilns, the CO_2 of the combustion gases of the coke or other fuel will be present in the kiln gases in the following proportions :—

100 lbs. lime rock, $CaCO_3$, will produce	44 lbs. CO_2
10 lbs. foundry coke, having 88 per cent. C will produce $0{\cdot}88 \times 44 \div 12 \times 10 =$	32 lbs. CO_2
Total	76 lbs. CO_2

Practical figures can be taken as 50 lbs. CaO and 70 lbs. CO_2 from each 100 lbs. of lime rock burnt.

In the carbonatation process, the CaO added in the form of milk-of-lime to the juice will be combined again to $CaCO_3$ by the CO_2, and the quantity of CO_2 produced by the kiln will be in excess of the required one, a convenient circumstance, as the efficiency of so-called carbonatation is between 50 and 80 per cent., the rest of the CO_2 being lost up the chimneys of the carbonatation tanks. The acidity of the juice will of course consume a part of the CaO that does not require consecutive carbonatation.

Due to the excess air necessary for the combustion of the coke or other fuel, the CO_2 content of the kiln gases will not exceed 33 per cent. by volume and generally 25 to 30 per cent. is the rule.

The kiln gases are cooled and purified in a scrubber and the temperature drops to about 140°F. (60°C.), sometimes even lower. At this temperature one lb. CO_2 occupies, as per formula (84) of Chapter XII, 9·8 cub. ft. (1·61 cub. m. per kg.) and in round figures 10 cub. ft. per lb. or 0·1 lbs. per cub. ft. will be correct for practical calculations.

In carbonatation factories, where the CO_2 is pumped by a carbonic acid pump, the displacement of such a pump with 25 per cent. CO_2 in the kiln gases has to be 40 cub. ft. per lb. CO_2 or about 56 cub. ft. per lb. CaO produced.

The capacity of the kilns is very flexible in respect to the amount of lime to be burnt per 24 hours, and the author has data at hand showing between 23 and 77 lbs. CaO produced per cub. ft. lime kiln volume in 24 hours (370 and 1230 kg. per cub. metre) ; and the average for new kilns should be around 30 lbs. CaO per cub. ft./24 hours (500 kg. per cub. metre). Calculated on the basis of $CaCO_3$, double the weight has to be taken.

By adding water to the quicklime, thus hydrating or slaking the lime, a considerable amount of heat is liberated and the lime-milk sometimes becomes boiling hot, causing a slight evaporation of the water used. The hydrating performance works out as follows :—

$$CaO + H_2O = Ca(OH)_2 \text{ (slaked lime)}$$

In the carbonatation tanks CO_2 is forced through the limed juice, when the reaction will be :—

$$Ca(OH)_2 + CO_2 = CaCO_3 + H_2O$$

and the $CaCO_3$ is of granular structure of very good filterable capacity.

Sulphur is of volcanic origin and generally purified at the refineries or mines. The combustion of sulphur produces :

$$S + O_2 = SO_2 \text{ (Sulphur dioxide)}$$

with an atomic weight of $32 + 2 \times 16 = 64$, so it will be obvious that one pound of sulphur requires one pound of oxygen for its combustion, which takes place at a temperature of 657°F. (363°C.). Moisture in the air, necessary for combustion, is prejudicial to the metal of the tanks and equipment, as the formula :

$$SO_2 + H_2O = H_2SO_3$$

will easily become, by further oxidation, H_2SO_4 (sulphuric acid).

When the sulphur oven or burner is not properly cooled by means of a water jacket, the temperature may rise and sulphur in gaseous form will be carried along in the pipelines and return to the fixed state as soon as it has been cooled, thus causing the dreaded sublimation, which may clog the pipelines for the SO_2 and is difficult to remove.

For the sake of economy the author has tried crude sulphur from the mines, 99 per cent. pure, but it did not burn well as it formed a tarry layer on top of the molten sulphur in the burner, impairing the combustion. The crude sulphur had to be discarded and 100 per cent. pure brimstone used, instead, for good operating results.

In the sulphitators the following reaction takes place :

$$CaO + SO_2 = CaSO_3 \text{ (calcium sulphite)}$$

For carbonatation factories where sulphitation is used after the carbonatation, the reaction is similar, thus :

$$CaCO_3 + SO_2 = CaSO_3 + CO_2.$$

Due to the nitrogen carried along with the combustion air, the percentage of SO_2 will be about 12 to 16 per cent. in the gas produced in the sulphur oven.

As the proportion of the atomic weights of CaO and S is as 56 to 32, each lb. of free CaO will require about 0·57 lbs. S.

In the author's practice in a sulphitation factory, 0·27 lbs. sulphur has been burnt per 1000 lbs. of cane ground, whereas in a carbonatation factory only slightly less (0·25 lbs. sulphur per 1000 lbs. cane) was used.

The area of the sulphur furnace, thus the locus where the sulphur is burnt, may be taken as 1 sq. ft. burning area for each 5 lbs. of sulphur to be burnt per hour (25 kg. per sq. metre per hour).

2.—Lime Kilns.

The size of the lime kilns can be calculated from the data given in the previous section, and a vertical kiln, made up of two cones of which the lower one is inverted, is the general design. Inadequate diameters have resulted in the limestone forming arches within the kiln, which not only prevent one drawing off the burnt lime at the bottom, but tend to clog the kiln completely and no draught can be produced. A diameter of 8 to 9 ft. is considered a normal one for large kilns of above 2000 cub. ft. volume. When arching occurs, the mass of limestone has to be broken up by ramming in iron bars from the top or inserting them through the peep holes. This ramming cannot always be done without causing damage to the refractory lining of the kiln.

In *Fig.* 360 is shown a *Small Lime Kiln,* measured by the author at a raw sugar factory, requiring about 5400 lbs. lime per 24 hours. The casing of the kiln is made of an old boiler shell and the two furnaces for external firing are made up also from old boiler material. The furnaces are arranged diametrically opposite and firewood is burnt on ordinary grates.

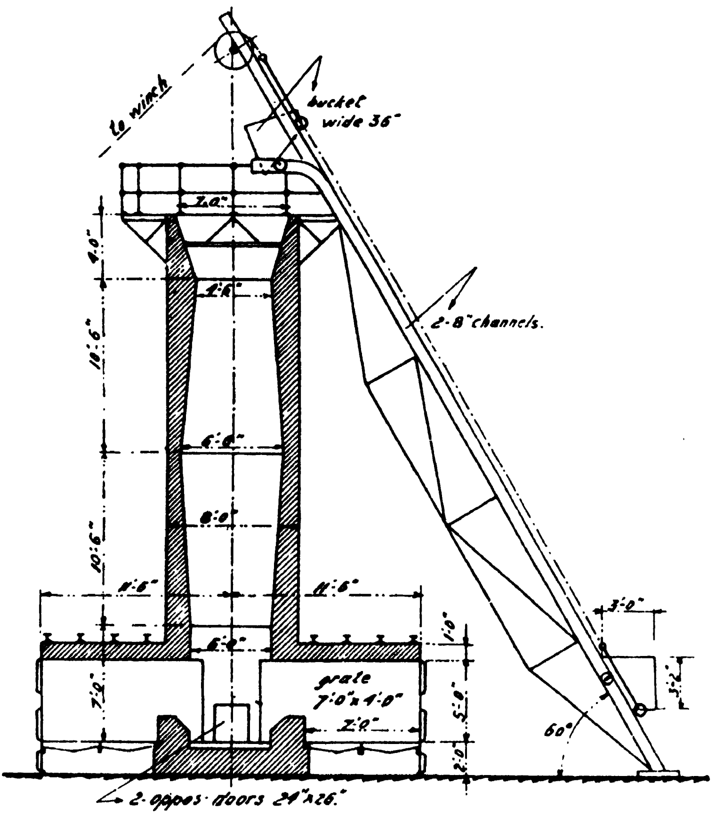


Fig. 360.—Small Lime Kiln.

As the fuel and limestone are not charged together, the quicklime is very pure and will not contain ash. It should be recollected that the calcination of large lumps of lime rock requires a higher temperature than that of smaller pieces, and pieces from 4 to 6 in. should be used. The size of the lime rock should be as even as possible.

The kiln shown has sometimes caused arching and this may be due to the upper cone being too short and the lower cone too long. Moreover, the heat of the furnace is delivered at a low level in the kiln, thus producing a low calcination zone and insufficient cooling of the quicklime, which is sometimes withdrawn at a red hot temperature.

The kiln is charged by an inclined elevator, having a bucket on four wheels, the top ones being guided between the channel flanges and the lower ones moving on the top flanges. The hoisting cable is attached by a forked tie rod, which will tilt the bucket, when on top of the kiln. The elevator cable is exposed to the hot gases emerging from the kiln mouth and suitable protection should be provided.

A conical shield is placed in the mouth of the kiln for distribution of the limestone, but in case of kilns of greater height the falling pieces might damage the refractory lining.

In *Fig.* 361 is shown the *Refractory Lining* of a Khern or Belgian type lime kiln of 1920 cub. ft. volume, as supplied by the author. This kiln has been arranged to allow for the withdrawal of the CO_2 gases from the top. The

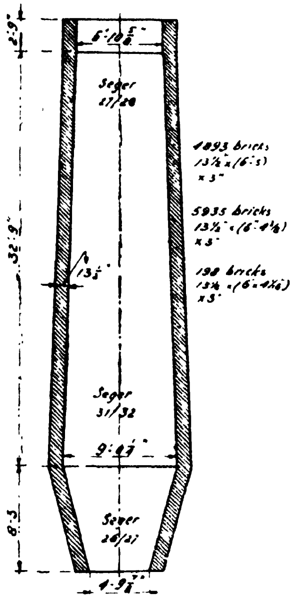


Fig. 361.—Refractory Lining.

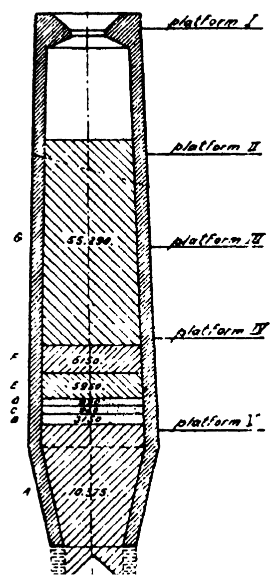


Fig. 362.—Initial Filling.

upper cone is long and the lower one short, the calcination zone being at the lower level of the upper cone. The kiln is charged with limestone and about 10 per cent. by weight of foundry coke, having a size of 3 in. and up.

The refractory lining of lime kilns should receive first attention as it will be subject to high temperatures and, moreover, the lime or the gases are liable to penetrate the joints or the pores of the material and associate with it, causing a combination having a much lower fusion point than the brick material itself.

The joints have to be made of the same quality of material as the refractory brick and as thin as possible, this necessitating an exact fit of the bricks. The fire bricks have, moreover, to be resistant to flaking, cracking and abrasion. The shrinkage after cooling must be very small, as otherwise the joints will open up; and the refractory capacity should also be high.

The best quality of brick, even at a higher initial cost, will prove economical in the long run. The thickness is usually 13½ in. and the best practice is to have this thickness of first class material, especially in the calcination zone, over a length of about 20 ft. Sometimes a first-class material is used on the inside, having 9 in. thickness and a layer behind it of second-class material, having a thickness of 4½ in. Between the refractory lining and the shell, a joint of asbestos material is sometimes laid, to take up the expansion of the bricks. There is a first-class material on the market for which this provision can be neglected, the expansion being practically zero.

In Java the following fusion points of refractory bricks for lime kilns are recommended :—

Upper cone—Upper part	Seger cone 30/31
Upper cone—Calcination zone....	„ „ 34/35
Lower Cone—Cooling zone	„ „ 29/30

The composition of common fire bricks is silicious clay or silica, but a high percentage will give a low fusion point and the best high temperature brick for lime kilns should therefore have a high alumina content, thus :

41 — 43 per cent. Al_2O_3 and 48 — 49 per cent. SiO_2.

As important as the fusion point is the structure of the brick, which has to be made of the finest powdered material, so as to give a low figure for porosity. The bricks are firmly pressed when manufactured and there are now natural refractory bricks on the market which have the highest degree of compactness.

The porosity is measured by the increase in weight, when the brick is immersed in water, 16 per cent. being a very good quality figure. The expansion coefficient should be less than 0·1 per cent. under operating temperature and the baking temperature is an indication as to the fusion point. As there are refractory bricks which have withstood higher operating temperatures than the laboratory fusion point, practical performance in lime kilns will be a guide when buying refractory bricks for this class of work.

For kilns, key size bricks are employed and the author has used :

Key No. 1 .... 9 in. × (4½ — 4 in.) × 2½ in.
Key No. 2 .... 9 in. × (4½ — 3½ in.) × 2½ in.

Key No. 1 .... 13½ in. × (6 — 5 in.) × 2½ in. or 3 in.
Key No. 2 .... 13½ in. × (6 — 4⅜ in.) × 2½ in. or 3 in.

Two to 3 per cent. allowance in dimensions is the accepted standard. The bricks for lime kilns weigh about 140 lbs. per cub. ft., as laid, but the specific weight of the material is about 200 lbs. per cub. ft.

The fusion point of fire brick is measured by the Seger Cones but there is a discrepancy as to the observation, some countries taking the beginning of the bending of the cone, whereas others choose the moment when the cone has completely bent over. For lime kiln work, the following Seger cones and corresponding temperatures are given :—

S.C.	°F.	°C.		°F
28	3074	1690	in U.S.A.	2975
29	3110	1710	„	3002
30	3146	1730	„	3038
31	3182	1750	„	3065
32	3218	1770	„	3101
33	3254	1790	„	3128
34	3290	1810	„	3164
35	3326	1830	„	3191

The *Initial Filling* of the lime kiln is shown in *Fig.* 362, referred to the same kiln as mentioned in the previous figure. As will be seen, the kiln is provided with 5 stages or platforms, and the shell has 4 peep holes on each stage, about 6 in. in dia. and provided with sliding covers, so as to observe the work of the kiln, and with unequal burning, air can be admitted through any of them at the spot desired.

The filling is composed as follows :—

		wood	coal	coke	limestone
A ..	firewood	10375 lbs.			
B ..	coal		3150 lbs. ..		
C ..	coal and coke (vol. 1 : 1)		560 lbs. ..	360 lbs.	
D ..	coke..................................			990 lbs.	
E ..	limestone and coke (vol. 1 : 1)..................			1650 lbs. ..	4300 lbs.
F ..	ditto (vol. 2 : 1)..................			990 lbs. ..	5160 lbs.
G ..	ditto (vol. 3 : 1)..................			6270 lbs. ..	49020 lbs.
	Totals	10375 lbs. ..	3710 lbs. ..	10260 lbs. ..	58480 lbs.

The material is loaded in 16 in. gauge tip-cars, each having about 9 cub. ft. capacity, the average loading being about 10·5 cub. ft. and having the following average net weights of material :—

coke	330 lbs.
coal	525 lbs.
limestone ..	860 lbs.

For operating two kilns four cars are required, the lift needing about 60 sec. for each car. The top platform is provided with rails laid in loops round the kiln mouths for even charging.

Taking the heat value of firewood as 4500 B.Th.U./lb., that of coal as 11,000 B.Th.U./lb. and that of coke as 12,500 B.Th.U./lb. the total heat amounts to 179,018·500 B.Th.U. or 3061 B.Th.U. per lb. of limestone. As the average heat requirement is only 1250 B.Th.U. per lb. of limestone, excessive heating is the result with too high temperatures, doing damage to the refractory lining ; and about 1800 B.Th.U. should be considered normal, when starting the kiln.

The lower opening of the kiln, where the quicklime will be extracted, is temporarily filled with refractory bricks in checker board fashion to allow for entry of air. These have to be kept clear of ash, so as not to prevent the air entering.

The top or *Mouth of the Kiln* is shown in *Fig.* 363. For filling purposes an inverted cone is provided, the centre opening being closed by a welded steel bell *a*, having a small seat on the cone, which cannot be covered by large pieces of lime rock, the dust or grit being of no importance for closing. The bell is hoisted or lowered by a small hand-operated hoisting drum and the carbonic acid is drawn in on both sides at *c*, so as not to divert the burning of the kiln to one side. The openings in the refractory lining at the locus of the CO_2 outlets slant at the bottom, to prevent accumulation of lime or fuel. The peep hole openings in the lower part of the upper cone are made accordingly.

A safety grating is provided on top of the cone mouth and flat irons, $1\frac{1}{2}$ in. $\times$ $\frac{3}{4}$ in., spaced about 9 in. at the outer circumference, will withstand the rough duty, when the tip-cars are emptied. These bars *e* are welded with braces to bent or straight flat irons *f* and every four bars form a removable section. The grates are supported on the angle ring *g* and the flat iron ring *h*.

The charged limestone and coke will fill the space *b* and should be equally distributed around the bell *a*. As soon as the bell is hoisted, the charge will be dumped into the kiln, falling free of the refractory lining.

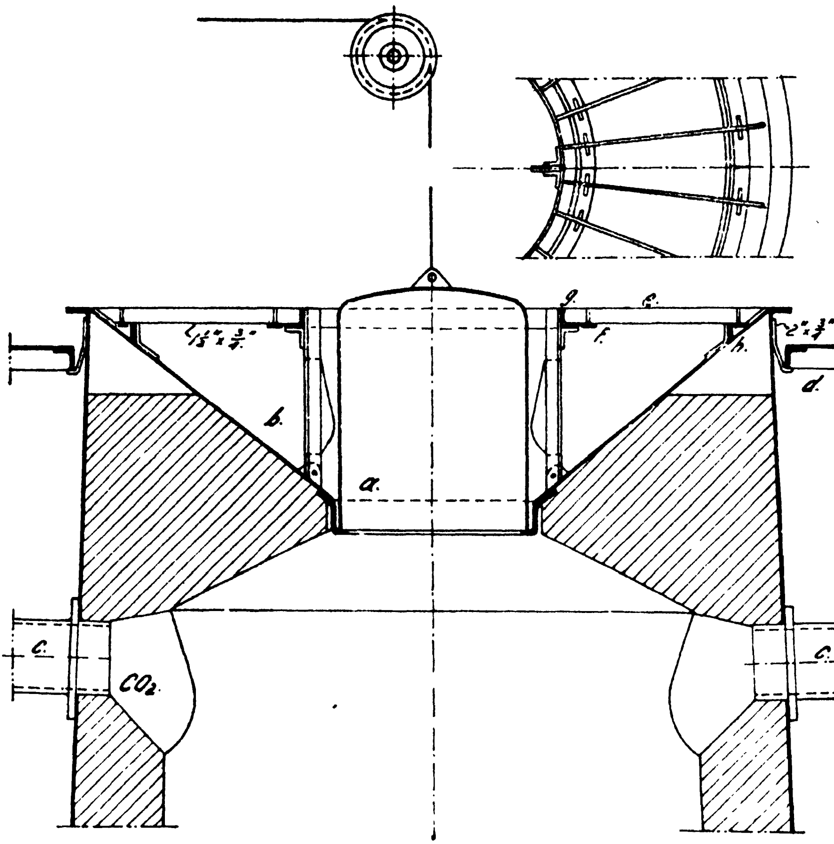


Fig. 363.—Mouth of the Kiln.

The upper platform is indicated by *d*, mounted on a ring of angle iron, having some clearance from the shell of the kiln, as the circular form sometimes suffers through rough handling of the plates during transport. The supporting angle rings are attached to the kiln by bent flat irons of 2 in. × $\frac{3}{8}$ in. and they can be easily adjusted at site. The floor sheeting for the stages or platforms is made of chequered steel plate. Sufficient stairs or ladders have to give access to these intermediate platforms.

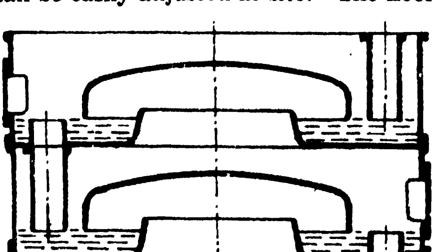


Fig. 364.—CO_2 Scrubber.

The shell material is from $\frac{1}{4}$ in. to $\frac{3}{8}$ in. thick, the lower cone having the heaviest material as well as the calcination zone. The kiln is supported on six columns of cast iron or rolled H-beams.

As the CO_2 gases will carry along a certain amount of dust, or sometimes tarry matter, they have to be cleaned, before reaching the CO_2 pump. The pipe lines can be made of cast iron or steel, as the CO_2 has little corrosive effect, and sulphur is not to be looked for in these gases. The gas cleansing is done in a washer or *CO_2 Scrubber* as shown in *Fig.* 364, composed of 6 compartments having about 3 ft. 6 in. outside dia., similar to the two shown in the figure, which are tightly bolted together by three or four tie rods, the bell-type joints being packed with cement. The CO_2 gases enter at the bottom and are forced six times to pass a watershed before emerging at the top. The dust is precipitated by the water, which is applied by a 2 in. connexion at the top, and, moreover, the gases are cooled, thus reducing their volume and the corresponding pump displacement required. Each compartment has cleaning doors for removal of the precipitated dirt. As the scrubber is under a partial vacuum of about 5 in. mercury, it has to be placed at such a height that the cooling water can be released by a barometric leg pipe about 10 ft. long, which discharges into a seal tank.

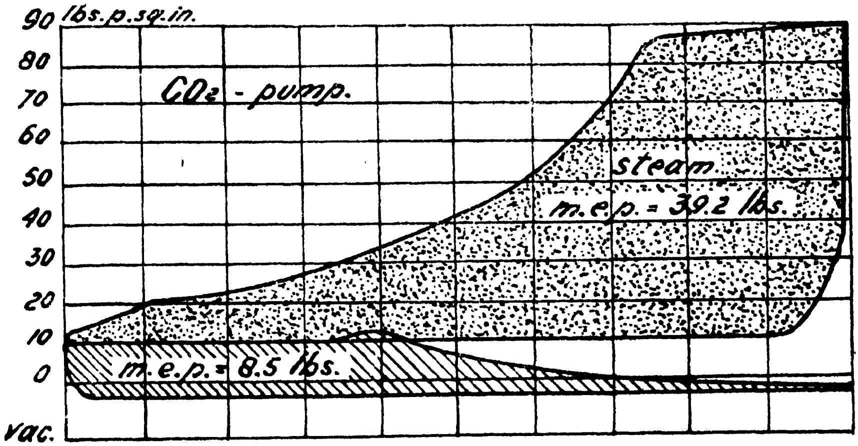


Fig. 365.—Diagram of Pump and Steam Cylinders of CO_2 Pump.

As some water might be carried along, a water separator is placed at the exit of the scrubber, having about 40 in. dia. and 6 ft. height, and provided also with a barometric seal pipe.

At the top of the kiln, where the two branches of the CO_2 pipe line join, a tee-piece is arranged, one side connecting to the scrubber and CO_2 pump, whereas the other side has a chimney of 12 to 16 in. dia. mounted on top, to operate the kiln by natural draught when the CO_2 pump is not working, e.g., during stoppages or when starting the kiln.

The CO_2 pump is generally of the piston type, similar to the dry vacuum pumps to be dealt with later on. A cooling jacket around the cylinder is not required on account of the small difference between suction and discharge pressures, and thus only a light compression performance is required.

The indicated performance is shown in *Fig.* 365, being a *Diagram* of the steam and pump cylinders of a CO_2 pump; 90 lbs. steam pressure and 10 lbs. back pressure is the order, the pump suction being 5 in. and the maximum discharge pressure 10 lbs. per sq. in.

The mean effective pressure for the steam end with quarter cut-off is 39·2 lbs. per sq. in., whereas for the pump end it only amounts to 8·5 lbs. per sq. in.

As the piston areas are proportionate to the square of the respective diameters, the diameter of the pump piston for this pump performance should be 2·15 times as great as the steam piston diameter. The pump has an 18 in. steam cylinder, 36 in. pump cylinder and 28 in. stroke, running at 80-100 r.p.m. in a carbonatation cane sugar factory, having two lime kilns of 1920 cub. ft. capacity each.

For higher steam pressures, the steam cylinder diameter can be made smaller, to allow a normal cut-off of about one quarter of the stroke.

Regulation of the displacement of the carbonic acid pump can be obtained by having a connexion between the pump suction and the pump discharge, this connexion being provided with a gate-valve, which can be set at wish, thus allowing a part of the compressed gases to flow back to the pump suction. Regulation of the engine speed will give a higher efficiency as regards power consumption.

Rotary blowers are also used for this service, but they do not have the efficiency of a piston pump, as the piston seal between suction and discharge is superior to the seal of revolving blades or impellers. The obtainable pressure is also less for the latter, but the first cost of these blowers will be less and a careful estimate as to initial and running costs should be made. The blowers, as well as the piston pumps for CO_2, may be driven electrically.

3.—Milk-of-Lime Preparation.

For the process work of the cane sugar factory, quicklime is hydrated to milk-of-lime, as this will easily mix with the juice. The apparatus for hydrating the quicklime, as it is produced by the lime kiln, is the *Lime Slaker*, as shown in *Fig.* 366. The author has installed such a slaker to the given dimensions for slaking about 4500 lbs. CaO per hour, but a few improvements are added here which will make the slaker more feasible for large capacities.

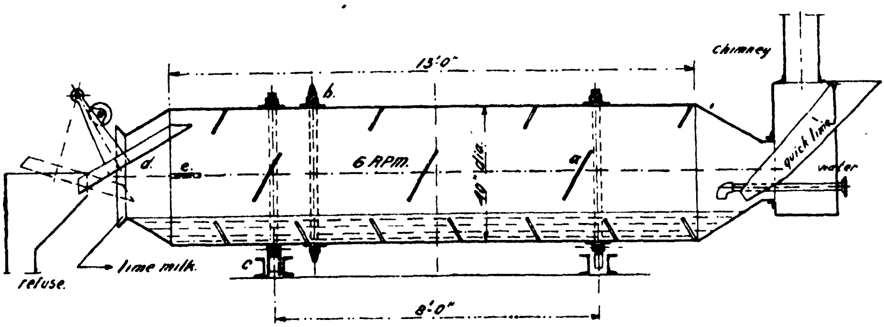


Fig. 366.—Lime Slaker.

It is mainly composed of a drum revolving at about 6 r.p.m., the quicklime being charged at one end, the inlet being about 16 in. in dia., at which end also is made a water connexion for the hydrating performance. The pieces of quicklime, not yet slaked, are moved forward by means of baffles *a*, which are strips welded on the inside of the drum, forming a spiral up to the discharge end. The milk-of-lime is discharged at the other end of the drum by an outlet about 24 in. in diameter, or larger than the inlet. Unslaked pieces of lime, stone or grit are lifted by a horizontal baffle *e* and this refuse can be discharged down a chute *d*, revolvably mounted on a shaft, outside the slaker.

The drum has two runners of chilled cast iron, which bear on four chilled cast iron rollers. One set *c* of these rollers has shrouds, so as to keep the drum in place.

The openings for charging and discharging are made as cones, so as to avoid splashing the lime-milk outside the drum. As the amount of heat produced by slaking the large quantity of quicklime mentioned is considerable, it leads to heavy evaporation and the ensuing vapours have proved a hindrance in this station, for which reason a chimney has been provided at the charging end, mounted on a steel casing in which the inlet of the slaker can revolve.

The drum is driven by a chain drive, the driven rim *b* being mounted on the drum the same way as the runners, by means of pieces of angle iron. The intermediate shaft, carrying the driving sprocket, can be located at any convenient place close to the slaker.

For slaking, fresh water or sweet-water from the filter-presses is used. The amount required depends on the density of the milk-of-lime and as this density generally is measured in degrees Baumé, the following table is for the accepted practical densities; 15° Bé is the one mostly used, but carbonatation factories, using a larger quantity of lime-milk, sometimes prefer 20° Bé, which is about the maximum for operating conditions, as heavier densities may cause difficulties as regards the lime-milk pump and the clogging of the lime-milk pipe lines.

Density.	Sp. Gr.	Milk-of-lime: Grms. CaO per Litre.	Milk-of-lime: Per cent. CaO by Weight.	Lbs. Water required per lb. CaO.
10° Bé	1·075	94	8·74	10·4
15° Bé	1·116	148	13·26	6·5
20° Bé	1·162	206	17·72	4·6

As the milk-of-lime, when it emerges from the slaker, contains grit, ash and unburnt or unhydrated pieces of lime and coke, it has to be strained, but straining can only be done over coarse perforations as fine strainers easily clog. The decanting or settling method, therefore, can be applied with advantage.

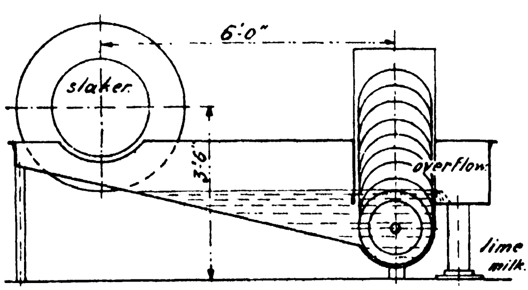


Fig. 367.—Lime-milk Classifier.

In *Figs.* 367 and 368 such an apparatus, called a *Lime-milk Classifier*, is shown; this is the author's design and has given good operating results. The lime of the slaker falls into a tray with an inclined bottom—*Fig.* 367—and is led to the lower part of an inclined scroll conveyor, which will transport the settled impurities above the lime-milk level. Floating impurities are retained by a division wall in the charging tray and can be easily skimmed off. The scroll rotates slowly, so as not to produce any stirring effect. The dissolved lime is delivered by an overflow into the pipe line to the lime-milk stirrers.

At *a* in *Fig.* 368 a water drip is arranged so as to prevent any adhering lime being discharged to the refuse opening. The driving is carried out by

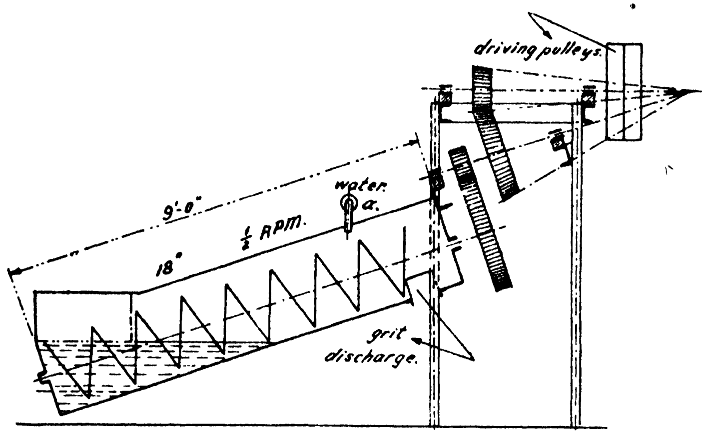


Fig. 368.—Lime-milk Classifier.

two sets of gears and a belt, and the apparatus has classified about 30,000 lbs. (2700 Imp. gals.) of lime-milk per hour.

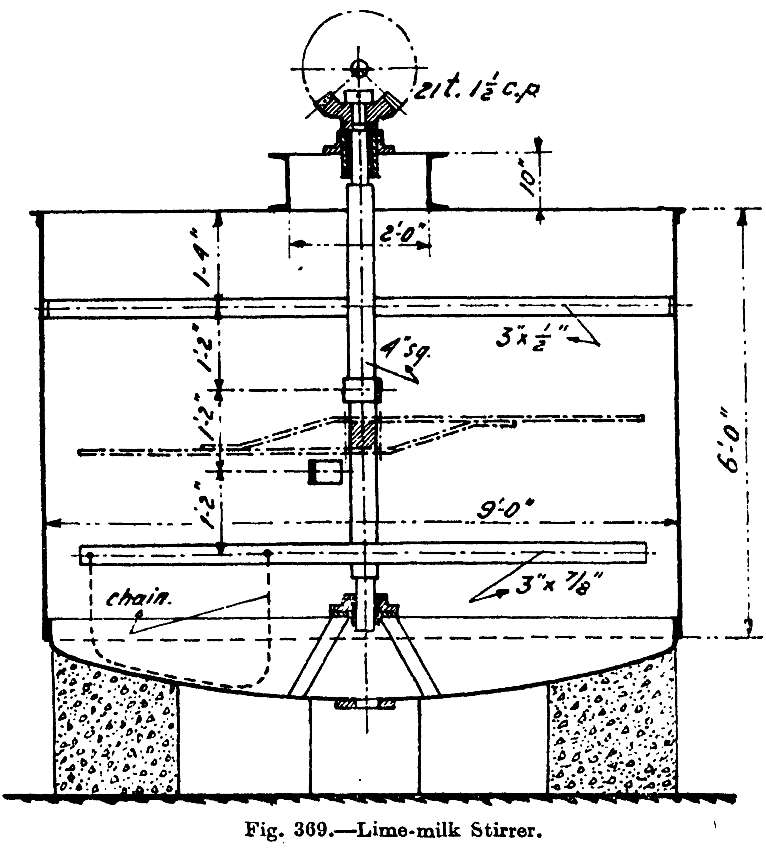
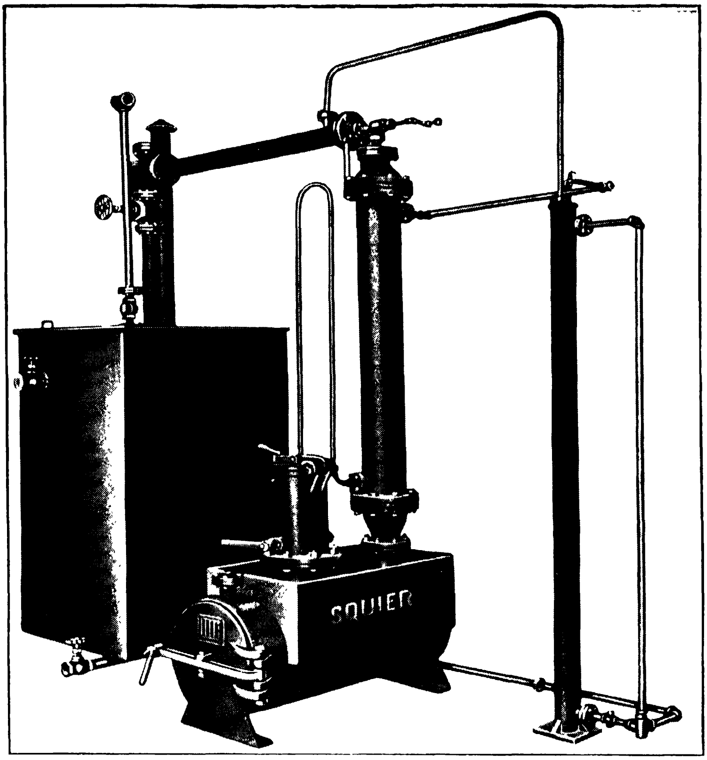
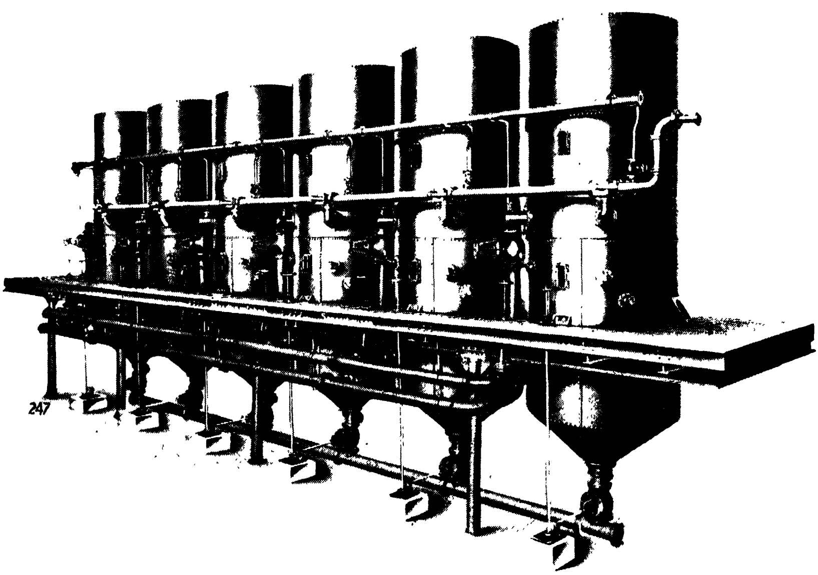


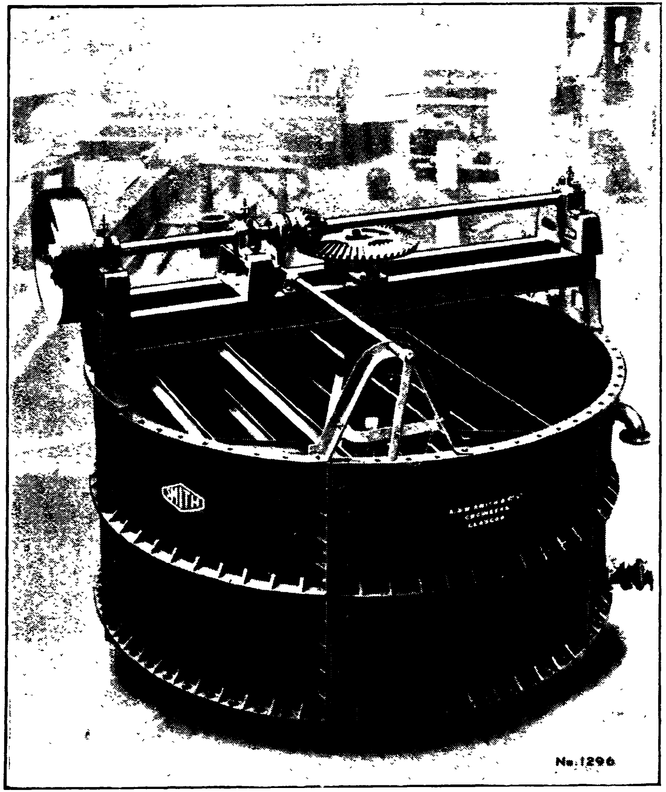
Fig. 369.—Lime-milk Stirrer.



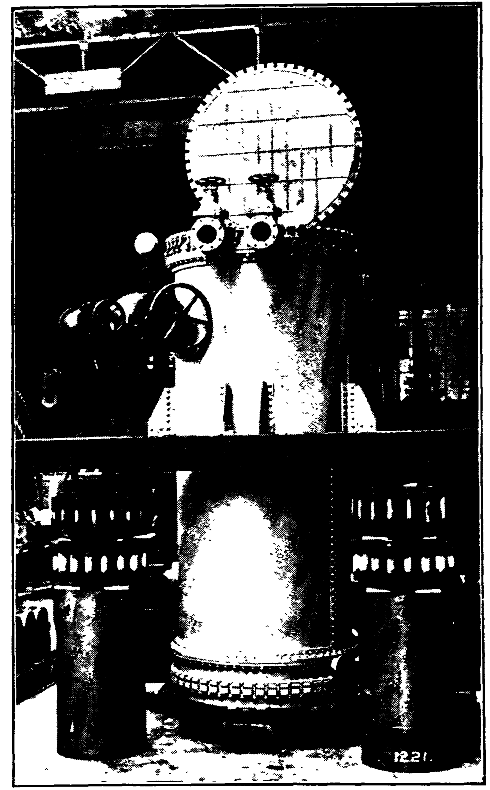
SULPHURING OUTFIT FOR STICK OR FLOUR SULPHUR.
(*Squier Manufacturing Co., Inc.*)



CARBONATATION OR SATURATION TANKS ON PLATFORM.
(*Maschinenfabrik Sangerhausen*)



LIME STIRRER IN CAST IRON TANK.
(A. & W. Smith & Co., Ltd.)



MULTI-FLOW VERTICAL JUICE HEATER.
A. & W. Smith & Co., Ltd.)

For the accumulation of the lime-milk which will settle from the hydrated lime, *lime-milk stirrers* are required and the storage capacity should be sufficient at least for a two-hour run of the factory, to allow for regulating the work of the lime kilns or for some minor repairs at the lime station.

The stirrer is a cylindrical tank, having a spherical or conical bottom with a vertical shaft of sufficient strength, carrying the stirrer blades and revolving at about 8 r.p.m. by means of a set of bevel gears, as shown in *Fig.* 369.

The stirrer blades are made of 3 in. $\times$ $\frac{7}{8}$ in. flat iron, set at right angles. In addition, there are two retainers made of flat iron, 3 in. $\times$ $\frac{1}{2}$ in., also set at right angles, free from the shaft and bolted to the tank shell. The milk-of-lime is discharged at the bottom and the lower shaft bearing is so arranged that the lime cannot settle out in it. Brass must not be used in this lower bearing, but a cast iron bushing.

For scraping the bottom, a chain is attached to the lower stirrer blade, which will drag over the bottom plate.

Lime-milk pumps are of the plunger type, having cast iron plungers and linings and spherical or ball valves, also of cast iron. The pumps are generally belt or electrically driven through a reduction gear. Centrifugal pumps, having open impellers of cast iron or aluminium, are also used. Care has to be taken, that the stuffing boxes of the shaft are provided with fresh water connexions, as well as the pump body, the latter for washing out, and to remove settled or adhering lime.

4.—The Sulphur Station.

Sulphitation and carbonatation factories require a sulphur oven or *sulphur furnace* and the general design, as made by several manufacturers, is shown in *Fig.* 370. The sulphur is charged into the filling cylinder c, which is closed by an asbestos-sealed cover. By revolving the cast iron flap at the bottom of the cylinder, the charge is dumped into the cast iron tray *g*.

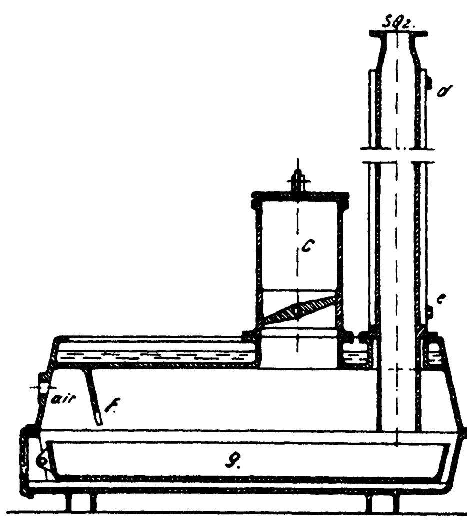


Fig. 370.—Sulphur Furnace.

Compressed air of 10 to 15 lbs. per sq. in. is admitted at the front end and is distributed by a cast iron baffle *f* over the surface of the sulphur in the tray *g*, which has been previously ignited by a piece of red hot iron, dumped into the tray through a peep hole having a threaded plug. A sight glass with a sheet of mica is sometimes provided, although the mica may be blown out by the pressure inside the furnace and must be replaced by an iron sheet, so the sight glass is not quite necessary, when a good plug hole is provided, which is generally arranged at the side of the furnace.

The vertical discharge pipe for the SO_2 gases is surrounded by a water jacket, the cooling water entering at *e* and being discharged at *d* into the main cooling jacket on top of the furnace.

The tray *g* can be removed through the front door of the furnace. All joints in the oven must be made from asbestos without rubber, as the high temperature will not allow the use of the latter material.

In this kind of furnace only pure brimstone can be burned, as unrefined sulphur from the mines will form a slag, which prevents combustion of the sulphur underneath.

The compressed air is supplied by an air compressor and the moisture in it has to be removed by an air dryer, this being a casing of wood, brick or steel provided with a double set of trays with quicklime, so that the air current can be led over one set, when the other set is cleaned and the lime renovated. For each 100 cub. ft. of aspirated air about one lb. of CaO is required. A 75 per cent. saturation of the air is assumed, but in sugar factories the dampness may prove higher, and for practical purposes two lbs. of CaO per 100 cub. ft. of aspirated air will be sufficient. The trays have to be of such a size that the lime needs to be renewed about every six hours.

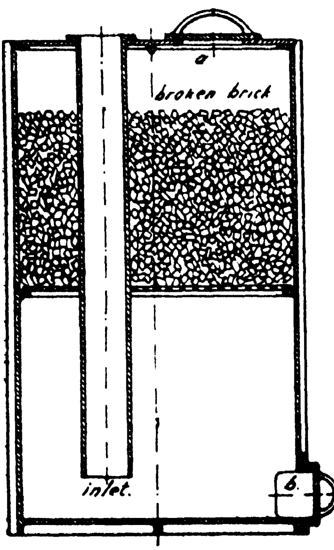


Fig. 371—SO_2 Scrubber.

One pound of sulphur requires about 60 cub. ft. of air at 85°F. for complete combustion, and as excess air has to be furnished, 100 cub. ft. of aspirated air per lb. of sulphur to be burnt will be sufficient for practical performance.

Further cooling and cleaning of the gases is achieved in the *SO_2 Scrubber*, as shown in *Fig.* 371, this having two compartments, the upper one being filled with broken brick of about 2 in. sizes. The SO_2 enters in the lower compartment and if any sublimation has taken place, it may precipitate here. The brick is piled upon a cast iron plate made in two halves and having large perforations. A cleaning door is provided for the lower compartment at *b* and for the upper one at *a*.

The general arrangement of a *Sulphur House* for burning about 1200 lbs. of sulphur per 24 hours, as designed and installed by the author, is shown in *Fig.* 372. The building is of wood, outside of the main factory building, to prevent the penetrating odour of the sulphurous gases from contaminating the factory atmosphere. A good size ventilation cap is provided and a storage place for the sulphur, which is delivered in barrels or sacks, has been included.

In front of the sulphur furnace *a* are laid a couple of rails on a special foundation, so as to support the sulphur tray when taken out from the furnace. The gas scrubber *b* with connexions both for inlet and outlet, is also shown, and also the lead-lined SO_2 valve, having a rubber diaphragm. Cast iron pipe lines about $\frac{3}{8}$ in. thick are used for the SO_2 gases, having removable covers for cleaning the pipe-elbows. Steel should not be used for this purpose as it will corrode rapidly.

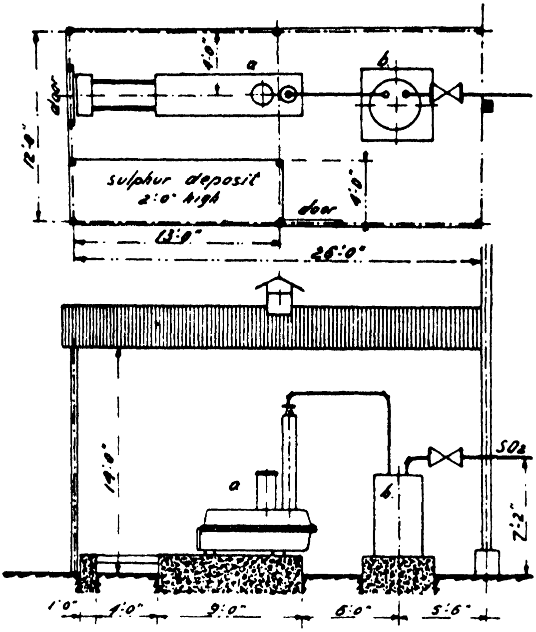


Fig. 372.—General Arrangement of a Sulphur House.

Where unrefined sulphur has to be used, which is available at a reduced price, a *Rotary Sulphur Furnace*, as shown in *Fig*. 373, can be used to advantage.

In the casing are two serrated discs, revolving in opposite directions, which enter into the molten sulphur and thus break the layer of slag which may have formed. The furnace has a charging and a cleaning door, the gas exit being cooled by a water-jacket.

Air is admitted bilaterally and this kind of furnace can be used in combination with a sulphur tower, having natural draught. The sulphur furnaces under pressure are connected to sulphitation tanks instead of to a tower, which point will be referred to further on.

Revolving drum type furnaces are also used for burning sulphur, similar to rotary lime kilns.

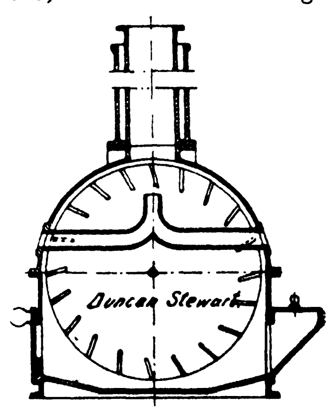


Fig. 373.—Rotary Sulphur Furnace.

CHAPTER XVIII.

JUICE HEATERS.

FUNDAMENTALS — DESIGNS IN USE.

To coagulate the albuminous or colloidal impurities and to induce separation, flocculence or precipitation of the insoluble impurities in the settling tanks, the juice has to be heated to a temperature of about 212°F. (100°C.).

This heating performance is carried out completely, or in some instances partially, in juice heaters, with which we are dealing in this Chapter.

1.—Fundamentals of Juice Heating.

A uniform heating performance is required of the juice heaters, and is secured by installing suitable heating elements. Periodical cleaning of the heating surfaces is essential, as it will be impossible to avoid incrustations. On the inside of the tubes, where the juice circulates, incrustations of silicates, sulphites, phosphates, lime compounds or cane fibre will adhere, thus greatly impairing the heat transmission. On the outside of the heater tubes, the lubricating oil from the cylinders of the reciprocating steam engines, carried along with the exhaust steam, will form a thin but insulating scale, resulting in the same inconvenience.

These incrustations have to be removed and the inside ones, being the most important, are generally dissolved by a 1 to 2 per cent. solution of caustic soda, circulated through the heater tubes and sometimes afterwards boiled out with a 1 to 2 per cent. solution of hydrochloric (muriatic) acid. The latter task must be done without circulation, to prevent the acid solution from corroding the steel juice lines. There is a divergence of opinion amongst different technologists as to the value of either or both of these treatments, and mechanical removal by means of scrapers is undertaken in those instances where the chemical reagents do not yield the desired results.

The scraping is done by means of steel brushes or tube cleaners, but these will roughen the inside tube surface, when not used with great care, and moreover the tube material, brass or copper, is easily scraped off. A rough tube will produce a quicker adherence of scale than will a polished one.

The author has used round steel bars of about $\frac{1}{16}$ in. less diameter than the inside of the tube with good results. The top part of the tube has to be cleaned with a small rod and a light hammer before the large rod, as long as the tubes, can be inserted.

Another way is to heat the apparatus dry, when owing to the differing expansions of the tube material and the scale, the latter will break off. This heating has to be done carefully, so as not to cause leaks at the junction with the tube plates.

For cleaning the outside of the tubes, the heater is filled with water, on top of which is poured a few inches of kerosine—gasoline is very dangerous—and the water slowly released by a dripping valve so that the kerosine layer will gradually sink in the course of a few weeks and thus dissolve the oily scale. Afterwards, the inside as well as the outside of the tubes should be thoroughly washed out with clean water.

The inside cleaning is generally done once a week during crop time, whereas cleaning the outside is only required once in the dead season.

A nearly constant juice temperature can be maintained, as shown from an actual *Recording Thermometer Chart* depicted in *Fig.* 374, being the average performance in a well-operated Cuban sugar factory.

The amount of heat to be supplied for heating the juice depends on its *specific heat*, which varies with the concentration or density. Any solution will have a specific heat coefficient, equal to the sum of the specific heat coefficients of its components. A sugar solution—as juice is assumed to be—has two major components, one being water having a specific heat coefficient = 1,

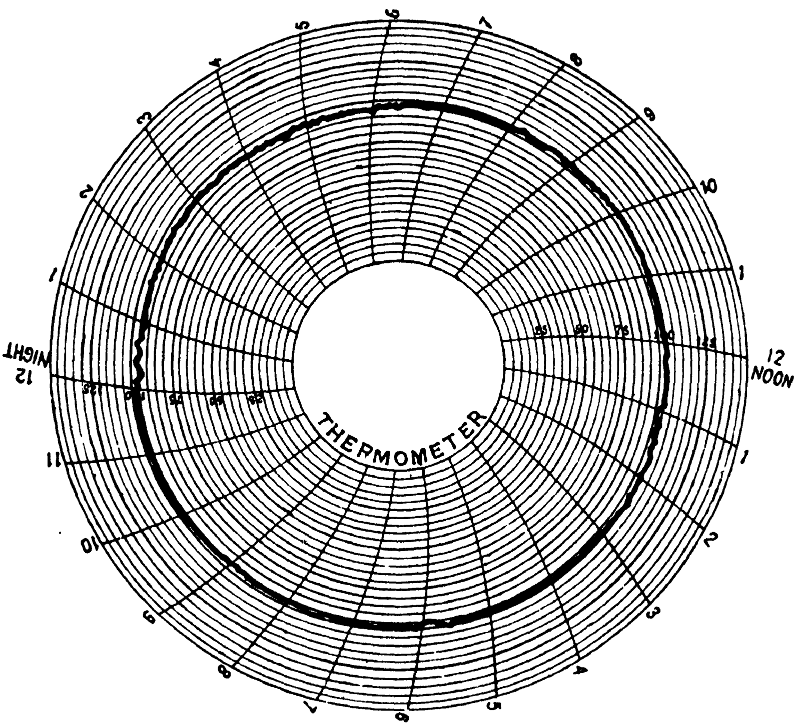


Fig. 374.—Recording Thermometer Chart for Juice Heating.

the other being sucrose, having a specific heat = 0·301. It will be obvious that juices of a higher Brix will have a lower specific heat, according to the general formula :—

$$C_j = C_w \times x + (C_s \times (1 - x)) \ldots\ldots\ldots\ldots\ldots\ldots \quad (95)$$

in which C_j = specific heat of juice in B.Th.U. per 1°F. temperature rise (or in cal. per 1°C.)

C_w = specific heat of water = 1.

C_s = specific heat of sucrose = 0·301.

x = percentage of water in the juice.

For diluted juice, assumed to be of 15° Brix, the specific heat thus will be :—

$$1 \times 0{\cdot}85 + 0{\cdot}301 \times 0{\cdot}15 = 0{\cdot}89515,$$

whereas syrup of 60° Brix has a specific heat of 0·5806 and one lb. of juice or of syrup will thus require respectively 0·89 and 0·58 B.Th.U. for each degree

Fahr. rise in temperature (or 0·89 and 0·58 cal. respectively for each °C. rise in temperature).

There are other ingredients in solution in the cane juice, such as glucose, salts, etc.; but the percentage is very low and the specific heat is only very slightly affected, so may be neglected for practical calculations (see diagram, *Fig.* 375, the dotted lines according to KOPP).

The heating of the juice is accomplished by exhaust steam or vapours from a pre-evaporator; combustion gases from the boilers are not used for this purpose. The latent heat only of the steam is transferred to the juice, the sensible or liquid heat leaving with the hot condensate. In practice, the condensate has generally a temperature slightly below the saturation temperature of the steam or vapours employed, especially in the first heaters of a series, which receive cold juice.

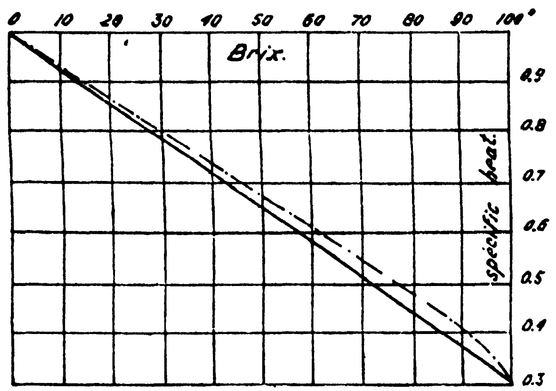


Fig. 375.—Specific Heat Diagram of Juice.

There is, moreover, a heat loss through radiation, depending upon the insulation of the juice heaters and the hot juice lines. The author has found that this loss fluctuates between 5 and 10 per cent. of the steam heat supplied.

The following calculation may thus be established :—

Total heat in one lb. of exhaust steam at 7 lbs.		1157·8 B.Th.U.
Heat in condensate	200·6 B.Th.U.	
Radiation loss	57·2 B.Th.U.	257·8 B.Th.U.
Available for heating		900·0 B.Th.U.

Specific heat of juice of 15° Brix .... 0·895
Temperature of juice on admission to heaters .. 82°F.
Temperature of juice discharged from heaters .. 212°F.
Temperature rise of juice .... 130°F.
Heat required per lb. of juice : 130 × 0·895 = 116 B.Th.U.

One lb. of dry exhaust steam will thus heat 7·75 lbs. of juice of the above-mentioned composition. To know the percentage of the total heat available in the bagasse fuel to be used for heating juice, the calculation for any given factory is as follows :—

1 ton of cane = 2240 lbs.
96 per cent. diluted extraction = 0·96 × 2240 = 2150 lbs.
22·3 per cent. bagasse on cane = 0·223 × 2240 = 500 lbs.
Average heat value of bagasse as fired ...... 3800 B.Th.U./lb.
Heat available in bagasse : 500 × 3800 = 1,900,000 B.Th.U.
Specific heat of juice .... 0·895
Temperature rise .... 130°F.
Heat required for juice heating : 2150 × 116 = 249,400 B.Th.U.
Percentage of total heat in bagasse required for heating juice : 13·1 per cent.

The percentage of factory "steam" required for heating depends upon the way it is used, viz., in simple or double or even triple effect. Where steam economy has to be considered, the latter scheme, i.e., using vapours of a pre-evaporator or second body, is the most advantageous.

The amount of heating surface depends upon the *heat transmission*, thus on the property to transfer the available heat in the steam to the juice. The theory of heat transmission is a very complex one, as so many factors are involved, e.g. :—

1 Heat transmission through a film of steam around the tube.
2 ,, ,, ,, ,, ,, condensate around the tube.
3 ,, ,, ,, the oil scale on the outside of the tube.
4 ,, ,, ,, the metal of the tube wall.
5 ,, ,, ,, a film of juice inside the tube.

These factors are of varying nature and are moreover aggravated by the following :—

6 Velocity of juice through the tubes.
7 ,, ,, steam around the tubes.
8 Presence of insulating air in the steam space.
9 Diameter and thickness of the tube.
10 Material of the tube : steel, brass or copper.
11 Mean temperature difference between the steam and the juice.

The heat transmission coefficient is generally given in B.Th.U. per sq. ft. heating surface per hour per degree F. rise in temperature or with the metrical system in cal./m²/hr./1°C. (when not otherwise stated kg.cal. are meant).

From practical tests carried out by different authors an average heat transmission of :—

$$K = 120 \text{ B.Th.U./sq. ft./hr./1°F.}$$
$$= 600 \text{ cal./m}^2\text{/hr./1°C.}$$

has been found.

The coefficient of heat transmission K is sometimes called the factor of thermal conductivity, and the author has found for Cuban conditions, that it varies between 100 and 170 B.Th.U./sq. ft./hr./1°F.

For conversion to metric units the following equation serves :—

$$1 \text{ cal./m}^2\text{/hr./1°C.} = 0{\cdot}205 \text{ B.Th.U./sq. ft./hr./1°F.}$$

In the beet sugar industry a new type of heater has been tried out, having a flat, hollow, heating spiral, made of 4 mm. plate, similar to the Gräntzdörffer bodies[1] for vacuum pans, with which a thermal conductivity of 480-640 B.Th.U./sq. ft. hr./1°F., thus approximately 500 per cent. of the accepted average for tubular heaters, has been reported.[2]

The temperature difference between the heating steam and the heated juice is not the arithmetical average of entrance and exit temperature differences; this *mean temperature difference* according to GRASHOF[3] amounts to :—

$$\Delta_m = \frac{\Delta_i - \Delta_o}{\log_e \dfrac{\Delta_i}{\Delta_o}} \quad \dots\dots\dots\dots\dots\dots \quad (96)$$

---

1 See *Fig.* 472.

2 See abstract, *Int. Sugar Jl.*, July, 1935, p. 276; and E. W. KERR and S. J. WEBRE, Bulletin 159 of Louisiana State University.

3 See E. HAUSBRANDT, "Verdampfen, Kondensieren und Kuhlen," 1918 Ed., p. 4.

where : Δ_m = mean temperature difference.
Δ_i = temperature difference of incoming juice and steam.
Δ_o = „ „ outgoing „ „
log_e = natural logarithm = $2{\cdot}3026\ log_{10}$.

For average conditions with the following properties, the mean temperature difference will amount to :—

Exhaust steam 7 lbs. gauge pressure	232°F.
Incoming juice	82°F.
Outgoing juice	212°F.

$\Delta_i = 232 - 82 = 250°F.$
$\Delta_o = 232 - 212 = 20°F.$
Δ_m = mean temperature difference.

$$\Delta_m = \frac{250 - 20}{log_e \frac{250}{20}} = 92°F.$$

The arithmetic average $(250 - 20) \div 2 = 115°F.$, thus showing a difference as compared with 92°F.

The heating surface of a heater can now be easily calculated from the following equation :—

$$Q_j \times C_j \times T = A \times K \times \Delta_m \quad \ldots\ldots\ldots\ldots\ldots \quad (97)$$

Since this formula serves equally for the British system as for the metric one the symbols signify :—

Q_j = hourly quantity of juice in lbs. or kg.
C_j = specific heat of the juice.
T = rise in temperature of the juice in °F. or °C.
A = heating surface of the heater in sq. ft. or m^2.
K = coefficient of heat transmission.
Δ_m = mean temperature difference.

From the previous assumption, $Q_j \times C_j \times T$ per ton of cane ground per hour amounts to 249,000 B.Th.U. and with K = 100 B.Th.U./sq. ft./hr./1°F. and $\Delta_m = 92$, the heating surface A amounts to 27 sq. ft.

Thus, for each ton of cane ground per hour, about 27 sq. ft. of heating surface is required. As there is generally one heater required for cleaning in the series, 40 sq. ft. per ton of cane/hr. is accepted standard, although the author knows many instances where only 30 sq. ft. has been applied with good results. But it should be borne in mind that, wherever possible, the heating surface required should be divided over three heaters, so that two-thirds of the heating surface remains in operation, while the remaining third is being cleaned. Where a weekly stoppage of from 6 to 12 hours is customary, this provision need not be made.

In respect to the factors influencing the thermal conductivity, the following can be said (the numbering remains as on page 359).

2.—The condensate has to be removed as soon as produced, the outlets being dimensioned for a maximum flow of 3 ft./sec.

3.—The exhaust steam line from reciprocating steam engines—not essentially required with turbines—should be provided with an oil and condensate separator.

4.—Copper tubes have the highest conductivity and are best, when muriatic acid is used for boiling out the heaters. Brass tubes are cheaper and harder, and are extensively used in Java, ranking second in thermal conductivity. Steel tubes should be avoided, as they corrode quickly and have greater adherence for incrustations, the heat transmission being also the lowest.

5.—Heater tubes sometimes corrode on the outside, thus in the steam space close to the tube plates. As there are no ammonia gases in the exhaust steam or vapours, oxidation or galvanic action may be the cause.

6.—The velocity of the juice has a bearing upon the incrustation as well as on the heat transmission, and for juices carrying a considerable amount of insolubles, as with the Petree compound clarification, the velocity of the juice should be kept high, above 6 ft. per second. Normal velocities are about 3 ft. to 5 ft. per second.

7.—Baffles in the steam space are used to advantage in horizontal heaters. In vertical heaters they are generally not applied, when care is taken that the coldest juice is led through the centre passes.

8.—One lb. of air at 232°F. occupies a space of approximately 17·3 cub. ft., whereas exhaust steam of the same temperature and equivalent pressure occupies approximately 18·6 cub. ft., the air thus being heavier; and as the separation in the heater proceeds slowly, most of the air has to be vented at the bottom, whereas lighter incondensible gases may accumulate at the top, so a vent should be provided there also. For heaters where vapours below atmospheric pressure are used, the vents should be connected to the consecutive body of the triple or quadruple effect from which the vapours are drawn.

9.—The tube diameters for practical reasons connected with incrustation and cleaning are from $1\frac{1}{2}$ to 2 in. external diameter, $1\frac{3}{4}$ in. being mostly used.

The heater bodies should be tested by about 45 lbs. hydrostatic pressure on the steam side, whereas the juice side should be subjected to a pressure of 75 lbs. per sq. in.

The juice lines to the heaters should not carry a velocity of flow over 6 ft./sec. and vapour and steam connexions not over 150 ft. per sec.

The juice pressure required for passing the heaters depends on the number of circulations, as it will be obvious that 18 passes at an obviously higher juice velocity will cause more resistance or friction than, e.g., 8 passes at a reduced velocity. This juice pressure according to the author's observations varies between 15 and 55 lbs. per sq. in., and parallel switching will necessitate a lower pressure than connexion in series. A manometer is an indispensable accessory at the juice pump, as any rise in pressure will indicate excessive incrustation of the heater tubes.

2.—Construction of Juice Heaters.

There are two principal types, the Vertical and the Horizontal, the multitubular type being extensively used, whereas the coil heater has largely disappeared. The vertical heater is subject to less formation of scale and occupies a reduced floor space as compared with the horizontal heater. In practical operation both give good results.

The coil type heater, having a spirally wound copper coil, through which the juice flows inside a steam drum, has the disadvantage of being difficult to clean from scale, a process which is only possible by using strong reagents. The author has seen these heaters over 40 ft in length, the steam pressure being

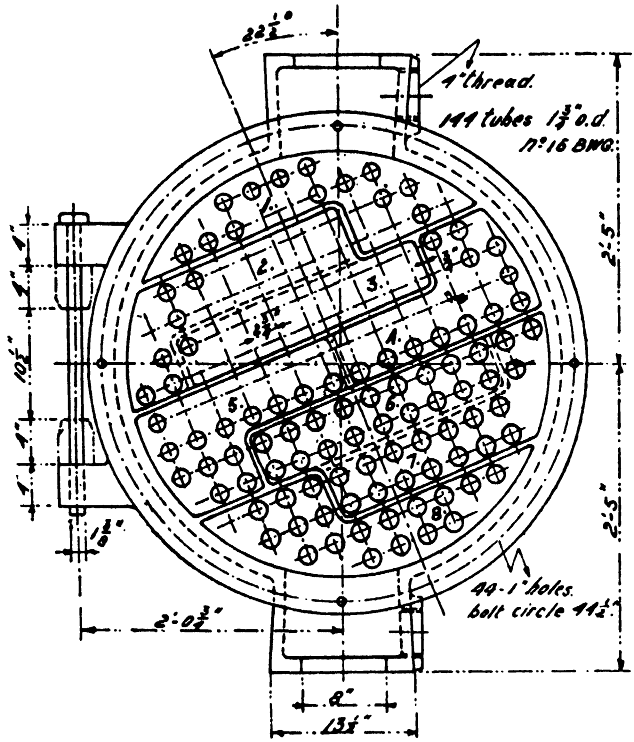


Fig. 376.—Header of a Horizontal Juice Heater.

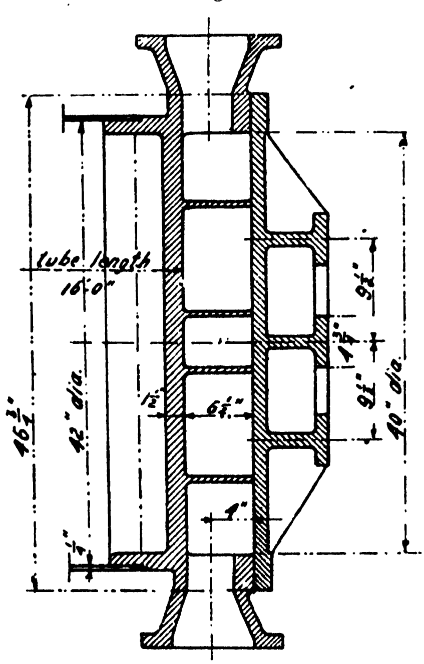


Fig. 377.—Cross Section of Above Header.

about 15 lbs. per sq. in. at one end and zero at the other end, this indicating counterflow of the juice and thus favourable heat transmission (see the Chapter on *Crystallizers*, where parallel and counterflow principles are explained.)

In *Fig.* 376 is shown the front view of the *Header of a Horizontal Juice Heater* as measured by the author, having 1000 sq. ft. H.S. and 8 passes for the juice. To achieve a counterflow performance, the steam enters at the front top end, whereas a baffle leads the flow to the rear and the condensate is withdrawn at the front again. The juice enters at the coldest spot of the heater, at the bottom, i.e., in space "8."

Inlet and outlet cocks have 4 in. nozzles for caustic soda and washout connexions.

In *Fig.* 377 is shown the cross-section of this same header and it will be seen that the tube sheet is integrally cast with the header, as is now general practice. The header or division head, as well as the cover, are made of cast iron, the latter having a hinged connexion, to swing it aside when the cover bolts have been removed. A rubber or lead plate serves to make a tight joint between the division walls and the cover.

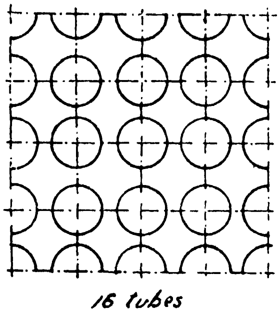


Fig. 378.—Square Spacing.

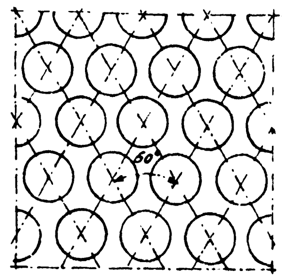


Fig. 379.—Diamond Spacing.

The *Tube Spacing* can be done in squares as shown in *Fig.* 378 or in rhombus or diamonds, as shown in *Fig.* 379. The first spacing requires more area to the same number of tubes, it having parallel steam paths, whereas with the rhombic spacing, staggered steam paths are obtained, which will increase the heat transmission.

The space ratio will be as the area of a square to the area of a 60° rhombus, both having the tube pitch as sides, or as $1 \div \frac{1}{2}\sqrt{3} = 1 \div 0{\cdot}866$, the rhombic spacing thus having 16 per cent. more tubes in the same area covered. *Figs.* 378 and 379 are drawn to the same scale, having the same tube diameter and pitch.

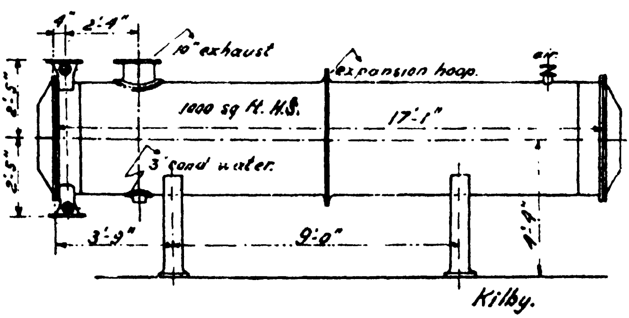


Fig. 380.—Arrangement of a Horizontal Heater of 1000 sq. ft.

In *Fig.* 380 is shown the arrangement of a *Horizontal Tubular Heater* of 1000 sq. ft., having the headers shown in *Figs.* 376 and 377. The heater has a 10 in. exhaust steam inlet and a 3 in. condensate discharge at the bottom. A baffle is laid horizontally in the middle of the tubes, so as to force the steam along a two-fold circulation. The counterflow principle is thus adhered to, the cold juice entering at the bottom end, so favourable heat transmission is obtained.

Air is withdrawn at the top end, but it is always preferable to do so from both top and bottom.

As the tube material has a coefficient of expansion different to that of the shell, the latter is provided with an expansion hoop arranged at the middle of the shell. There are now also heaters on the market, having the tube ends on one side expanded in the fixed header and the other ends fitted in a loose header, inside the heater shell, so that they may freely expand with the tubes.

The temperature range inside the heater will lie between the cold juice and the steam or vapour temperature, and the first passes are less subject to expansion than the last ones ; for normal exhaust steam conditions, the difference in linear expansion will not reach 0·02 in. for each 10 ft. tube length, so arrangements for taking up this expansion difference are generally omitted.

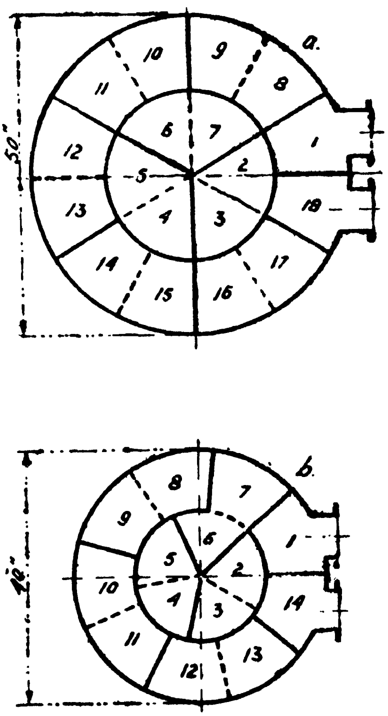


Fig. 381.—Heaters.

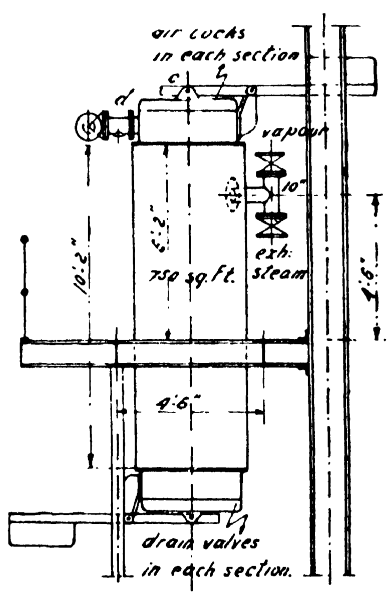


Fig. 382.—Vertical Heater.

The author has designed heaters having a 14-fold circulation as shown in *b* of *Fig.* 381 for 750 sq. ft. H.S. and 18-fold circulation as shown in *a* of the same Fig. for 1130 sq. ft. H.S. The maximum known to the author is 30-fold circulation per heater ; in Java a 6-fold circulation has been applied in more recent installations.

In the course of operating heaters, the author has noted in case of three units in series (each having 14-fold circulation, 1¾ in. outside tube diameter and 10 ft. 2 in. tube length, with a velocity of flow of approximately 4 ft. 6 in. per second) that the required pressure lies between 45 and 55 lbs. per sq. in., of which 15 lbs. has to be deducted for hydrostatic head, due to the difference in level between the pump intake and the juice discharge plus pipe friction.

The arrangement of a *Vertical Heater* of 750 sq. ft. H.S., as furnished by the author, is shown in *Fig.* 382. The covers are balanced by counterweights for easy manipulation, and are tightened by hinged bolts of sufficient strength to take up the prevailing juice pressure. Between the juice valve and the upper header a branch piece *d* is inserted, having 2 in. connexions for caustic soda or wash water.

The top cover is provided with small air cocks in each passage or section, for expelling air when the heater is started. The bottom cover is provided with ¾ in. cocks in each pass for draining the heater. As contrasted with the horizontal heater, a vertical one will remain filled with juice, caustic soda or wash water.

A 10 in. T-piece is attached to the shell, so arranged that the counterweight of the top cover will keep clear of it, and serves for vapour and exhaust steam connexions. Sometimes a 2 in. connexion for live steam is also provided, for boiling the heater during cleaning operations when exhaust steam is not available.

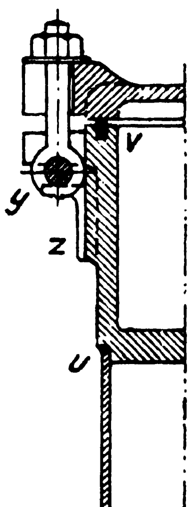


Fig. 383—Integrally Cast Header and Tube Sheet.

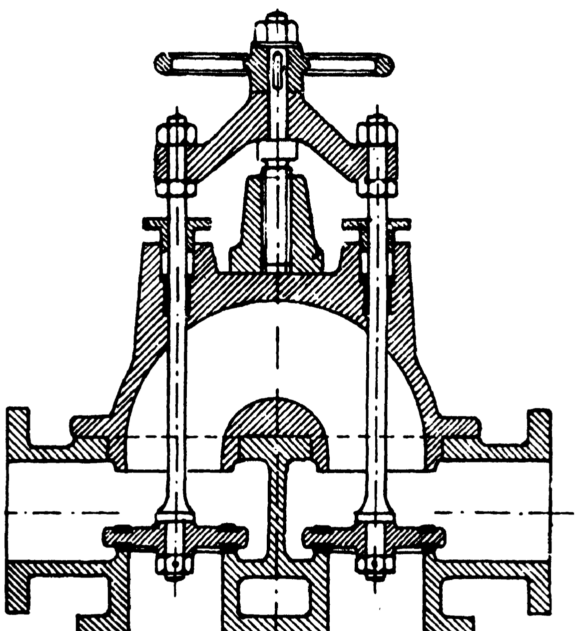


Fig. 384.—Double Juice Heater Valve.

It should be recollected that exhaust steam of, e.g., 7 lbs. per sq. in. cannot be applied simultaneously with vapours of, e.g., atmospheric pressure. The author knows instances where this has been attempted, but the result has been that a pressure in the first body, which supplied the vapour, has been built up to 3·4 lbs. per sq. in. and as this body was heated with the same exhaust steam of 7 lbs., its evaporating capacity was greatly impaired.

The heater has cast iron division heads and brass tube plates, the tubes being of copper. These division heads will readily leak at the joints between them and the tube plates, and a header integrally cast with the tube plate is therefore to be preferred. Moreover, the author has known instances where tube plates had been deflected inwards and a stay tube has had to be provided in the centre of the shell, between the two plates.

An *Integrally Cast Header and Tube Sheet* of cast steel is shown in *Fig.* 383. The heater shell is welded to the header at *u*, thus making an economic construction, which is now employed by several manufacturers. The cover joint is made by means of rubber strips *v* laid in the corresponding grooves of the header. The hinged cover bolts are attached on a ring-shaped round iron bar *y* (as designed by the author) which has been laid underneath the header flange, the latter being provided with slots at the position of the bolts. This ring *y* is held by a few angle braces *z*, bolted to the header.

A *Double Juice Heater Valve* of compact design is shown in *Fig.* 384. As the three valves—see *Fig.* 385 (*c*)—might be wrongly operated, in which case the juice line would get locked, with a corresponding danger from increased pressure, this double valve is a useful piece of equipment. The two valve discs have inserted white metal or hard lead seats and are attached to brass valve spindles. The two spindles are operated by a *yoke* and it is easy to ascertain when the valve is open or closed.

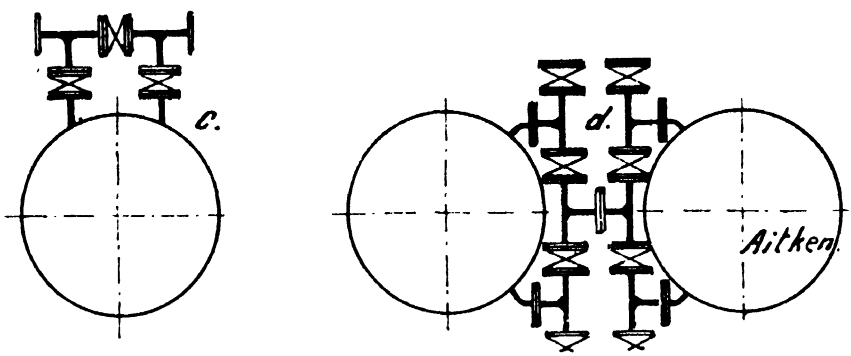


Fig. 385.—Juice Line Connexions.

For sulphitation factories, raw juice is passed through heaters, which have been operated previously with sulphitated juice, the sulphites thus being dissolved and the heater cleaned. A double set of the valves as shown in *Fig.* 384 is then required, one on the sulphitated and the other on the raw juice line.

The *Juice Line Connexions* are shown in *Fig.* 385. At *c* the widely used three-valve arrangement is shown, and to cut out a heater, the centre valve has to be opened, before the two valves next to the header are closed. At *d* a four-valve arrangement is shown for two heaters, which can be operated in parallel or in series.

Thermometers should be provided in the juice line before and after each heater, so that the heating performance of each unit can be judged.

Thermostatic control of an air or electric-operated type is nowadays employed in factories with varying exhaust steam pressures. A drop in the hot juice temperature will automatically cause the steam valve to open further, and thus the juice temperature will be kept within very close limits of a few degrees F.

CHAPTER XIX.

CLARIFICATION EQUIPMENT.

LIME MIXERS — DEFECATORS — SUBSIDERS — SULPHITATORS — CARBONATATORS.

The heated mixture of juice and lime-milk, in which the chemical and coagulating performance has taken place, is passed into defecators and/or settling tanks or subsiders to separate the insoluble matter from the clear juice.

1.—Principles and Nature of Defecation.

Defecation, decantation, subsiding or settling mean the process in which the insoluble solids in the juice are separated by force of gravity. It will be obvious that such a performance is only feasible, when there is a definite difference in specific weight of the materials to be separated, and after decantation is complete there will remain three zones, viz. :—

1.—The scum, floating on top of the juice.
2.—The clear juice itself.
3.—The mud beneath the clear juice zone.

Any true gravity performance is not to be expected, as the settling does not take place under vacuum, but in a resistant medium, and is to be compared with the fall of chips of paper, which will reach lower levels in a whirl. Any air current wafting in a different direction to that of the falling chips will divert or retard the falling action. Moreover, the air resistance and the size of the chips will affect the time of falling.

With juice decantation similar phenomena can be observed, the size of the settling particles and the juice resistance generally being of unascertained magnitude. Juice currents are produced by the cooling action of the tank walls setting up a difference of temperature between the juice layers close to the walls and the more central ones. It is, therefore, advisable to insulate the subsiding tanks with asbestos magnesia, hair-felt or similar insulating materials, which will reduce the prevailing temperature difference and also avoid unnecessary heat losses. The scum forms a good protection against evaporation and the corresponding cooling caused by it, but covered subsiders are used to advantage in many instances.

In non-insulated open subsider tanks the juice with a charging temperature of about 212°F. (100°C.) will have cooled down to approximately 185°F. (85°C.), by the time the decanting performance has been completed, or still lower in well-ventilated sugar houses, whereas covered and insulated subsiders have a discharge temperature of about 200°F. (93°C.).

No difference has been found by the author in the time of settling when using round or rectangular tanks. The former will occupy more space, whereas it is sometimes assumed that rectangular tanks will show juice currents that retard the decantation ; but the latter feature is seemingly of no practical importance.

The decanting force obviously amounts to $S_1 - S_2$, where S_1 is the specific gravity of the materials to be separated and S_2 that of the juice. As the difference in specific weight is not great and the size of the particles to be decanted may offer an area well above microscopic dimensions (especially when fine fibre or bagacillo is present in the juice) the decanting proceeds very slowly, as it is hampered by the viscosity or the presence of colloidal matter in the juice.

It is a well-known fact nowadays that a thin layer of liquid will produce a larger quantity of clear juice in the same time period than a thicker one, and use is made of this phenomenon in several continuous juice clarifiers, as well as in other industries.

Common settling tanks are about 6 ft. in height and the complete cycle for defecation will last from 1½ to 2 hours; but juices derived from different cane varieties may require additional reagents like phosphoric acid (lime phosphate with about 50 per cent. P_2O_5) in order to settle properly.

Means have been sought to increase the decanting force or to diminish the resistance of the juice, the latter being achieved by employing a temperature close to 212°F., and for the former centrifugal force has been applied. This last amounts to :—

$$C = \frac{M \times V^2}{R}$$

with M = the mass of the centrifugalled material = weight ÷ gravity acceleration ($W \div 32{\cdot}16$).
V = speed in ft./sec. of the material $= 2\pi \times R \times N \div 60$.
R = centrifugal radius in ft.

Instead of the weight, the difference in the specific gravities $S_1 - S_2$ may also be employed :—

$$C = \frac{S_1 - S_2}{32{\cdot}16} \times \frac{4\pi^2 \times R \times N^2}{3600} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (98)$$

For a centrifugal having an average radius of 1 ft. and revolving at N = 1000 r.p.m., the centrifugal force, according to formula (98), amounts to 293 times $S_1 - S_2$. Centrifugal separation has been attempted in South Africa, but the author has no information as to the working results. More recently, centrifugals with blind baskets, charged at the bottom and overflowing over the top lip, have been used for decanting impurities in syrups and molasses with good results, and this type of centrifugal might be of value also for decanting juice, a special device being added for removing scums.

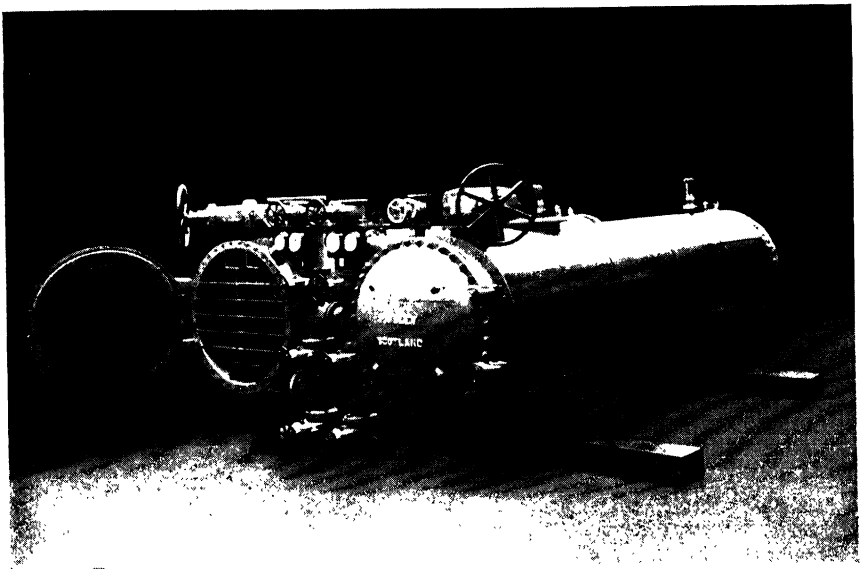
The total capacity of subsider tanks for a raw sugar cane factory can be calculated from the following data :—

t_a = Time for filling — 15 minutes assumed.
t_b = Time for decanting — 90 minutes.
t_c = Time for discharge of the mud and cleaning — 15 minutes.

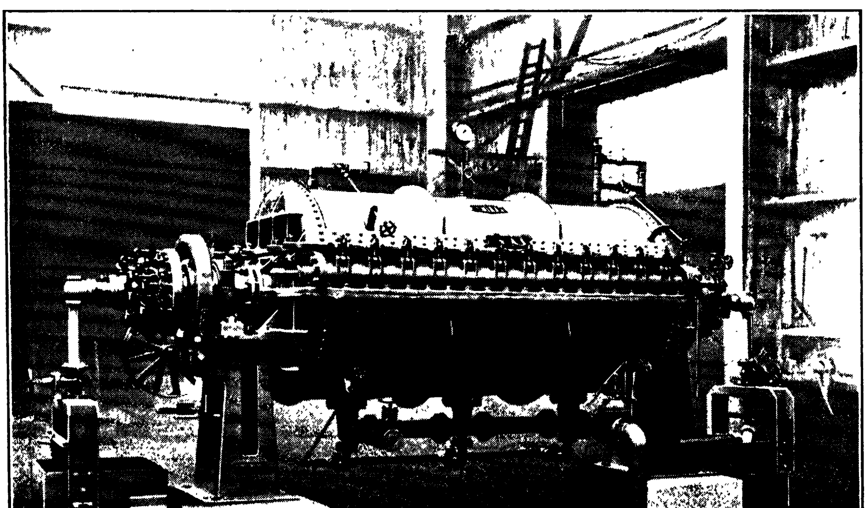
The capacity of each tank should be such that it can be filled in about 15 minutes under normal grinding conditions, and sizes from 120 to 1000 cub. ft. are used. The total cycle thus amounts to 2 hours, and as a ton of cane will produce 2150 lbs. of juice (as assumed in the previous Chapter) having a specific weight of 1·07 on an average or 67 lbs. per cub. ft., 32 cub. ft. of juice are produced per ton of cane. Allowing 25 per cent. surplus for emergencies, the total capacity in subsiders is thus derived from :—

$$x = \frac{1{\cdot}25 \times 32 \times Q \times T}{24} = \frac{5\,Q \times T}{3} \quad \ldots\ldots\ldots\ldots \quad (99)$$

where x = total subsider capacity in cub. ft.
Q = grinding capacity in long tons of cane per 24 hours.
T = time for a complete cycle of each tank, in hours.

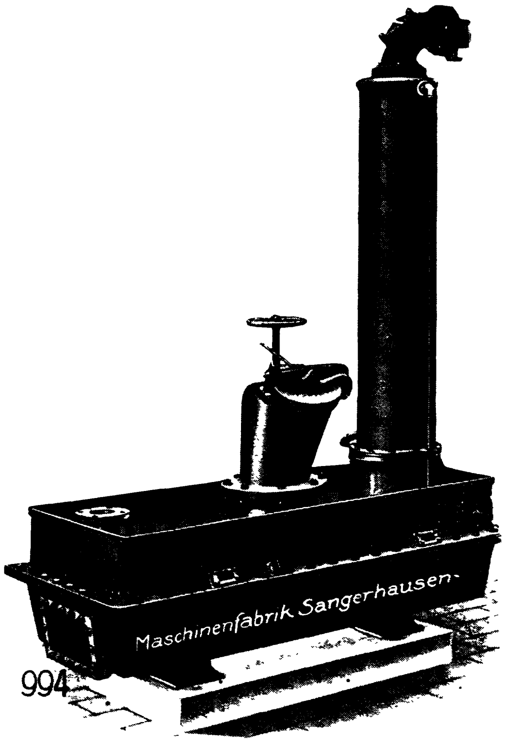


MULTI-FLOW HORIZONTAL JUICE HEATERS.
(H. W. Aitken Co., Ltd.)

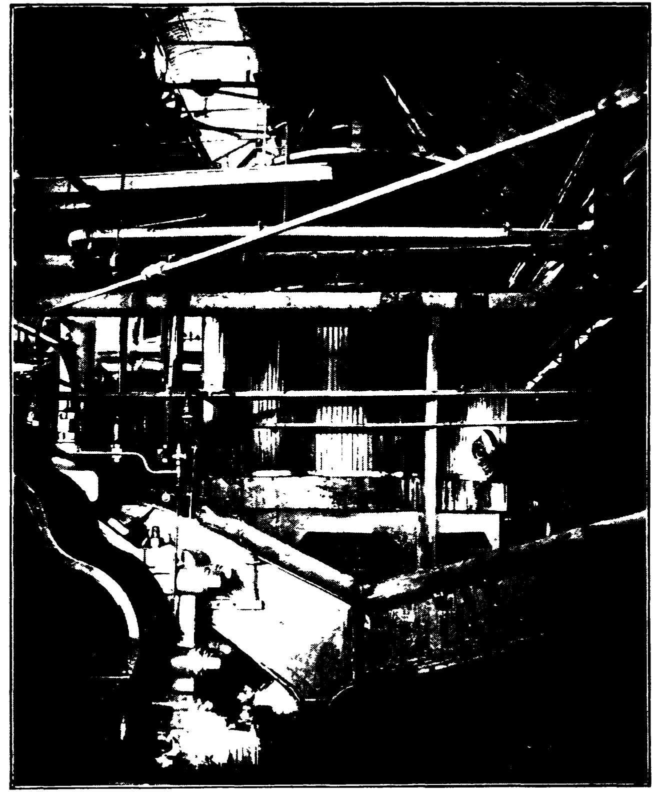


SUCHAR AUTO-FILTER.
(A. & W. Smith & Co., Ltd.)

PLATES 69 & 70.



SULPHUR FURNACE WITH COOLING JACKETS.
(Maschinenfabrik Sangerhausen)



CONTINUOUS SETTLING TANK.
(Petree & Dorr Engineers, Inc.)

Per ton of cane ground per 24 hours, 3 to 4 cub. ft. are thus required, the larger capacity for those factories which return the settling mud to the intermediate carriers between the mills, this needing a longer settling cycle.

2.—Lime Mixers.

Fully automatic mixing of the lime-milk with the juice is not very practicable, as the acidity of the juice is not a constant figure and variations in liming have thus to be performed by the laboratory staff, so the equipment should be adjustable.

In *Fig.* 386 is shown a *Semi-automatic Liming Device*, as designed by the author for a raw sugar factory, where it has given good operating performance. The milk-of-lime is pumped into a trough *a*, having an adjustable overflow *b* and falls on an inclined bottom plate. The chute, thus formed, is divided

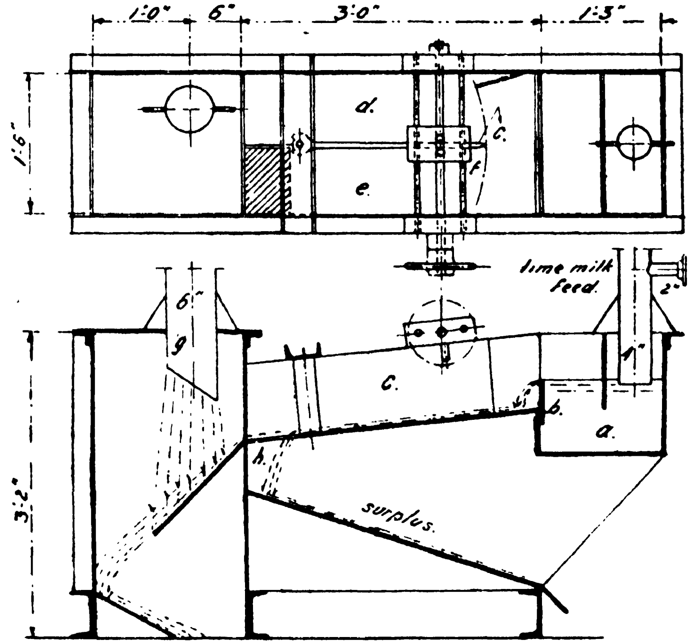


Fig. 386.—Semi-Automatic Liming Device.

into two parts *e* and *d* by a rotatorily mounted lip *c*. The lip can be set in any position by the crosshead *f*, which is operated by a handwheel on a threaded spindle. The milk-of-lime, which flows along *d*, will mix with the juice, which is discharged in a continuous flow from the pipe *g*, on to a pair of baffles, before reaching the receiving tank beneath.

The surplus milk-of-lime flows along *e* and falls down at *h* on an inclined plate, it being returned to the lime-milk tank.

The apparatus shown is designed for a 1600-ton factory, and needs about 0·007 per cent. lime-milk of 15° Bé. on the volume of juice.

Where very thorough mixture is required, the *Forced Circulating Agitator* or mixer, as shown in *Fig.* 387, can be used to advantage. This mixer is of the intermittent type, which can be built into any existing tank, and is driven either by belt or electrically.

A parabolic funnel *a* is attached by four baffle plates *b* to the tank wall, thus rectifying the whirling action caused by the impeller *p*. The lime-milk can be added at any desired point, as it will be thoroughly mixed with the juice, and the time required for mixing and discharge will be within the filling time of a second tank, when these stirrers are used in pairs. A large discharge connexion is convenient for this kind of equipment, so that the tank can be readily emptied into a supply tank placed underneath.

These agitators are also very efficient for mixing syrups or liquors with vegetable char for refining purposes.

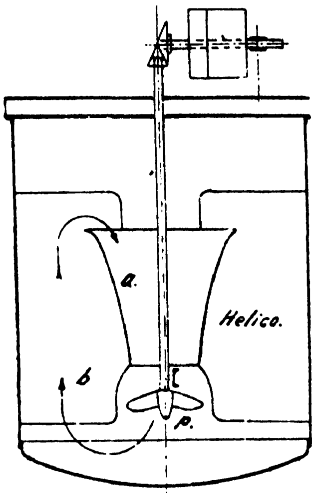


Fig. 387.—Forced Circulating Agitator.

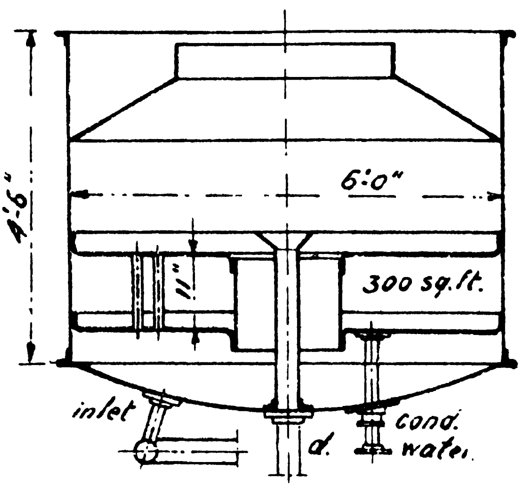


Fig. 388.—Calandria Eliminator.

In the limed juice supply tanks, a micro-organism, known as *Leuconostoc*, will propagate rapidly and develop a spongy cake of dextran, which may obstruct the pump intake, when a strainer is not at hand, or otherwise, and may grow in the pump cavities to such an extent that piston pumps have been known to burst from the compression of this material against the covers. *Leuconostoc* has to be removed, and a disinfectant employed, but in severe cases of this kind, the unlimed juice must be heated to boiling point, at which temperature the organism is killed; the liming thus has to be done in hot juice.[1]

3.—Defecators and Subsiders.

For proper decanting, the juice has to be raised to boiling point; this can be judged by the cracking of the scum layer on top of the juice. Where the subsiders are not equipped with heating coils, defecators or eliminators are used. In *Fig.* 388 is shown such a *Calandria Eliminator.* The charged juice is equally distributed at the bottom, and will circulate through copper tubes of $1\frac{7}{16}$ in. outside dia. and be discharged into the funnel, at the tank centre, connected to the outlet *d* of the tank.

Coils can also be used in these eliminators, which then are heated by reduced pressure live steam, so as to have a heating medium of higher temperature.

[1] See H. C. PRINSEN GEERLIGS, "Cane Sugar and its Manufacture," pp. 29, 143.

The temperature of the juice should be raised nearly if not quite to boiling point in the juice heaters, and the eliminator has thus only to supply the heat for reaching the cracking point. The heating surface required for live steam coils is about 100 sq. ft. for each of two eliminators for a 1500-ton factory. Calandria eliminators, using exhaust steam, require about three times this heating surface.

A scum gutter must be provided, having a few holes at the lower end for returning the scummed juice. The juice level of these eliminators is kept only a few inches above the coils or top tube plate of the calandria.

From the eliminators, the juice proceeds to the subsiders or settling tanks. A *Float Type Subsider* which is extensively used in Java and some other countries is shown in *Fig.* 389. At a swivel joint *c* a swinging pipe 3 to 4 in. in dia. is connected, having a double slot for the clear juice inlet at *b*, the pipe end being capped. A cylindrical copper float *a* is attached to the end of the swinging pipe and thus rises or falls with the juice level. Care should be taken that the upper part of the slot *b* remains below the juice level, so as not to admit any scum into the clear juice pipe.

A sampling cock is attached to the main right-angled discharge valve, from which the operator learns when to open this main valve. The clear run-off is observed frequently by means of a 4 in. long glass sampling tube, held in a wooden handle. An electric light bulb should be handy close to each subsider, to assist in judging the decantation by the transparency of the juice in the sampling tube.

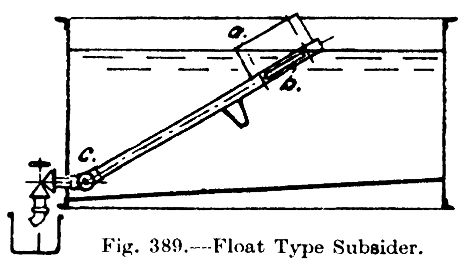


Fig. 389.—Float Type Subsider.

As soon as cloudy juice emerges from the discharge outlet, the swivel pipe is swung to the muddy juice partition of the juice gutter. To avoid excessive evaporation and corresponding cooling of the juice in open gutters, pipelines with double funnels can be applied to advantage at the position of the subsider orifices.

A *Three-Valve Subsider* of recent design is shown in *Fig.* 390, the three valves being connected to inverted funnels, inside the tank and the lower funnel being carried to the tank centre, so as to avoid carrying over the mud from the inclined bottom.

The juice level is shown by a float indicator and there is an overflow arrangement included. The charging pipe is at the bottom to avoid the excessive formation of scum. The aeration during filling can also be avoided when charging from the top, by extending the charging pipe inside the tank well towards the bottom.

The juice discharge valves have air vents to prevent syphonic action which might force scum into the clear juice or disturb the juice when the syphon is broken.

The mud is discharged by a gate valve at the bottom, operated from above the platform. The subsider has a steel cover and is well coated with insulating material; this last should be applied to every settling tank.

The clear juice discharge has a swivel orifice, so that the juice flow can be directed into the clear juice partition c or into the muddy juice partition *m*, the latter being pumped back to the subsiders.

The round shape of the tank and the conical bottom will assist in discharging the mud quickly and efficiently without excessive washing. The mud is collected in a second set of subsiders or *cachaceras*, the capacity being calculated at 25 per cent. of the total juice to be handled and 3 hours settling cycle (about 1 to 1·5 cub ft. per long ton cane grnd./24 hrs.). Heating coils or tubes with perforations on the under side are laid on the bottom of these *cachaceras*, so as to bring the juice again to boiling temperature by means of live steam ; the boiling is judged by the cracking of the scum layer, as already mentioned.

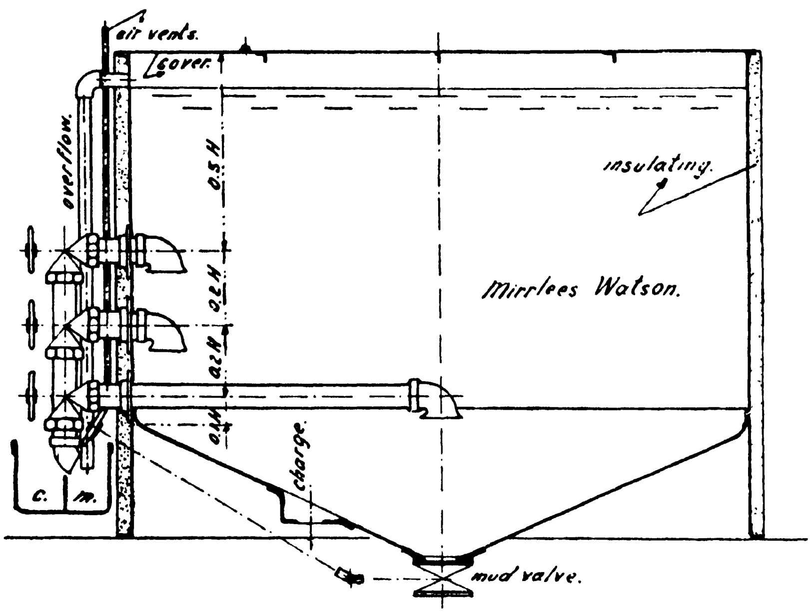


Fig. 390.—Three-Valve Subsider.

The clear juice is released similarly as in *Fig.* 390, but sometimes five cocks or valves are needed, owing to the less efficient settling of this secondary juice.

The *cachaceras* discharge the mud into *scum tanks*, placed underneath, which are provided with stirrers. A steam ejector for heating and additional circulation can be arranged to advantage. The mud contains generally between 15 to 20 per cent. of solids, which is pumped either diluted or undiluted to the filter-presses or else is delivered on the bagasse blanket of the intermediate carriers between the mills and thus will be burnt with the bagasse, the filter-press station being eliminated.

Instead of using eliminators or coils in the subsiders, the juice can be heated in the juice heaters above the boiling point ; this is feasible when a hydrostatic head is produced in the ascending juice line behind the heaters. The author knows installations where 215 to 220°F. is attained, but this highly heated juice cannot be released directly into the subsiders, as it will flash as soon as it comes under atmospheric pressure, so a *Flash Tank* is required, a design of the author's being shown in *Fig.* 391. The juice enters at the periphery close to the top in a tangential direction, so as to cause centrifugal or cyclonic action, which will keep the juice particles close to the tank walls, until released at the bottom. The flash tank should be insulated, while the vapours are discharged by a 6 in. pipe connexion through the roof. This tank, designed for a 1200-ton factory, measures 4 ft. in diameter.

To simplify the settling station, continuous subsiding or clarification is now used in many cane sugar factories and operating results have proved satisfactory. The *Continuous Closed Clarifier* shown in *Fig.* 392 illustrates the early design of this kind of equipment; it is a cylindrical vessel, having a steep conical bottom and provided inside with an inverted conical shield.

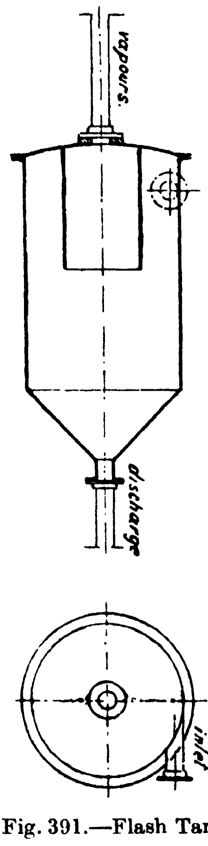


Fig. 391.—Flash Tank.

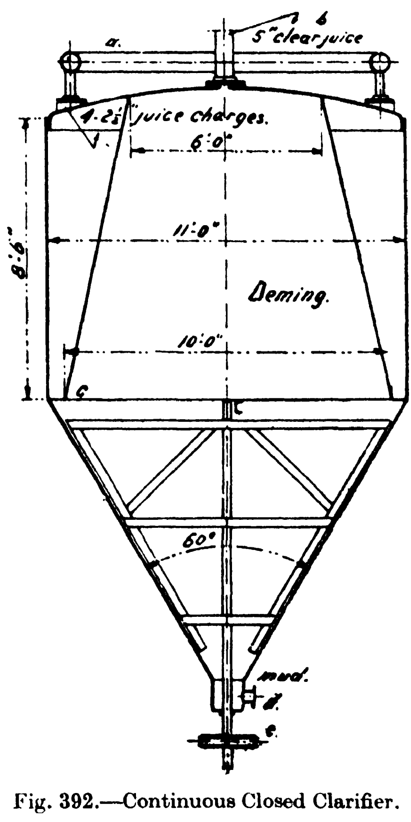


Fig. 392.—Continuous Closed Clarifier.

The juice is charged in through four inlets on the outside periphery of the dome and is forced down on the outside of the shield, whence the clear juice rises to the clear juice discharge *b* at the top centre. A distributing juice pipe *a* is connected to the four inlets. The mud will settle in the conical bottom, where it is guided by scrapers towards the mud discharge *d*. The scraper shaft is driven by a worm wheel *e*, making about one revolution in five minutes.

The clarifier shown is sufficient for the juice from about 500 tons of cane per 24 hours and a fair settling performance is achieved, as the author has found. Manholes are provided for easy access to the inside for cleaning in the dead season.

So as greatly to reduce the space occupied a very ingenious clarifier, originally used for sewage and ore settlings, has been applied to cane juice clarification with equally good operating results. The essentials of this *Multi-tray Counter Flow Clarifier* are sketched in *Fig.* 393. The designers have put into practice a shallow depth tray design and the construction may be taken as consisting of four super-imposed trays. These shallow trays will separate a larger quantity of clear juice than the deep tray type in the same interval of time.

The tank is closed and is well covered with insulating material to minimize heat losses, the discharge temperature being around 200°F. (93°C.). Moreover, the mud is concentrated to a porridge-like consistency, not attained in the common type intermittent subsiders, so the amount is only about 5 per cent. on juice with well over 20 per cent. solids.

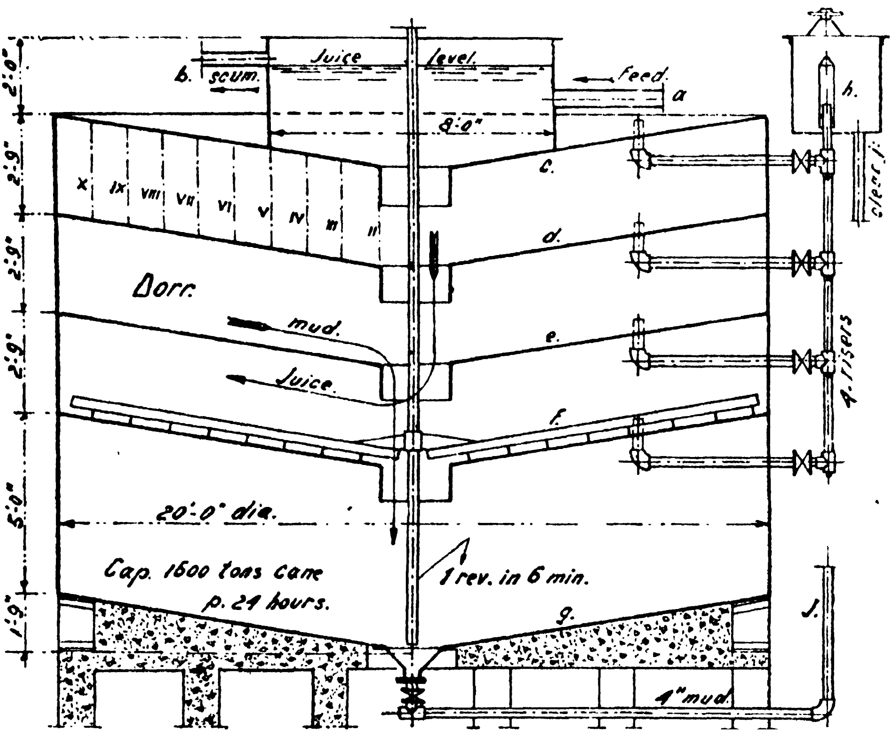


Fig. 393.—Multi-tray Counter Flow Clarifier.

The juice is charged in at *a* to the top basin, where also is a scum discharge, which is floated off at *b*, this being also the juice level control overflow. The juice enters through a 2 ft. opening into the upper compartment and through the centre well into the lower ones, which all have flush conical bottoms. All five bottoms, *c-d-e-f-g*, are scraped by brass scrapers, which are attached to arms rotated by the vertical centre shaft. The trays have to be made very exact, to render this scraping action efficient.

The clear juice is released through several inside orifices at the highest points of the trays, these outlets being connected to ascending pipes outside the tank proper, which discharge into the clear juice recipient *h*. All four risers have telescopic level adjustments, so that it will be possible to draw from each compartment as much juice as may be desired; normally more clear juice is drawn from the upper trays than from the lower ones. The author has seen glass tubes, connected parallel to the risers, lighted by an electric bulb so that the quality of the clear juice from the four compartments could be judged by its transparency. The lower compartments generally show the darkest and sometimes a cloudy juice, and maintenance of maximum temperature is very essential for good settling performance.

As the juice enters the centre well, through which the mud from the tray bottoms is also released, a part of the mud will be dragged along by the flow, this being especially the case in the lowest compartment.

The mud is released through a bottom pipe having a riser *j* reaching above the upper tank level, where a vertical diaphragm pump is attached, the pump having an adjustable stroke, so as to be able to pump the mud at the desired consistency. Instead of discharging the mud by the said pump, the author has seen the mud extracted at a lower level by a throttling gate valve in the riser and the operators found this method caused no inconvenience. The hydrostatic juice pressure thus has to overcome the pipe friction and the difference in specific weight between the mud and the juice.

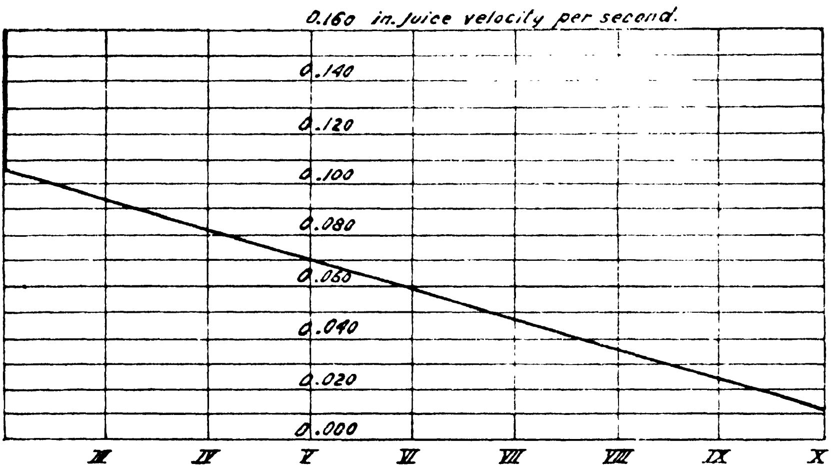


Fig. 394.—Diagram of Juice Velocity in Multi-tray Clarifier.

When starting such a continuous clarifier, care should be taken when the clear juice discharges, as any air entrapped in the compartments may cause the hot juice to spout, to the danger of the operators. The telescopic level adjusting tubes are therefore fitted with shields. Manholes for cleaning are provided at each compartment, so as to remove after each crop dried scum which is found adhering to the upper part of the compartments.

Properly operated, these continuous clarifiers will give a good clarification and the cleanness of the defecation department is greatly increased, the occupied space being considerably reduced as compared with the common type of subsider. The author has seen many of these clarifiers in satisfactory operation. But before making a purchase a comparative estimate of initial cost, and operating and capital charges should be obtained.

In *Fig.* 394 the author has drawn a *Juice Velocity Diagram* of the clarifier of *Fig.* 393, it being assumed that each compartment receives the same amount of juice and it is seen that very low rates of flow are obtained. The compartments in *Fig.* 393 are divided by vertical lines *II* — *X*, which lines are also to be found in *Fig.* 394. At *X* a minimum velocity of 0·0112 in. per sec. is reached, this being so low as not to disturb the settling performance.

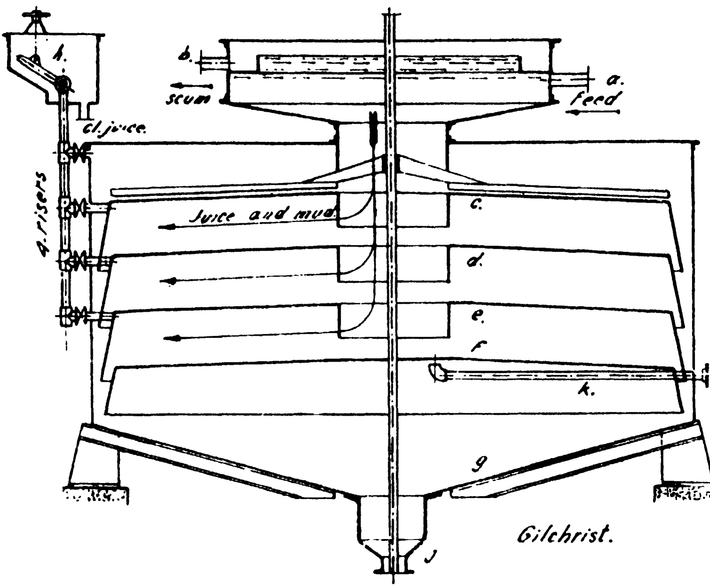


Fig. 395.—Multi-tray Parallel Flow Clarifier.

In *Fig.* 395 is shown a *Multi-tray Parallel Flow Clarifier*, based on a similar principle to the one of *Fig.* 393, but here the juice and mud flow in the same direction. The inclination of the tray bottoms is not as favourable for the withdrawal of the clear juice as in the case of *Fig.* 393. Such an apparatus has been put to practical test in a cane sugar factory but with results as yet unknown to the author. The clear juice discharge is adjusted by swivel pipes *h*, the trays *c-d-e-f-g* being scraped by a central scraping device. The lower compartment returns the extracted juice to the juice feeding tank, as it might be cloudy. The mud is released in a concentrated state at the bottom. Further developments of this parallel flow clarifier should be watched with interest.

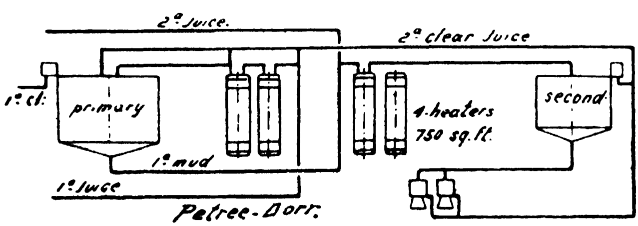


Fig. 396.—Compound Clarification.

With our present-day milling methods, more sucrose is extracted, but at the same time also more impurities, generally of a colloidal nature, which require special attention at the defecation and filtration stations. The juice of the last mills of a train will contain more of these impurities through the prolonged milling and the application of imbibition and maceration.

For this reason the clarification of the impure or secondary juice has been separated from that of the clearer or primary juice. A scheme of so-called *Compound Clarification* is shown in *Fig.* 396 for a 1200-ton cane sugar factory. The primary clarifier of the continuous type receives the first mill juice, which is heated by two heaters, each of 750 sq. ft. heating surface to 220°F. or well above the boiling point and a flash tank is therefore required. The mud of the primary clarifier is mixed with the secondary juice, which will increase the decantation in the secondary clarifier, after having been heated in a heater of 750 sq. ft. to 212°F. The mud of the secondary clarifier is filtered in filter-presses, or returned to the bagasse blanket of the intermediate carriers of the last mills. The clear juice from the secondary clarifier as well as from the presses is returned to the primary clarifier, after having been heated to the desired temperature. The clear juice from the primary clarifier is pumped to the evaporators in the boiling house.

The capacity of the primary clarifier is based on the total amount of juice handled, whereas the capacity of the secondary one is fixed at half that amount.

Compound clarification results in a better removal of the colloids than does single clarification, the latter removing approximately 12 per cent. as judged from tests made, whereas compound clarification removes about 21·5 per cent. and a slight rise in purity of the treated juices is generally obtained.

4.—Centrifugal Separators.

Centrifugal separation originated in the dairy industry for separating the cream from the milk. Today this kind of equipment is to be found to a wide extent in many chemical industries and has also been tried in the clarification of cane juice. An original *Juice Centrifugal Clarifier* is shown in *Fig.* 397, which has been put into practical operation in the cane sugar industry in South Africa. The juice is charged by a pipe *f* at the bottom of the basket and fills the double conical drums *d*, which are shown in open position for mud removal. The heavier solids will be thrown in the grooves between the drums *d* and the clear juice will flow over the top lip, being discharged at *g*.

The hollow spindle *a*, which drives the basket, has a rod *b* inserted, which operates the toggle gear *c* for lifting or lowering the sleeve *e*, and has lugs attached for opening the conical drums. In opened position, the mud is thrown into the monitor casing and dropped at the bottom into a suitable mud conveyor underneath.

The rod *b* is operated by a hydraulic cylinder on top of the spindle drive and a timing gear is

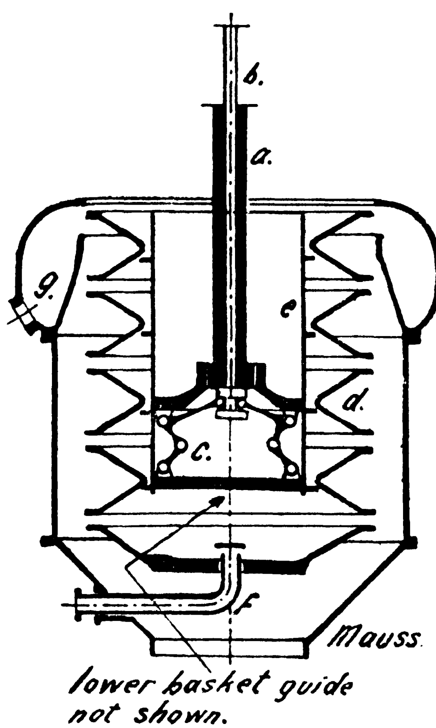


Fig. 397.—Juice Clarifying Centrifugal.

provided for periodical washing with water, thus de-sugarizing the cake, and giving a periodical discharge of mud.

The centrifugals installed in 1921 had baskets of 28½ in. outside diameter revolving at 1000 to 1200 r.p.m. for a capacity of about 800 gallons of juice per hour. The unsettled juice as analysed contained 3·81 per thousand solids in suspension, whereas the clarified juice had only 0·13 per thousand. Further developments, as well as data about the mechanical operation of this centrifugal, have still to be published.

Centrifugals with blind baskets, thus without perforations, of the 40 in. × 24 in. size, revolving at 1000 r.p.m., have been used for clarifying molasses with satisfactory results. According to the authorities concerned, these centrifugals can be used to greater advantage for syrup and even thin-juice clarification and the tests made with the centrifugal clarifier of *Fig.* 397 may demonstrate that centrifugal separation is feasible for cane juice. Additional power output is of course required and future tests or applications should first be tried in those factories where the common subsiders cannot cope sufficiently with the task of proper decantation.

5.—Sulphitation and Carbonatation Apparatus.

Raw sugar factories employ common subsiders, and no further clarification is required for raw sugar of 96° polarization. But for plantation white sugar of well over 99° test, the requirements for juice clarification will be quite in excess of what the regular subsider can cope with, and a process of sulphitation or carbonatation of the juices has to be resorted to. Competition with refined sugar is no easy job, but the manufacture of plantation white sugar has been greatly improved, and the process of sulphitation is successfully carried on in many countries for producing direct consumption sugar at the cane sugar factory. The carbonatation process is used exclusively in beet sugar factories, but in Java this method of manufacturing has been successfully applied also to cane juices, and many improvements have been achieved by the leading sugar experts of that country, the yield or recovery in sugar for carbonatation houses being from 1 to 2 per cent. higher than for sulphitation or defecation factories.

The performance of sulphitation is carried out in cylindrical tanks, variously called sulphitation tanks, sulphitators, saturators or sulphurators, the SO_2 gases being admitted at the bottom through cast iron, lead, copper or brass distributing devices. A foaming capacity of about 50 per cent. of the juice volume is advisable. The tanks are covered, and are provided with a chimney, generally made of wood, for carrying off the noxious fumes, which affect the throat and eyes of the operators when not properly removed beyond the roof.

In *Fig.* 398 is shown a *Sulphitator*, having the principal features of present day designs. The SO_2 is charged in at the conical bottom by a cast iron vane piece, while the juice is charged and discharged at this same spot. The sulphitator, therefore, is of the intermittent or batch type, which will allow for proper trituration of the juice. A thorough mixing of the SO_2 gas is essential and any SO_2 contained in the fumes after passing through the juice is lost. A liquid height of 7 to 10 ft., through which the SO_2 is forced, is hence well established practice. Test cocks for sampling have to be provided and the performance can be observed through sight glasses. When milk-of-lime has to be added, a special connexion is provided, the main line for the milk being well above the juice level, so that the juice cannot enter it. A closed circuit with taps, from the lime stirrers and back, has to be made, and sharp bends or branches avoided, as these will get clogged with the decanted lime.

A perforated steam pipe at the bottom is arranged for agitating, a thermometer also being provided. The tank cover is of cross laid double boards, whereas the chimney is also made of wood, to safeguard against the corrosive action of any sulphuric acid which may be produced.

Instead of passing the SO_2 through the juice column, which requires a gas pressure equal to the hydrostatic juice pressure, *Sulphitation Towers* are sometimes used, these being wooden columns about 16 in. square and up to 30 ft. high, provided on the inside with baffles, so as to give a zig-zag flow to the juice, when charged at the top. The SO_2 is introduced at the bottom, thus in counter-current. It will be obvious that the efficiency depends on the height of the tower and the number of baffles fitted.

Another type of sulphitation apparatus is the *Shower Type Sulphitator*, this being a cast iron round tank, with the juice entrance at the top. The juice is spread over five or six perforated cast iron or copper plates, to fall in drops to the bottom of the tank, the velocity of fall being interrupted by the perforated baffles. The SO_2 is introduced at the bottom, thus again in counter-flow. Both the sulphitation tower and shower type of apparatus give continuous operation, the regulation of the acidity of the juice being achieved by the juice flow or the amount of SO_2 supplied by the sulphur stove. Batch type sulphitation, which has a better regulation, is therefore preferred in several countries.

A *Continuous Jet Sulphitation Apparatus* of unique design is shown in *Fig.* 399. The juice is charged at *a* into a closed tank *b*, having a partition wall *c*. The juice is aspirated by the circulating pump *d* after having passed a strainer to get rid of coarse impurities, and it is pumped through the ejector *e*, which draws the SO_2 gases from a sulphur furnace at *f*, the pumped juice being discharged again into the closed tank, whence it is released at an overflow *g*.

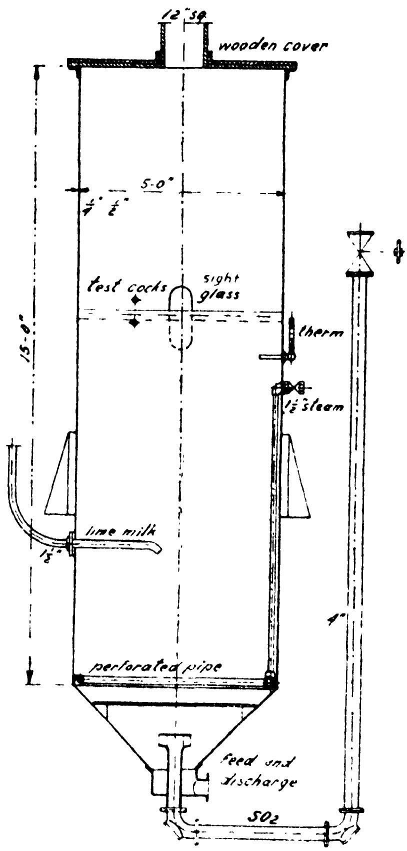


Fig. 398.—Sulphitator.

It will be clear that the maximum pump output has to be greater than the charging volume of juice, and by speeding up the circulating pump a more intensive sulphitation is achieved. For this purpose the partition wall *c* does not reach as far as the tank bottom.

The fumes left over, consisting mostly of nitrogen, are released at *h*.

The sulphitation of cold juice in batch type sulphitators requires about 10 minutes on an average and for hot juice still less. For a complete cycle, on which the required tank capacity has to be calculated, about 24 to 30 minutes should be allowed. In those instances where only slight sulphitation is required, as with the carbonatation-sulphitation process, the cycle may be shorter.

Intermittent or batch type sulphitators have to be arranged in pairs or in threes and if the existing capacity happens to be low, the position can be improved by installing a supply tank, the latter receiving the juice or syrup as it is produced and giving rapid discharge into the sulphitators; thereby considerably shortening the filling time.

Carbonatators are based on a performance similar to that of sulphitators. Carbonatating towers are not known to the author, but the shower type is used to a larger extent in beet sugar factories, it being observed by SHUMILOV

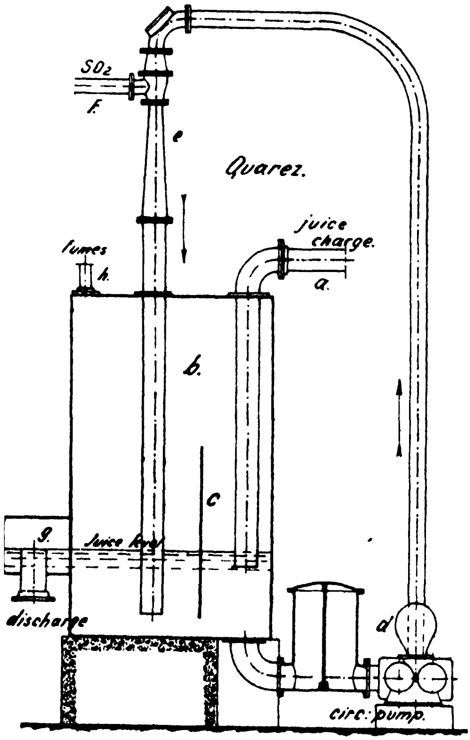


Fig. 399.
Continuous Jet Sulphitating Apparatus.

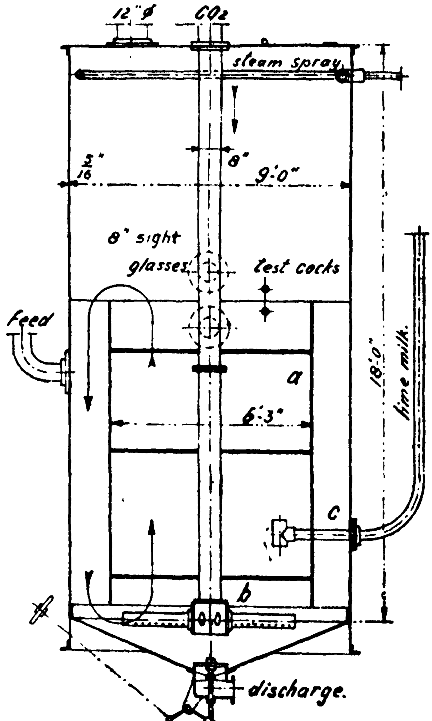


Fig. 400.
Batch Type Carbonatator.

that the number of trays and cones has a direct bearing on the saturation, the effectiveness being observed as follows :—

One tray and cone	43·5 per cent.
Two trays and cones	49·5 ,,
Three ,, ,, ...	54·3 ,,
Four ,, ,, ...	59·2 ,,

This shows clearly the importance of proper intermixture and the intermittent type carbonatators used in Java and other cane-growing countries do not show any inferior performance to that just cited.

Carbonatators have also been applied in compound arrangement, the fumes of the primary carbonatator being charged at the bottom of the secondary.

The gas pressure in the primary carbonatator varies then between 7 and 14 lbs. per sq. in.

The general design of an *Intermittent or Batch Type Carbonatation Tank* is shown in *Fig.* 400, the size being taken from carbonatators supplied by the author. The tanks are covered with a sheet iron plate, and have a chimney of the same material, as the CO_2 fumes have little corrosive action on iron or steel.

This carbonatator is a cylindrical tank in which a special feature recommended in Java[1] is added, consisting of a concentric cylinder *a*, which has a diameter of 0·7 the tank diameter, a few perforated plates being attached in this cylinder for distribution of the CO_2 gases released from below. The juice circulation produced as indicated by the arrows will increase the efficiency of this intermittent carbonatator. At the beginning of the carbonatation cycle the efficiency is high, as the juice will readily absorb the CO_2, but as the saturation proceeds, the efficiency drops to about 50 per cent. The CO_2 is charged at the top through a centre pipe, made of cast iron, carrying six perforated distribution branches *b* at the lower end. In the De Haan carbonatation process, the juice temperature is kept at about 125°F. (50°C.) in the first carbonatation, the lime being added simultaneously with the CO_2; the lime charging pipe *c* is arranged accordingly.

A perforated exhaust steam pipe is placed near the top of the tank to obviate excessive foaming. With the De Haan process, the foaming can be kept within reasonable limits and a 10 ft. space above the juice level is sufficient. The author has had experience with heavy liming at 185°F. (85°C.) before the juice was charged into the carbonatators, but the carbonatation performance caused so much foaming that this was blown out through the chimneys, about 30 ft. above the carbonatator top on the roof.

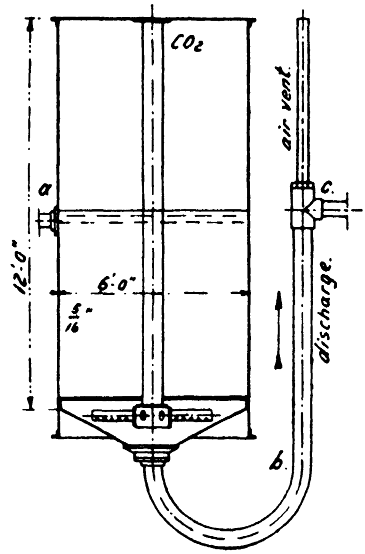


Fig. 401.
Continuous Carbonatator.

In *Fig.* 401 is shown a *Continuous Carbonatator*, as used for second carbonatation, the juice being charged at *a* and discharged by a syphon *b*, provided with an air vent for overflow at *c*. The CO_2 is charged in similarly as shown in *Fig.* 400.

The time required for a complete cycle of the intermittent type of carbonatator is from 30 to 45 minutes, from which time the capacity of the station has to be calculated.

1 See article by VAN DORT, *Het Archief*, 1933, Vol. 41, p. 257.

CHAPTER XX.

FILTRATION EQUIPMENT.

FILTERS — FILTER-PRESSES — LEAF FILTERS AND THICKENERS.

The coagulated or precipitated insoluble impurities of the juice have to be separated, and as this separation cannot be completed in the clarifiers or subsiders, without heavy loss of juice or sucrose, the final stage of separation is done in filters, dividing the impure sugar solution or mud into a cake with a low sugar content and a clear filtered juice. As the quality of the sugars made depends in the first instance on the absence of all insoluble matter, it will be obvious that raw sugar factories practising clarification by settling need only to filter the sludge from the defecators, whereas with direct consumption sugars the whole of the juice itself may require to be filtered.

1.—Filtration Principles and Data.

The act of filtration is accomplished by passing at low or higher pressure the juice, syrup or liquor containing the insoluble impurities through a porous filtering medium, which will retain those impurities and give free passage to the clear juice through its capillaries.

There are different filtering media, which are used either alone or in combination, and the following are used for filtering impure sugar solutions :—

For clarified juice or syrup :
: Coke, gravel, sawdust, wood shavings (excelsior), bagasse, sand or palm fibre.

For defecation sludge, sulphitation or carbonatation :
: Cotton, hemp or other fibre cloth, metallic cloth with or without filter-aids (such as diatomaceous earth [kieselguhr] and the like), and porous stone.

For complete removal of impurities and colour in white sugar manufacture :
: Animal or vegetable char.

All the filtering media create resistance to the passing juice, syrup or liquor and this resistance increases as the pores or capillaries clog up or the cake layer thickens, thus diminishing the rate of flow and requiring a higher liquid pressure to overcome it.

The finer filtering media need a support of coarser structure and where such is applied, the coarser pores of the support have to be covered with the finer filtering media, before complete filtration is reached, the first run-off thus being cloudy and having to be returned to the filtering cycle. When using filter-aids, like diatomaceous earth (kieselguhr) or vegetable char, this precoating of the woven filter-cloth is essential.

The rate of flow per unit of time and per unit of filtering area is a definite indication of the filtering performance, and several factors, mentioned below, have a bearing upon it :—

(*a*) The viscosity of the liquid to be filtered.
(*b*) The granularity or capillarity of the precipitate, depending upon the size.
(*c*) The pressure of the liquid.

(*d*) Temperature of the liquid, which has a bearing upon the viscosity (*a*) but also upon the coagulation of colloidal or gummy matter.

(*e*) The thickness of the cake formed.

(*f*) The amount of gummy or colloidal matter in the liquid.

These factors may attain such magnitude that filtration may become impossible.

The *Viscosity* of syrups will be lessened by raising the temperature before filtering, and according to BRENDEL the flow rates of syrups from a beet factory are as shown in the graph *Fig.* 402; these may be applied equally to cane sugar syrup. The rate of flow increases with the temperature and therefore syrups should be filtered at as high a temperature as the process work will allow, generally at about 85°C. (185°F.).

The *Granularity* of the precipitate is the direct result of lime or other reagents and a proper temperature of the juice, and the sugar chemist has to pay full attention to the matter, to ensure the best possible mechanical performance of filtration. Defecation mud is always difficult to filter and a percentage of 10 to 20 fibre in the cake as produced will be required to form

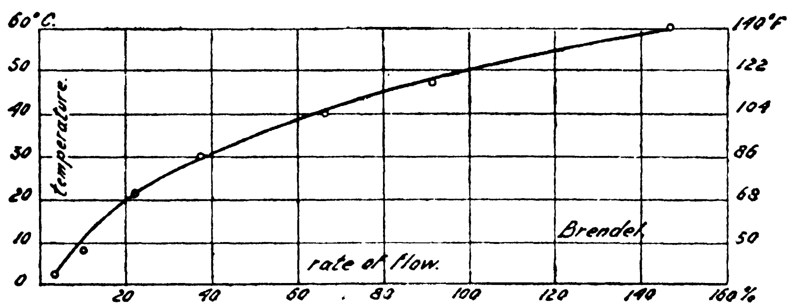


Fig. 402.—Graph of Syrup Flow (Brendel).

a good cake. Fortunately, present-day cane preparation by revolving knives, shredders and deep mill roller grooving has materially increased the amount of fine bagacillo in the juice—sometimes to such an extent that fine juice straining becomes a necessity at the mill.

Carbonatation and sulphitation muds are easier to filter and here an increased thickness of cake can be allowed.

Juice Pressure starts at about 20 lbs. when the filter-presses are charged, and runs up to 40—60 lbs. per sq. in. by the end of the filtering performance. Leaf filters will stand the same pressure, but low pressure filters, like bag filters, are not made to work at any pressure much above the atmospheric. But the point should not be missed that pressure may be detrimental to complete filtration, and colloid filtration nowadays is carried out at very reduced gravity pressure, the rate of flow being thus very low and requiring large filtering area. Filter-aids of proper microscopic structure have therefore a well-defined purpose, as they allow a higher pressure with a correspondingly increased rate of flow, the capillaries in their mat being of such a size that impurities cannot be dragged along through this increased rate of flow.

The filtering pressure should be maintained without any wide fluctuations, as these would impair good filtration work. When a clean filter is cut into a battery of already operating filters, the pressure will drop and the operation of the latter filters is weakened until sufficient cake has been formed in the clean filter to restore the pressure hitherto prevailing. It is advisable to have a charging filter pump for large installations, so as to avoid this fluctuation in pressure.

Another reason for this pump is that it also serves to prevent the cake falling from the filtering cloth, a contretemps which may make sweetening-out (by washing later on) less efficient if not well nigh impossible.

Sometimes use is made of a *Montejus*, a closed cylindrical vessel, filled with juice, where a steam or air pressure of about 60 lbs. acts on the liquid surface, the discharge being through an internal pipe, the open end of which is near the bottom of the vessel. This apparatus will give an even pressure, but its performance is intermittent and an unfavourable steam consumption obtains. Constant attendance is also required, so the *montejus* has been discarded in some cases, although fully automatic ones are now on the market. Plunger duplex pumps with spherical valves, either steam, belt or electrically driven, or single acting triplex pumps are used to a wide extent, it being of paramount importance that the juice flow should be steady and uninterrupted, to avoid any hammering effect in the filters. Large air vessels at least five times the pump displacement per stroke should be provided, the larger the capacity of the air vessel, the better the pump performance.

Centrifugal pumps are excellent so far as pumping performance is concerned, delivering a constant flow at nearly constant pressure, but in Java drawbacks have been found when working carbonatation juices, as the whirl inside the pump has been found to break up the granularity of the precipitate. Open impellers are more generally used, but heavy wear is to be expected, owing to the gritty material present in the juice.

The *Temperature of the Liquid* affects the viscosity, but on the other hand may have a detrimental influence on reversable colloids, when these are not coagulated or flocculated.

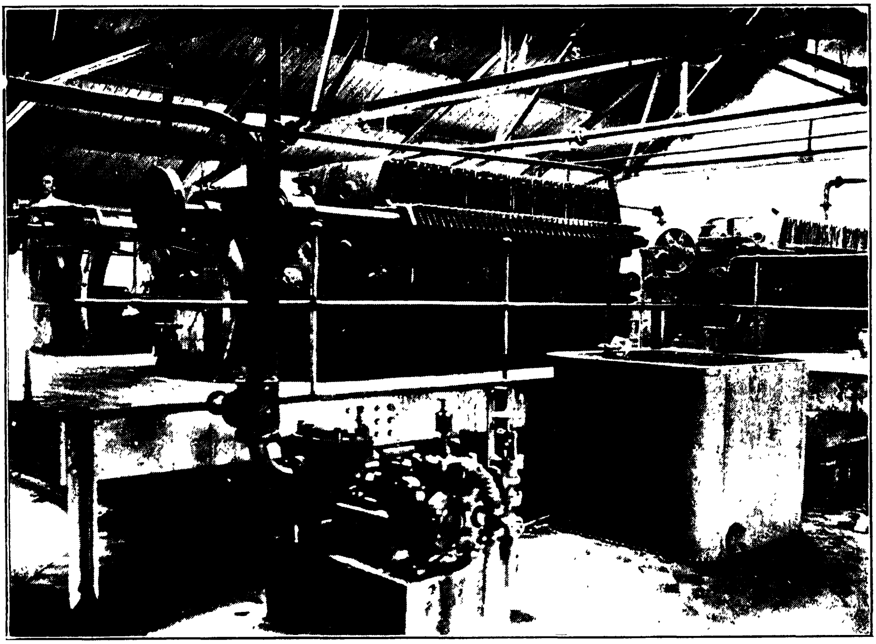
The *Thickness of the Cake* will cause proportionate resistance and 1 in. is the accepted standard for defecation factories, whereas carbonatation presses have frames up to 1½—1¾ in. thickness, thus increasing the holding capacity of the press.

The *Gummy or Colloidal Matter* in the filtration juice may make filtering impossible without the use of filter-aids, as in the case of syrup filtration and the filtration of certain juices.

The author has had experience with defecation mud, where the fine bagacillo was removed by a strainer as shown in *Fig.* 286, the average analysis of the filter cake, without fine straining, being :—[1]

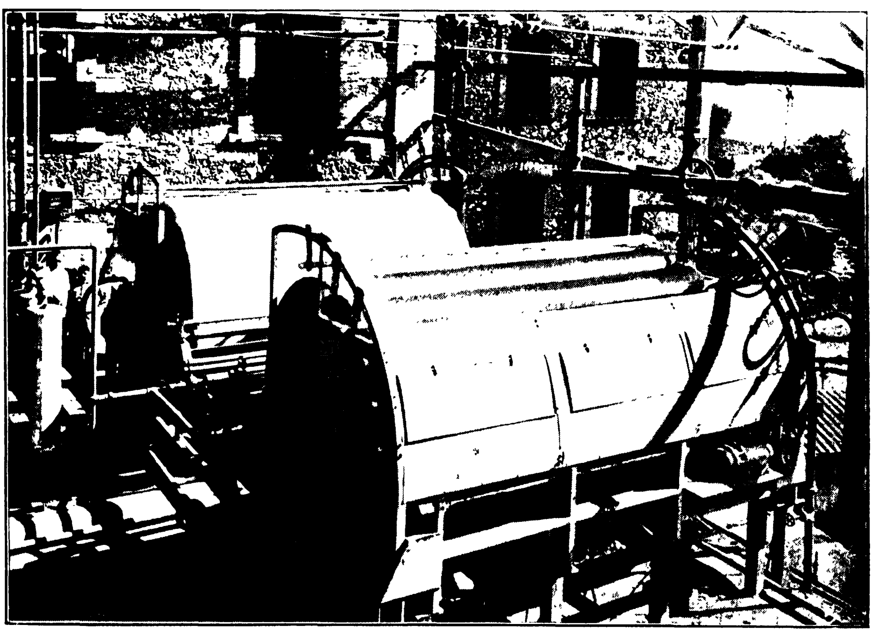
Moisture	50·68
Fibre	21·65
Wax	4·43
Albuminoids	2·33
Sucrose	6·12
Glucose	0·26
Gums	1·28
Phosphate of lime	4·28
Silica	4·24
Sand and clay	1·27
Undetermined	3·46
	100·00

[1] See the paper of J. C. GONZALEZ, Proceedings 4th Conference, Association of Cuban Sugar Technologists, 1930.



Norit Filter-press Station for 400 Tons Daily Refining Capacity.

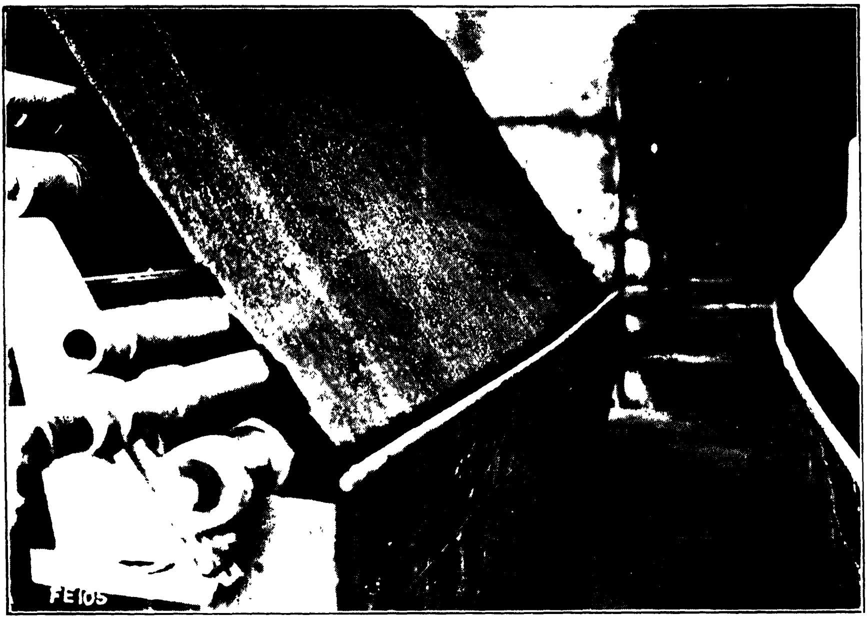
(N.V. Norit-Vereeniging)



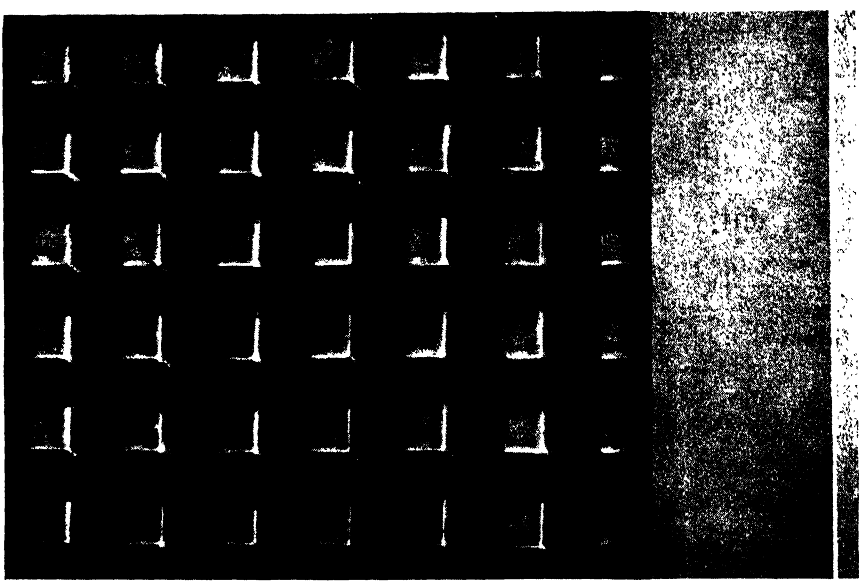
Drum Filters with String Discharge for Cachaza in Tropical Cane Sugar Factory.

(Filtration Engineers, Inc.)

PLATES 73 & 74.



CAKE DISCHARGE ON STRING TYPE DRUM FILTER.
(Filtration Engineers, Inc.)



" PYRAMID " PATTERN FILTER-PRESS PLATE.
(S. H. Johnson & Co., Ltd.)

It will be apparent that a very thorough preparation of the cane was effected, the fibre content being high. Moreover, the cake was not washed, this giving a higher loss in sucrose. After the installation of the strainer mentioned, about 80 per cent. of the fibre and some wax were eliminated and the formation of cake in the filter-presses became impossible, only a thin slimy coating being formed on the filter cloth, which clogged the pores. The filter mud therefore had to be returned to the bagasse apron, as shown in the scheme in *Fig.* 403. The advantages of this procedure are the elimination of the filter-presses and about 5 per cent. more fuel value in the bagasse, but the defecation requires about 33 per cent. more capacity and the use of a small amount of phosphoric acid had become necessary for good clarification. The hot clarified juice is weighed, as weighing cold juice is now less feasible owing to a correction factor being required to allow for the return of the mud to the milling station.

The mill in *Fig.* 403 consists of a 14-roller unit, preceded by revolving knives. The sludge from the second subsiders or *cachaceras* is mixed in a tank *a* with the fourth mill juice. Imbibition water up to about 20 per cent. on cane weight is applied at the last mill alone, and the mixture of residual juice and sludge is pumped by an unchokeable pump *b* (see construction, *Fig.* 274) to the distribution gutter *c* over the previous intermediate carrier.

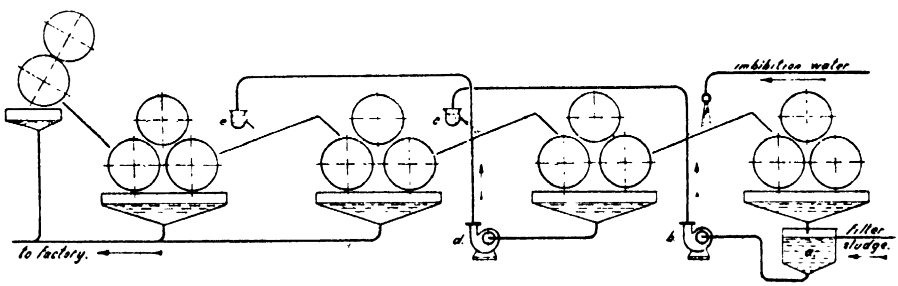


Fig. 403.—Mud Returning Scheme.

The cycle is repeated by a second unchokeable pump *d*, which delivers into the gutter *e*. As the cane has been treated by knives the raw juice trash from the drag type strainer is taken by an elevator and delivered in front of the crusher on the knifed cane mat. The raw juice from the crusher and the first two mills is pumped to the factory.

This *modus operandi* has worked satisfactorily in a raw sugar factory, the sludge being filtered twice through a bagasse blanket, before the raw juice is withdrawn to the factory. Good concentration of the sludge is very essential, as it reduces the amount of sucrose returned to the milling station. Continuous clarifiers or thickeners can be used to advantage for this class of work.

The returned sludge will increase the sucrose content of the final bagasse and the additional sucrose loss has of course to be kept at par with the loss of sucrose in the press-cake formerly prevailing. In *Fig.* 404 the author has drawn the ***Equivalent Sucrose Loss Diagram*** on a logarithmic scale, and it can be used as follows : On the line *II*, the point *b* indicates the amount of press-cake formerly obtained per hundred cane, being actually 1·65 per cent. The line *a-b* is now drawn and next the sucrose per hundred press-cake is marked at *c* on line *I*, it being 6·75 per cent., and *c-d* now is drawn parallel to *a-b*. The vertical *d-e* will yield at *e* on line *III* the sucrose loss in press-cake per hundred cane. The amount of bagasse (it being 26·76 per cent. on cane) is marked

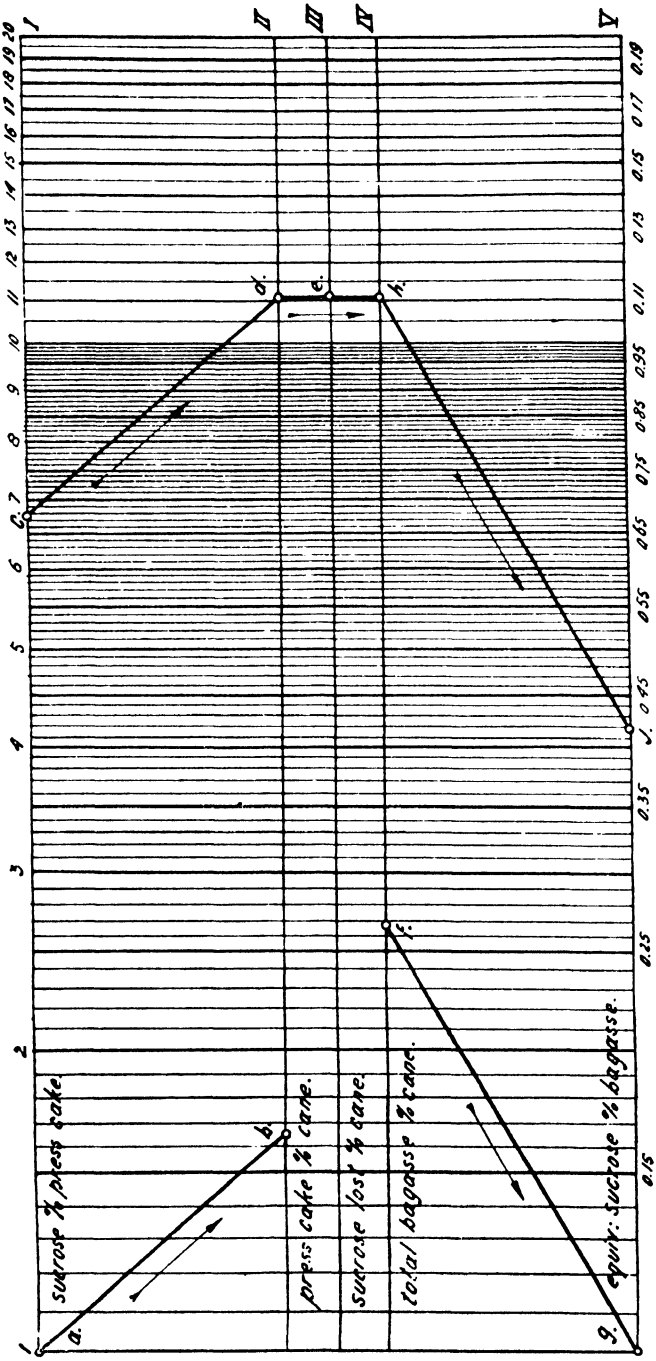


Fig. 404.—Equivalent Sucrose Loss Diagram for Return of Mud to the Mills.

at f on line IV, and remembering that the amount of bagasse has been increased by the amount of press-cake, the line f-g is drawn. Parallel to this line, h-j will now indicate at the intersection j on line V, the sucrose loss per hundred bagasse, equivalent to that formerly lost in the press-cake, it being 0·418 per cent. Before the defecation mud was returned to the bagasse blanket of the mills the sucrose content per hundred bagasse was 2·3, and there will now be no heavier sucrose losses till it rises above 2·7 per cent.

The calculation must yield the same result, thus :—

Sucrose in press-cake, 6·75 per cent.
Press-cake on cane, 1·65 per cent.
Sucrose lost on cane, $6{\cdot}75 \times 0{\cdot}0165 = 0{\cdot}1113$ per cent.
Total bagasse plus mud, $25{\cdot}02 + 1{\cdot}65 = 26{\cdot}67$ per cent. on cane.
Equivalent sucrose loss on bagasse, $0{\cdot}1113 \div 0{\cdot}2667 = 0{\cdot}418$ per cent.

The diagram can be used for any prevailing figures.

Animal and vegetable chars remove not only the colour, but also colloidal and other impurities. The first is porous and granular, whereas the latter has a very hairy appearance, when observed under the microscope, resulting in a great adhering capacity. As these filtering media are expensive, they are generally revivified for subsequent use, the loss in revivification being only a few per cent.

Alkaline juices, as a rule, will give a better filtering performance than neutral or acid juices.

The *Rate of Flow* depends on several factors, which are of unknown or varying magnitude, and manufacturers of filtration equipment are guided by practical filtration tests and rely upon these working data.

The Java Experiment Station has published a theory about filtration, according to the general rule :—[1]

$$V = C \times t^n \qquad (100)$$

in which V = the total volume of filtrate in the time t.
C = a coefficient being for Defecation filtration 5—6.
Liming with saccharate 3—5.
Part saccharate and part lime 4—6.
n = exponential factor at constant pressure = 0·50 — 0·55.

In the graph, *Fig.* 405, the *Total Flow Diagram* is drawn for C = 3—6 and $n = 0{\cdot}5$, from which it may be assumed that the filtration time for cane sugar factories is about 2 hours, as the rate of flow will then diminish rapidly. Of course the thickness of the cake to be formed is a deciding factor as well.

In the construction of filters, different types are used for different filtration performances ; these may be listed as follows :—

(*a*) Packed filters for sand, excelsior, bagasse, palm fibre and animal char.
(*b*) Bag filters for filtration through cotton bags from the inside to the outside of the bag.
(*c*) Plate and plate-and-frame filter-presses.
(*d*) Leaf type filters with fixed or revolving leaves.
(*e*) Drum type revolving filters.
(*f*) Thickeners or concentrators for thickening the mud.

These filters will be discussed in the corresponding sub-headings, the filter-presses *c* being the ones mostly used for reliable performance. The newer types *d*, *e* and *f* are nevertheless of ingenious design, having increased efficiency and reduced operating and maintenance charges. The washing of the cake formed in some of these more recent filter types achieves better de-sugarizing with a reduced amount of sweet-water.

[1] See the article of P. HONIG and W. THOMSON, *Archief*, Vol. 41, 1933, pp. 233–248.

2.—Packed Filters.

These filters are generally used for clarified juice or syrup, thus for removing only the last traces of impurities in the liquids to be treated. The construction consists of upright cylindrical vessels, having a perforated false bottom and a perforated plate on top of the filtering medium for compressing it. The false bottom, as a rule, is covered with a circular sheet of woven filter-cloth and a sheet is also laid on top under the perforated pressure plate. The flow is mostly upwards, the unfiltered liquid entering at the bottom and the filtered liquid issuing from the top. De-sugarizing with water takes the same direction of flow, whereas cleaning is achieved by a counterflow of clean water, which will carry along the filtered impurities. Compressed air is also used in several instances.

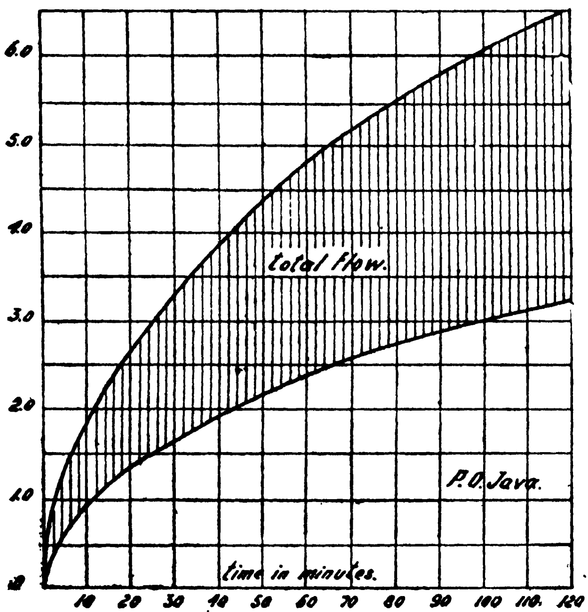


Fig. 405.—Total Flow Diagram.

Low gravity pressure is generally used and the rate of flow is small. Bagasse is subject to fermentation, when the filtering medium is not removed every five to six hours, the temperature of the juice being slightly above boiling point.

Eight to 10 cub. ft. per ton of cane ground per hour is required and for juice clarification the material is packed from 5 to 10 ft. high. Sand filters require sharp sand about 0·08 in. in size, and a square foot of filter area, about 10 ft. high, will treat about 25 cub. ft. of syrup per hour. These filters are made from about 2 ft. 6 in. to 5 ft. in diameter.

Bonechar filters are used for refinery sugar liquors, being up to 10 ft. in diameter and 20 ft. high, and having both ends conical. The filtration cycle is from 48 to 72 hours, after which the char has to be removed through the lower manhole for revivification. One lb. of char per lb. of sucrose present in the liquor to be filtered, is accepted standard practice in the U.S.A.

Packed filters have only a limited application for cane sugar factories.

3.—Bag Filters.

Bag filters also belong to the low pressure type and they will remove more impurities at a higher rate of flow than the packed filters, but slimy matter will quickly clog the woven bags. A pressure above 3 to 5 lbs. per sq. in. may rupture the bags, so a gravity head of 5 to 10 ft. is used in general practice.

The filters consist of a rectangular tank, generally having an inclined bottom and provided with a horizontal division plate, close to the top. This plate is removable and provided with a number of round nozzles, about 6 in. in dia. ; to these the bags are attached by twine or by an interior bronze nipple, the bags being about 5 ft. to 6 ft. long and hung vertically. A fine cotton bag usually forming the inside is enclosed within a coarse hemp bag to give additional strength, and the juice is passed from inside outwards, when it is collected on the inclined bottom for withdrawal. The filter chamber may also have a low pressure steam connexion to maintain a temperature of about 175°F. (80°C.).

Some 60 to 110 sq. ft. filtering surface is required per ton of cane ground per hour, or 8 to 15 bags of the above-mentioned size, the filtering cycle being from 12 to 18 hours.

Another type consists of rectangular bags, fitted over a frame, so as to maintain the shape and to prevent collapse. The juice is charged on the outside of the bags and the clear run-off from the inside space is discharged through a perforated pipe, having a connexion to the exterior of the filter tank. By folding bags of larger size, so that the filtering surface is arranged in accordion-like fashion, the rate of flow may be increased in nearly the same proportion as the ratio of folded surface to straight filter area.

Bag filters are used only to a small extent in cane sugar factories, their principal field being refineries and beet sugar factories.

4.—Plate and Plate-and-Frame Presses.

These two types comprise the filtration equipment mostly used in cane sugar factories and although maintenance, labour for operation, and the cost of cloth have to be considered, a very reliable filtering operation is performed.

The oldest type is the *Plate Press*, having recessed plates of square area, with the mud fed in through an opening in the centre of the plates. The plates are mounted on side bars and pressed together by a screw gear. The cakes are formed in the recesses of the plates between the two filter-cloths of each section or chamber. The centre feed opening necessitates the cutting of a corresponding hole in the cloths, and *two* adjacent ones are sewn together at this spot or else provided with clip nuts.

The advantages of the plate press are : fewer joints, lighter weight and reduced cost as compared with the plate-and-frame press. The plates are heavier and sweetening-out is less efficiently done. Its application in cane sugar factories is secondary to the plate-and-frame press.

The *Plate and Frame Press* has long been in use for nearly every filtration performance in the sugar industry at large, and from *Fig.* 406 the *modus operandi* can be learnt. The plates and frames are arranged alternately with the filter-cloth interleaved at each joint, the whole being held between a fixed head *h* and a follower or press plate, tightened by the screw gear. The head *h* is subject to deflection, causing the head ribs to be under tension at *x* and as cast iron has only a low tensile strength, sufficient material has to be provided at this spot, to reduce the tensile stress per sq. in. sectional area.

For sugar factory work these filter-presses have their frames made in sizes from 24½ in. by 24½ in. to 45½ in. by 45½ in. (outside dimensions). The plates and frames are supported on two heavy side bars of round, rectangular or channel section, attached to the fixed head and the front standard, which contains the nut for the heavy central thrust screw (operated by hand or hydraulic gear) for compressing the follower plate against the plates and frames.

Between the thrust screw and the follower a thrust block is generally inserted, to make the thrust screw movement the shortest possible ; this speeds up the operation of opening and closing the press.

The closing pressure needs to be well above the prevailing operating pressure, as otherwise the joints will leak. With frames of 30 in. × 30 in. inside dimensions, at a maximum pressure of 60 lbs. per sq. in., the juice pressure will exert a force of :—

$$30 \times 30 \times 60 = 54{,}000 \text{ lbs.}$$

This juice pressure tends to open the press, and therefore the closing force should be at least 80,000 to 90,000 lbs. in order to tighten the joints properly; and the side bars and screw gear have to be designed to sustain such a force.

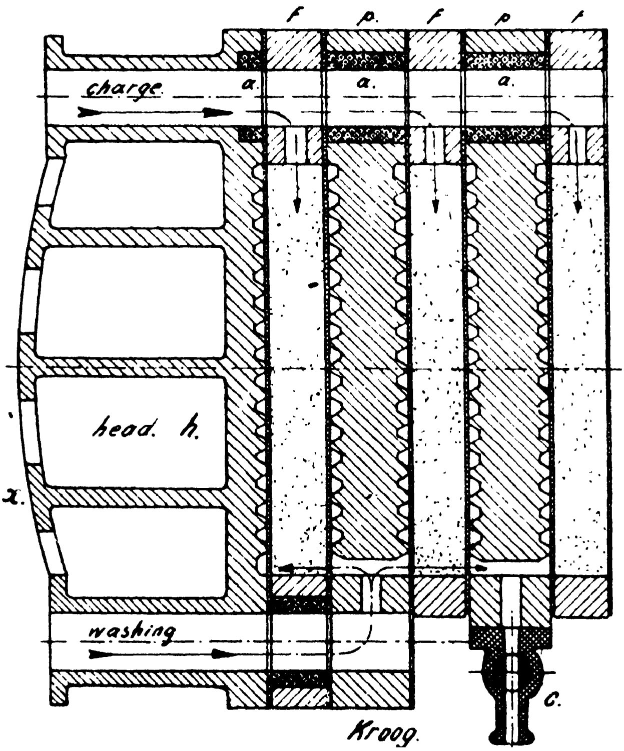


Fig. 406.—Plate and Frame Press.

The filter-cloths are hung over each plate, making a double joint on both sides. The juice is charged at the top of the press and enters the chambers, formed by the frames, the solids being separated by the cloth sandwiched between the plates and frames. The plates have grooved surfaces on both sides to drain off the clear filtrate. At the lower edge of the plates is a port (drawn in *Fig.* 406 on the right side) connected to a drainage cock *c*. Although plug cocks are extensively used, they require regrinding and any gritty material, which may enter the cock when a cloth is ruptured, impairs its tightening effectiveness. Flap cocks are therefore used to advantage for this service, only requiring the renewal of a rubber seat in the flap in case of wear.

At the lower left end is drawn the wash-out connexion, the sweet-water entering at every second plate and being forced through the cake, to be released by the cock of the adjacent plate. The drainage and non-drainage cocks for the washing process are sometimes made differently, the former having long and the latter short plugs to distinguish them. Washing presses of this type need thus to have an even number of plates.

Rubber collars *a* are inserted to form joints between the lugs of the plates and the frames, but this object can be equally secured by having the lugs of the same thickness as the plates and frames (the openings being of the required net diameter) and by applying cloth gaskets. As cloth material is always at hand, this will increase the reliability of the operating performance at a reduced cost.

Instead of the plain rubber collars *hydraulic lipped collars* are more efficient and will give a tight joint when the pressure in the press is applied. It will be obvious that these collars do not lose their elasticity as plain rubber rings do.

In *Fig.* 407 is shown a *Filter-Press Frame* of European make, as measured by the author. The juice enters at the opening *j* whereas sweet or wash-water is applied at *w* to the alternative plates only, as already mentioned.

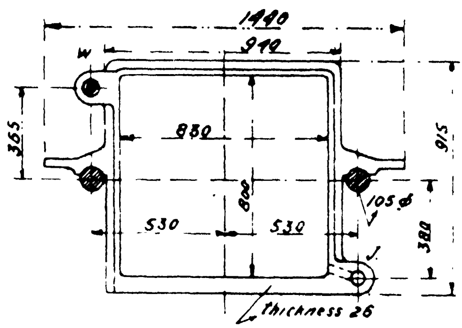


Fig. 407.—Filter-Press Frame.

When the opening for the mud at *j* gets clogged, the adjacent cast iron plates will receive the pressure only from one side and as these plates are not designed to withstand such pressure, they are liable to break; so the openings must be kept clear and be cleaned out each time the press stops work for mud discharge or replacement of cloths.

In *Fig.* 408 is shown the *Filter-Press Plate* for this same press ; it will be noticed that the handles for plates and frames respectively point in a different direction to allow the operators the more readily to grasp each plate and frame for sliding along the side bars. In case of long presses these side bars are sometimes supported in the middle, as deflection has to be avoided, since it will affect the proper tightening of the press.

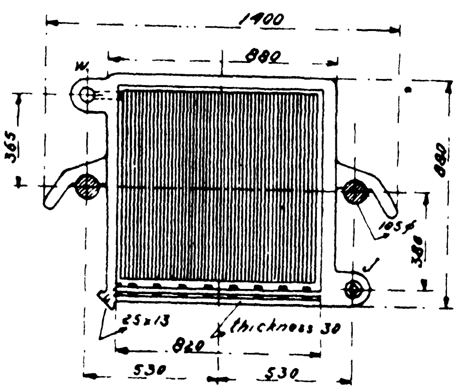


Fig. 408.—Filter-Press Plate.

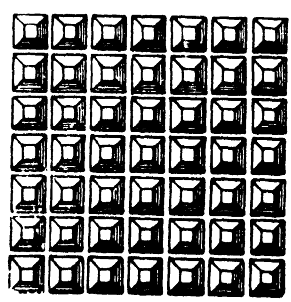


Fig. 409.—Pyramid Plate Surface.

Instead of the corrugated plate surface as shown in this figure, a *Pyramid Plate Surface, Fig.* 409, with the grooves horizontal and vertical or diagonal, will ensure better drainage and an increased rate of flow, about 30 per cent. in excess of that of a corrugated plate. For the washing performance the "pyramid" pattern plates also have an advantage over the older forms and their improved filtering is particularly noteworthy with thick liquids. The support of the filter-cloth on the truncated square pyramids of the plate is efficient, as lengthwise and crosswise stresses are equally distributed.

The *filtering surface*, *F.S.*, of a plate-and-frame press amounts to :—

$$F.S. = \frac{H \times B \times 2 \times N}{144} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (101)$$

in which : H = height inside frame in inches.
B = width inside frame in inches.
N = number of filter-press chambers.
$F.S.$ = filtering surface in sq. ft.

Similarly, the *Holding Capacity* of a plate-and-frame press amounts to :—

$$Q_h = \frac{F.S. \times t}{2 \times 12} \times C_f \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (102)$$

in which : t = frame thickness in inches = thickness of press-cake in inches.
C_f = coefficient of fullness = 0·75 — 0·98.
Q_h = holding capacity in cub. ft.

The fullness of the press frames depends upon the way the cakes are formed ; trapped air may have accumulated in the frames, for which reason air cocks are provided in many instances at the top of the frame, whereas some designers take it for granted that the air will escape with the juice through the filter-cloth.

A juice or mud inlet is provided at the top or as well as at the lower part of the frames. It does not matter where this juice inlet is located, as the formation of cake inside the frame will take place gradually, and since the resistance to the flow increases with the thickness of the cake, it will be obvious that the flow of mud will be directed to the thinnest spots in the cake layers.

The amount of cake to be formed, taken at 50 per cent. moisture as a normal figure, depends on the amount of impurities separated and the lime settling will have great influence. Four lbs. of cake produced, for each lb. of CaO added to the raw juice, may be accepted for overall calculations, and moreover :—

For defecation	0·75–1·75 per cent. on cane.
For sulphitation	1·00–2·65 per cent. on cane.
For De Haan carbonatation	5·00–8·00 per cent. on cane.

For batch carbonatation up to 13 per cent. on cane.

The weight of filter-cake is about 90 lbs. per cub. ft., the specific gravity of the wet cake being between 1·4 and 1·5 at 50 per cent. moisture.

The rate of flow is a figure subject to great fluctuations ; up to 3 gals. per min. and more per sq. ft. F.S. and down to 0·4 gals. or less have been reported. From practical work, the following requirements in F.S. may be quoted :—

Defecation without washing	60– 80	sq. ft./ton cane/hr.
Defecation with washing	85–100	,, ,,
Double filtration	100–120	,, ,,
Sulphitation with washing	110–125	,, ,,
First carbonatation ("De Haan")..	70– 85	,, ,,
Second carbonatation	40– 50	,, ,,
First carbonatation (general)	100–120	,, ,,
Syrup filtration	40– 60	,, ,,

Where filter-aids are used, the lower figures should suffice, but for new installations, where the filtering quality of the juices is not known, an extra allowance should be made.

In those cane sugar factories where sugar is refined, the following figures are quoted for liquors of about 60° Brix :—

Without filter-aids	300 sq. ft./ton sugar/hr.
With filter-aids	180 „ „

When applying diatomaceous earth (kieselguhr), about 0·09 lbs. per sq. ft. F.S. is required per hour.

The valves on the presses for the mud and sweet-water should be preferably of the full-way or gate-valve type, the disc being in vertical position, to avoid uneven wear.

A hopper is arranged under the presses for receiving the dumped press-cake. A large scroll conveyor, of 18 to 24 in. diameter, may be arranged underneath to transport the cake outside the factory. These scroll conveyors have rendered good service for wet cake, but for dry cake a drag carrier with two chains and wooden slats, running at about 100–120 ft. per minute, is to be preferred.

A drip pan is laid under the plates and frames, when the filter-press is in use to collect the inevitable drippings. A double juice gutter receives respectively the clear and the cloudy juice. The drip pan slopes gently towards the cloudy juice gutter. Sheet iron tumblers are available, to be placed individually over the clear juice section, so as to return cloudy juice to its corresponding gutter.

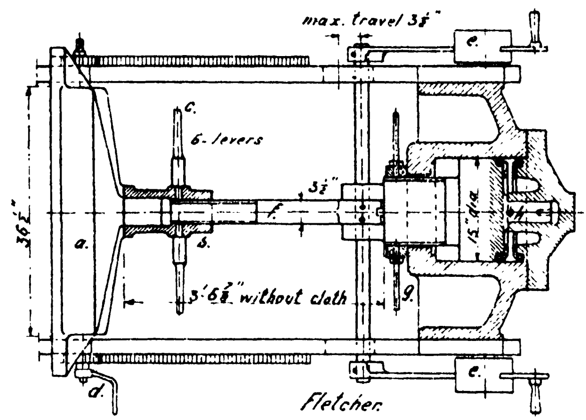


Fig. 410.—Hydraulic Closing Gear.

It is good practice in defecation factories to return the clear filtrate to the subsiders, and a thorough separation of clear and cloudy filtrate is therefore not required. In case of carbonatation factories, where all the juice is filtered, provision for separation has, of course, to be made.

In *Fig.* 410 is shown a *Hydraulic Closing Gear* which will allow quicker press operation with less labour. The follower *a* is tightened by the nut *b*, which carries six levers *c* for a tommy bar of the socket type. The thrust screw *f* is attached by a hinge to the hydraulic ram and can be brought into position or raised by two levers with counterweights *e*, convenient stops being provided for both extreme positions. When the follower plate has been tightened by the tommy bar, the hydraulic pressure is then applied and the press will become tight. The hydraulic controlling valve is on top of the front standard with a lever for easy operation. When the pressure has been applied, the ram nut *g* is tightened, and the ram does not then require to remain under pressure. The ram and cover have U-leathers with special guard rings, so that the hydraulic pressure may have access to the inside of the leather.

When untightening the press, the hydraulic pressure has to be re-applied for untightening the nut *g*. The ram travel is kept within the smallest practical limits, to reduce the accumulator capacity, and the water consumption.

The follower plate can be withdrawn when the thrust screw *f* has been raised, by means of the hand gears *d*, acting on spur racks alongside the side bars.

For very large presses, especially when there are several, hydraulic means for tightening, also for opening and closing, will result in a considerable saving of labour. For single units or a small number of presses, a *hand hydraulic tightening gear* is very convenient, each press being a self-contained unit. The tightening pressure can be observed from a manometer and over-straining of the press, which is possible with powerful mechanical gear, is thus avoided.

The *filter-cloth* used is generally bought hemmed and shrunk. Different materials have been employed for this cloth, cotton and hemp being mostly used, but better fibres are tucum (the palm *Bactus setosa*), banana (*Musa chinensis*) and aramina (*Urena lobato* L.).[1] Filter paper is also used. The tensile strength of the cloth per inch width is about 230 lbs. when dry and about 130 lbs. when wet. The weight is about 0·12 lb. per sq. ft.

Filter-cloths have to be cleansed in washing machines, to remove the adhering dirt, especially at the end of the crop. Wooden or metallic washing drums, thoroughly smooth on the inside, can be used. Soda is sometimes used, but very effective rinsing is afterwards required, as otherwise the cloth fibre will be damaged. The drying of the cloths can be done in any well-ventilated space above the boilers, but direct sun rays should be avoided.

5.—Washing Filter-Press Cake.

Filter-press cake contains about 50 per cent. moisture on its weight and the amount of sucrose, carried in solution, will equal the sucrose in the filtrate. Thus if the filtrate carries 12 per cent. sucrose per 100 water, the cake water will carry nearly the same amount, or with 50 per cent. water in the cake, this will yield 6 per cent. sucrose in the press cake.

It is obvious that this sucrose should be removed as completely as possible and this necessitates the sweetening or washing of the cake. This washing can be done either within or without the filter-press, the second method being considered the best; but the diluted mud from the presses has to be filtered a second time and thus double filtration is required.

Within the press, the juice pressure is followed by steam and sometimes air, before the wash-water is applied. As all fluctuation of pressure should be avoided, some manufacturers provide a special gear for operating the valves in consecutive order, the process being effected by slowly rotating a hand wheel on the cam shaft of this gear. To reduce the amount of sweet-water, the cake should contain as little moisture as possible, before applying the water.

The washing of the cakes can be done in two ways; the first to be considered, the author has termed *Split Washing* as shown in *Fig.* 411. The cakes are formed until a split or open space of about ¼ in. remains between the two layers. To control the filling of the press, one frame has a ¼ in. less thickness than the others, and as soon as this chamber ceases to discharge any filtrate, the cake has completely filled it, whereas the others are assumed to have still a centre split of about ¼ in. Steam and/or compressed air is now admitted to drive out the juice, after which the wash-water is forced through. Any thorough admixture of the wash-water with the cake water does not take place and it is to be noted that the first run-off will have about the same Brix

[1] See the article of F. W. FRAISE, *F.A.S.*, 1933, p. 252.

as the previous filtrate, but after a while a sudden drop to about 2 to 3° Brix will take place and the washing performance should then be stopped. The subsequent application of steam or compressed air will dry the cake.

The position of the drainage cocks is shown in *Fig.* 412, all of them being open.

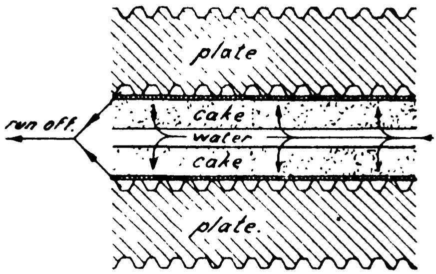


Fig. 411.—Split Washing.

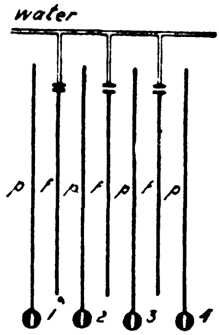


Fig. 412.
Position of Drainage Cocks.

In *Fig.* 413 *Transverse Washing* is shown, the wash-water being forced from one plate to the next, through the full cake. In *Fig.* 414 the position of the drainage cocks is shown, the uneven numbers being open and the rest closed. Such a washing press obviously has to have an even number of chambers.

Sometimes cracks are formed in the cake, which will reduce the effectiveness of the washing performance. With split washing, de-sugarized mud can be mixed with the wash-water and so fill up these cracks. As to the respective merits of these methods, there is a difference of opinion; but transverse washing is generally considered the best. Split washing can nevertheless be applied in those presses where there is no special washing connexion provided.

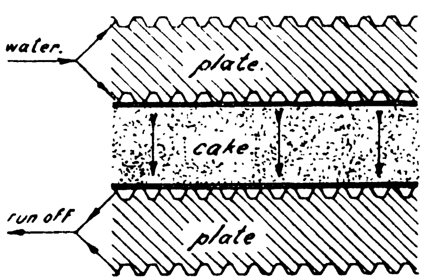


Fig. 413.—Transverse Washing.

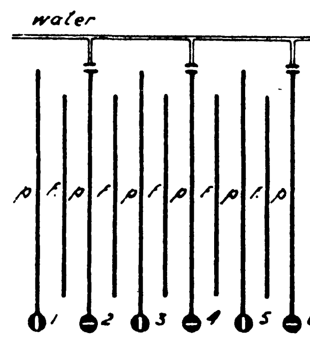


Fig. 414.—Drainage Cocks with Transverse Washing.

Theoretically, the effect of the washing can be explained as follows : The cake C is composed as follows :—

$$C = S_i + S_{ii} + W$$

being : S_i = insoluble matter per cent. cake.
S_{ii} = soluble matter per cent. cake.
W = water per cent. cake.
W_w = wash-water applied per cent. cake.

It will now be apparent that the wash-water will only affect the soluble matters, and the amount of these remaining in the cake is thus :—

$$S_{ii} \times W \div (W + W_w) \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (103)$$

and, when the amount of sweet-water discharged is equal to the amount of wash-water applied, the latter has carried along :—

$$S_{ii} \times (1 - [W \div (W + W_w)]) \quad \ldots\ldots\ldots\ldots \quad (103a)$$

In practice the closest results are obtained by mixing the cake outside the presses with the wash-water, this secondary mud being then re-filtered. The washing lasts about 15 to 20 minutes, the amount of wash-water being taken as about 0·9 the weight of the wet cake, and a final sucrose content of 1 to 2 per cent. is obtained.

The wash-water should be applied with a temperature of about 140°F. (60°C.) and when alkaline it should not be used for imbibition, as this will lead to yellow coloration. In such a case it can be used for lime mixing or lime slaking.

6.—Leaf Filters.

For almost continuous operation, easy de-sugarizing and cake removal, leaf filters, where a granular cake can be formed, have been successfully applied. The filtering surface for the same amount of scums to be treated is smaller than with plate-and-frame presses, thanks to the increased rate of flow.

The leaves are made of bronze, and consist of a frame, covered with strong wire gauze, over which the filter-cloth is laid.

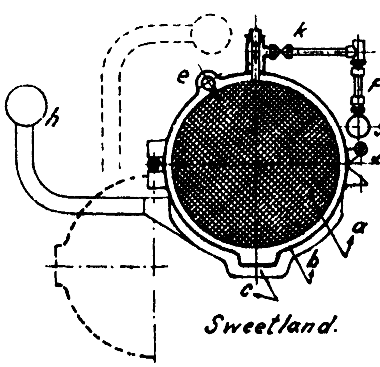


Fig. 415.—Fixed Leaf Filter.

A *Fixed Leaf Filter* is shown in *Fig.* 415, this being of a unique design which has found use in carbonatation practice in cane and beet sugar factories, as well as in refineries for filtering liquors with diatomaceous earth. The construction comprises a cast iron cylinder, of which the bottom half can be opened by the so-called clam shell arrangement, this bottom half *b* being swung on pivots, balanced by a counterweight *h*, as indicated in the dotted lines. The pivot bolts *j* are tightened or loosened by rotating a shaft, carrying as many small eccentrics as there are bolts.

The leaves *a* are arranged at 3 in. centre distances guided in the upper half of the shell, so as to remain perpendicular to the filter axis. As the leaves have to be removed from below, sufficient height above the floor has to be arranged for.

The juice is drawn off in the upper half, each leaf having a separate connexion, which can be closed by a valve *k* in case of cloudy run-off, and provided with an inspection gauge glass tube *f*, discharging into the pressure-free main *g*.

Between each adjacent couple of leaves a spray nozzle is arranged on a spray pipe *e*, which can be rotated for about 90° from the outside of the shell. The pressure of the wash-water has to be about 60 lbs. per sq. in. in order to be able to wash the entire leaf surface.

The feed under pressure is through the bottom channel *c*, the fixed pipe connexion being in the upper half. A baffle is arranged in this channel for equal distribution. The mud, which cannot be delivered as dry as is the case with plate-and-frame presses, is sluiced down through the bottom connexion at *c*.

As the discharge of each leaf is at the top, it will be obvious that the liquid contained in the shell, after the filter operation is stopped, cannot be removed by compressed air, for which reason the leaves have an inside pipe connexion, reaching close to the bottom.

The performance of these filters has proved a very good one, 72 sq. ft. F.S./ton of cane/hr. being sufficient for carbonatation work. About 1·4 lbs. of dry cake per sq. ft. F.S. per hour have been separated, against about 0·9 lbs. for plate-and-frame presses.

For defecation factories from 30 to 40 sq. ft. F.S./ton cane/hr. is required and for syrup filtration the same areas have to be applied.

The rate of flow is generally from 15 to 25 per cent. higher than with plate-and-frame presses, but for washing 50 to 70 per cent. more wash-water is required.

The filters are built in units up to 700 sq. ft. F.S., having 48 leaves on the average and 0·06 cub. ft. of cake will be produced per average cycle of 90 minutes per sq. ft. F.S. The thickness of the cake varies between ¾ in. and 1 in.

The initial cost of a leaf filter is high compared with that of a plate-and-frame press, but the expense for cloth and labour is much less.

An improvement has been sought in the *Revolving Leaf Filter*, having circular leaves. rotating on a central shaft, through which the juice is drained. Good operating results have been reported, but the individual juice drainage cannot be observed and the eventual replacement of a leaf is troublesome.

To overcome this inconvenience, leaves have been designed, composed of several panels or segments, and also of a horse-shoe type, so that each can be slipped over the shaft.

With such a full leaf filter, having 530 sq. ft. F.S. and 30 full circle leaves, the filtering cycle has been completed in 45 minutes, at from 20 to 40 lbs. per sq. in. juice pressure, and giving an average rate of flow of 11 gals. per sq. ft. F.S. for the complete cycle, this being a favourable performance.

A revolving leaf filter of recent special design, known as the *Auto Filter* is shown in *Fig.* 416; it is used in vegetable char refineries on white sugar liquors, and on carbonatation sludge. It comprises a cylinder *a*, opening on the line *b*, and it is a pressure filter in which the filtering elements *c* are carried on a rotating frame *l*, as shown at *d*. Half the leaves discharge through the left hand trunnion *e* and the rest through the right hand trunnion not shown on the drawing. The filtrate from each leaf passes through a separate sight tube *f*, and any individual leaf can be shut off, should the filtrate run cloudy, by a valve *g*, without stopping the filter or in any way interfering with the flow from other leaves. The clear run-off is discharged at *h*.

The shaft is supported on roller bearings *e* and driven by a worm drive *k*, rotating slowly at about 3 to 5 r.p.m., thus producing sufficient agitation to prevent the settling of heavy solids and assuring an even thickness of cake, which results in a high filtration rate and clarity of the filtrate. The filter is charged at *j* and is provided with an overflow connexion for re-circulation, where this may be desirable. The filters are supplied in various sizes up to 900 sq. ft. filtering surface and are built for operating pressures up to 50 lbs. per sq. in.

The filter has to be opened only for renewal of the filter-cloths, sluicing and washing devices being provided for continuous operation. There is no mechanical wear on the cloths, as scrapers and kindred details are absent and therefore the life of the cloth is long. The manufacturers state that the filter will operate without opening, for periods of several months.

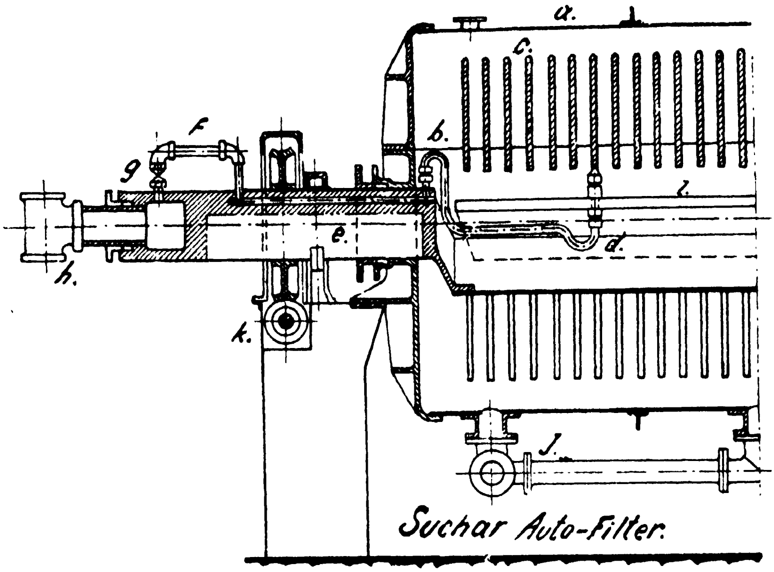


Fig. 416.—Revolving Leaf Filter.

The cake is discharged as a sludge, which can be run to the sewer or re-introduced into the system. When regenerating or revivification is required, it has to be dried before it can be sent to the revivification ovens. This drying is sometimes done in a small plate-and-frame press or by other means.

The amount of wash-water required is larger than with the plate-and-frame presses, but the upkeep and labour costs are low.

7.—Rotary Drum Vacuum Filters.

In the beet sugar industry these filters have been used with good operating results, and quite a considerable number are now used in cane sugar factories for the scums or mud of defecation factories. As a filter-aid fine bagacillo of about 0·12 in. size (0·3 mm.) has to be employed to make possible this unique method of filtration.

A recent interesting construction of the *Rotary Drum Vacuum Filter*, shown in *Fig.* 417, has been applied in several cane-growing countries for *cachaza* (defecation scum) filtration. The essentials of this design comprise a drum *a*, mounted on a shaft, which is supported in ball or roller bearings. This drum consists of two cast iron discs, which carry 24 plate casings, having brass plates with 625 perforations per sq. in. of 0·02 in. diameter on the outside periphery, the latter having little adherence for the defecation mud. The perforated brass plates can be easily replaced, as may be required.

Inside the drum, each casing or chamber is connected by one or two pipe connexions to a rotating valve mechanism *b*, thus making a through connexion between the drum chamber and one of the receptacles *m*, which are kept under vacuum, one for cloudy and the other for clear filtrate or wash, and having the discharge *p* at barometric height or connected to a pump suction.

The lower part of the drum is submerged in a semi-circular tank of small capacity, in which the scums to be filtered are charged, it being provided with an overflow at *l* and a rocking agitator *f* close to the bottom of the tank, suspended by levers *g* and operated by eccentrics from a shaft *e*.

The operation now is as follows : When a filtering chamber is submerged in the mud or scum, the rotary valve connects to a vacuum of about 10 in. and a cake is formed, the first filtrate being cloudy for about 10 seconds and being discharged to the right hand receptacle *m*. As soon as this coating is completed, the valve is connected to a 20 in. vacuum and the clear juice outlet to the left hand receptacle *m*. When the casing emerges from the scum, a good cake has been formed which is readily dried through the prevailing

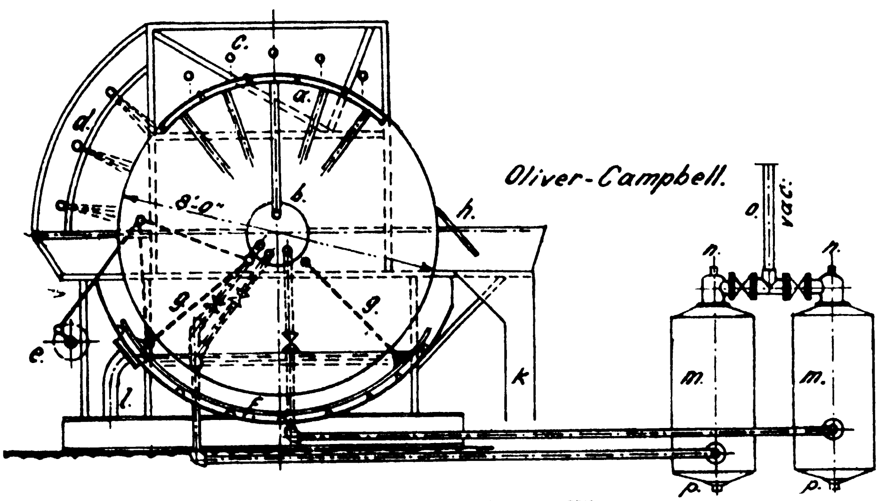


Fig. 417.—Rotary Drum Vacuum Filter.

vacuum. Wash-water is now finely sprayed at *d* and dripping pipelines *c* complete the washing, resulting in an efficient de-sugarizing of the cake. As soon as the casing has passed the last dripping, the vacuum is disconnected and the cake is removed from the filter-cloth by a rubber scraper of special design in the hopper *k*, which discharges on a conveyor or by other means, for removal outside the factory. A weighted brass trowel is provided to iron out cracks in the cake.

The receptacles *m* are provided with floats, attached to the regulating valves *n*, which are connected by a vacuum line *o* to a barometric condenser and a vacuum pump. The condenser serves the purpose of condensing the vapours due to flashing, and of reducing the air volume to be extracted.

The filter is made with a drum diameter of 8 ft. and a width from 3 to 12 ft., revolving at a speed of 1 revolution every 3 to 9 minutes. For driving the filter proper, about 2 h.p. is required, while the auxiliaries for a 300 sq. ft. filter require about 25 h.p. The vacuum pump displacement is given as

1¼ cub. ft. per minute for each sq. ft. of filtering surface. The amount of bagacillo required is from 15 to 30 per cent. on total dry matter in the cake, to make filtration practicable for defecation scums. The ratio between cloudy and clear filtrate is as 1 : 5 to 1 : 2. The filtering capacity depends on the amount of solids in the scums, and the average is quoted as :—

2·7	tons cane/24 hrs.	per	sq. ft.	F.S.	for	3	per cent.	solids.
5·3	„	„	„	„		6	„	
9·0	„	„	„	„		10	„	

The cake contains about 80 per cent. moisture, the juice temperature being above 176°F. (80°C.), whereas the rate of flow in beet houses is reported to be about 16 gals./sq. ft. F.S./hr. and in cane sugar factories from 10 gals. upwards. The wash-water required is about 150 per cent. on wet cake and a sugar content of 0·8 to 1·7 per cent. on wet cake has been achieved and about 0·15 short tons cake will be delivered per sq. ft. F.S./hr.

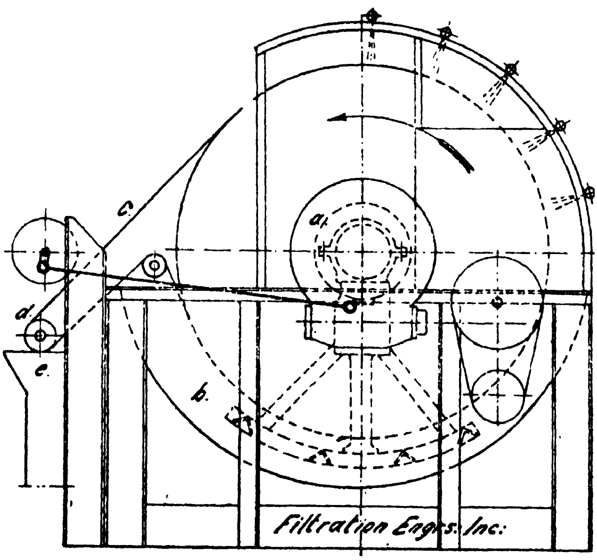
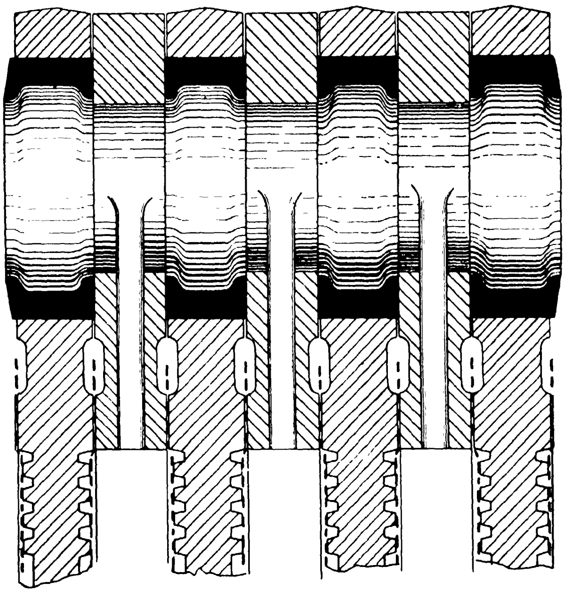


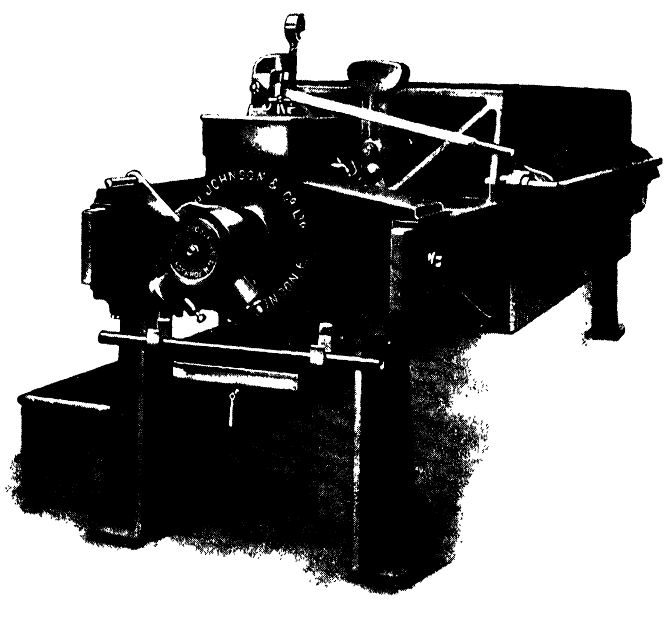
Fig. 417A.—String-type Drum Vacuum Filter.

Another type of drum filter, which has found a wide application in the chemical and mining industries, is the *Rotary Drum Filter with String Discharge*, shown graphically in *Fig.* 417a ; this is now also applied for *cachaza* filtering. The drum is composed of a number of shallow compartments to reduce the volume of free air to be removed, when brought under vacuum. The compartments have drainage members of spiral wire, which can be easily cleaned or replaced. A revolving vacuum valve *a* provides the vacuum connexion to the submerged compartments in the tank *b* below and a fine filter-cloth can be used on these drum filters, as they do not have scrapers, but the cake formed is discharged by a great number of strings *c*, which lift off the cake from the drum in a continuous layer. The return roller *d* causes the cake to fall into the hopper *e*, from which it is removed outside the factory. It will be obvious that, with this unique design, the filter-cloth can be laid around the drum without further attachment than that caused by the strings, and even when the drum periphery is not a true circle, it will not affect the filtering performance. A thin cake from $\frac{1}{16}$ in. up to 2 in. can be produced by this filter, depending upon the nature of the cake. Efficient spray washing of the cake is assured and cloudy filtrate is not withdrawn, as cloth of fine mesh can be used.

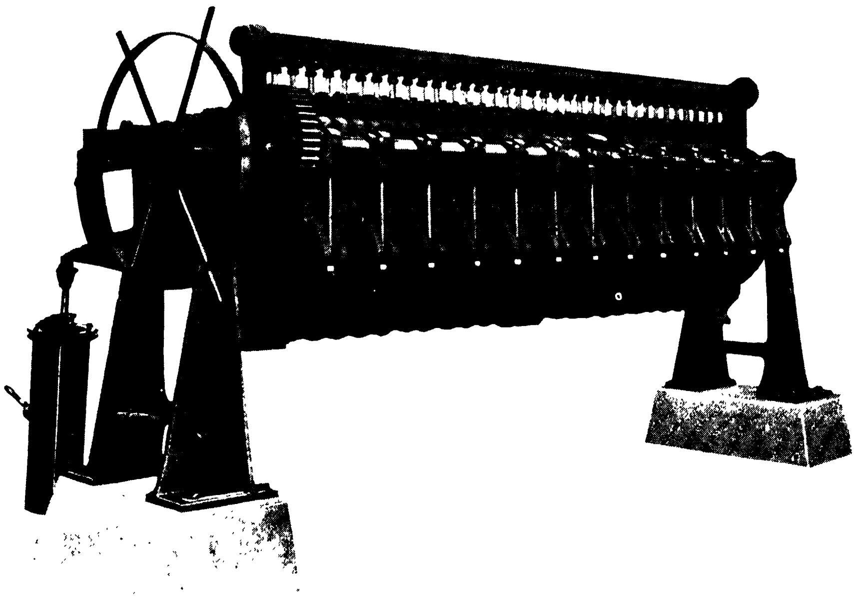
These rotary filters occupy the smallest space of all filters known to-day, as the filtering area of the drum need only be about 10 per cent. of that of a plate-and-frame press, and a very clean installation results.



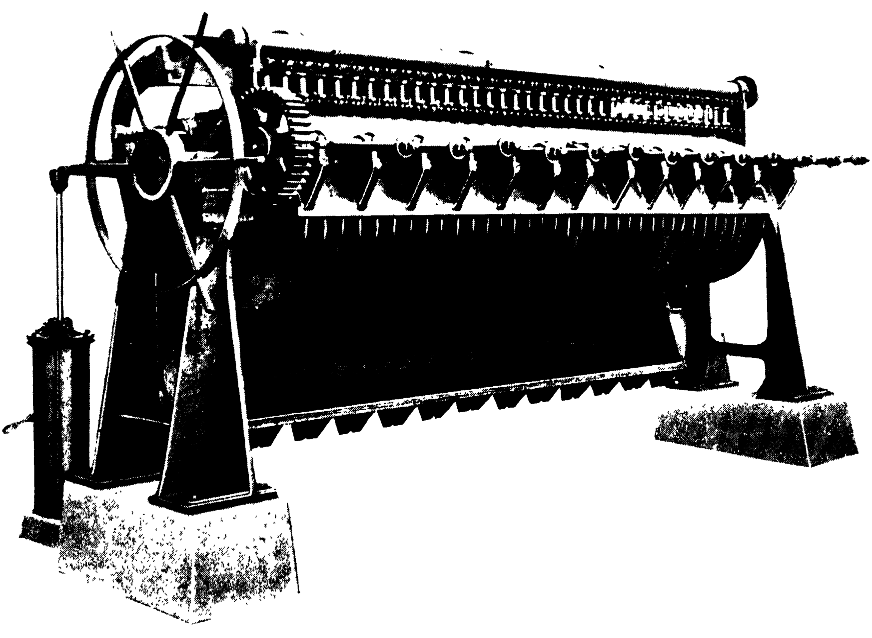
Hydraulic Lipped Collars for Filter-presses.
(*S. H. Johnson & Co., Ltd.*)



40 in. Filter-Press with Hand Hydraulic Tightening Gear.
(*S. H. Johnson & Co., Ltd.*)



Sweetland Filter with Hydraulic Closing Gear.
(*Oliver-United Filters, Inc.*)



Opened Sweetland Pressure Filter, showing Leaves and Clam Shell Arrangement.
(*Oliver-United Filters, Inc.*)

8.—Thickeners.

As already mentioned, filters will be relieved of much work or excess rate of flow, if the mud or scum is concentrated; and so the filtering performance or the filtering cycle will be considerably shortened. Continuous clarifiers deliver scums of a porridge-like consistency and further concentration may not be needed. But such conditions cannot be produced in all instances, and the *Concentrator* or *Scum Thickener* has a definite purpose. In *Fig.* 418 is shown an ingenious design of such a thickener, it being composed of two upright tanks with conical bottoms having a shaft with mixing paddles *g* at the lower end.

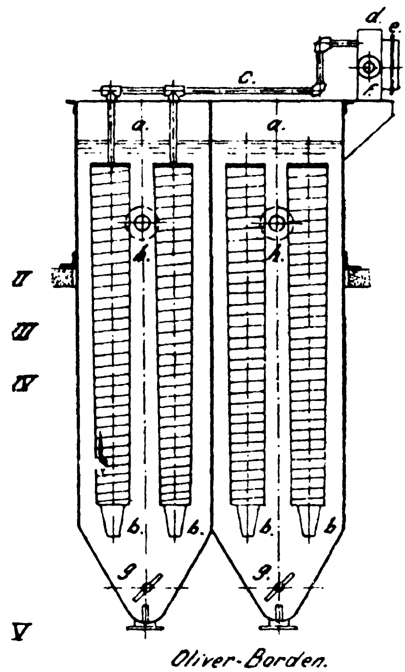


Fig. 418.—Concentrator or Scum Thickener.

In the tanks are arranged vertical perforated tubes, slightly conical towards the lower end, encased in filter-cloth, which is held by spirally-wound galvanized wire. These tubes are connected in groups of two or four to a rotating valve *d* by means of a pipe connexion *c*, the rotary valve being connected to float receptacles as shown in *Fig.* 417. The rotary valve is driven by a ratchet wheel *e*. The operation comprises two stages; first a vacuum operation for about 12 minutes and then the dislodging (by compressed air at about 30 lbs. per sq. in.) of the formed cake, which falls to the bottom of the tank, whence the concentrated mud can be removed continuously. The tanks are charged at *h* and the filtering tubes have necessarily to be completely submerged.

For carbonatation scums, these thickeners have been used to full advantage; and the capacity, depending upon the system, is from 1 to 3 tons of cane per sq. ft. thickener surface. In beet sugar houses in Europe and the U.S.A., these thickeners have been successfully applied in combination with rotary vacuum filters.

The power consumption for a thickener of about 300 sq. ft. amounts to 1·5 h.p., without considering the power input for the juice and vacuum pumps or the compressor.

CHAPTER XXI.

EVAPORATORS.

The juice in the cane, which is nothing more than an impure sugar solution, has a density equal to about 20 parts sugar by weight in 100 parts juice. The density is measured by a hydrometer, having generally an inserted scale, graduated according to Balling or Brix, which is based on the per cent. sugar in pure solution; and thus the sugar content can be read directly from this Brix spindle. Another scale division used is that of Baumé, but this was first intended for salt solutions, and a correction has to be made when it is used for sugar. The sugar content of impure solutions is not indicated by the Brix readings, as it has to be multiplied by the true purity coefficient, this being about 85 per cent. for common cane juices. By means of maceration and imbibition at the milling stage, the cane juice is diluted to about 15° Brix before it enters the clarification section of the factory.

The juice, nevertheless, has to be concentrated to a nearly solid mass, by evaporating the water which holds the sucrose in solution, so as to crystallize out the latter in the form of commercial sugar.

This concentration is done in two stages for practical reasons, the first stage (Evaporators) being a continuous one from 15 to about 60° Brix, thus well below the crystallization point, which for cane juice lies around 78 to 80° Brix. The second stage (Vacuum Pans) is an intermittent or batch performance, being the crystallization stage proper. The first is called the evaporating and the second the boiling performance. The above-mentioned coefficient of purity has no direct bearing upon the mechanical performance of evaporation, although it may occasion difficulties in the production of a good commercial sugar on an economic basis and, also, it may be the source of incrustations on the heating surface of the evaporators.

The underlying principles are only referred to so far as required for the design of the apparatus. For more theoretical considerations, the standard works of CLAASSEN, HAUSBRAND, ABRAHAM, *et al.* should be consulted.

1.—Principles of Evaporation.

As explained, the evaporators will only eliminate a part of the water in the juice, the density or concentration being raised from about 15 to 60° Brix.

To know the *quantity of water* to be evaporated, the basic fact has to be borne in mind that the quantity S of solids in solution within the juice will remain unchanged, whatever concentration may exist below the crystallization point, and the general formula for all concentrations can be written :—

$$S = W_j \times B \quad \ldots\ldots\ldots\ldots \quad (104)$$

where S = weight of solids in solution in lbs. or kg.
W_j = weight of the juice at the desired concentration in lbs. or kg.
B = corresponding Brix of the juice.

For two concentrations of the same juice, having densities B and B_i, it will follow that :—

$$S = W_j \times B = W_{ji} \times B_i \qquad \text{or :}$$
$$W_{ji} = W_j \times B \div B_i \quad \ldots\ldots\ldots\ldots \quad (104a)$$

W_{ji} being the weight of the juice at the concentration B_i, and the quantity

W_e of water evaporated, in lbs. or kg., obviously amounts to :—

$$W_e = W_j - W_{ji} = W_j \times (1 - B \div B_i) \quad \ldots\ldots\ldots \quad (105)$$

If the clarified juice entering the evaporator has a density of 15° Brix, whereas the syrup leaving the evaporator has a density of 60° Brix, a quantity of water has been evaporated equal to :—

$$W_j \times (1 - 15 \div 60) = 0{\cdot}75\ W_j$$

and a 75 per cent. evaporation has been obtained. This 75 per cent. is considered a good performance, the final density of the syrup lying between 55 and 65° Brix, dependent on the density of the clarified juice and the evaporative capacity of the apparatus.

The heating of the evaporators is effected by steam; open pan evaporators, heated by a direct fire underneath, are only found in small scale factories for native consumption sugars in some cane-growing countries.

Generally, the juice enters the evaporator below the prevailing boiling temperature within the evaporator, and the juice temperature has thus to be brought to this level, requiring a quantity H_i of heat :—

$$H_i = W_j \times (t_i - t) \times C_j \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (106)$$

in which : t_i = boiling temperature of the juice under the prevailing pressure within the evaporator.

t = juice temperature, when entering the evaporator.

C_j = specific heat of the juice.

As this formula suits equally the British and the metric systems, the corresponding values are : H_i in B.Th.U. or calories.

W_j in lbs. or kg.

t_i and t in degrees F. or degrees C.

C_j in B.Th.U./°F./lb. or cal/1°C./kg.

Secondly, the quantity W_e of water has to be evaporated and as the liquid heat has already been accounted for in evaporation, only the latent heat of the emerging vapours has to be supplied, thus :—

$$H_{ii} = W_e \times L \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (106a)$$

The latent heat L is taken from any Steam Tables and it varies from 1021 B.Th.U./lb. for 26 in. vacuum to 928 B.Th.U./lb. for 30 lbs. pressure (respectively 568 and 516 cal./kg.).

As a matter of fact, the boiling temperature for a sugar solution is higher than for pure water, so the vapours produced are generally slightly superheated, a point that has been neglected in formula (106a), as being of only secondary importance.

The total heat required will thus be :—

$$H = H_i + H_{ii} = W_j\,(t_i - t)\,C_j + W_j \times F \times L =$$
$$W_j\,[(t_i - t)\,C_j + F \times L] \ldots. \quad (106b)$$

F being the factor of evaporation = $(1 - B \div B_i)$.

The formula applies as much for a single effect as for the first body of a multiple effect evaporator, but in the latter case the term $(t_i - t)$ acquires a negative value for the subsequent bodies, the juice thus entering with a higher temperature than corresponds with the pressure inside the body or vessel, and hence gives rise to flashing.

The *Heating Steam* required to produce the heat H obviously will amount to :—

$$Q_s = H \div L_s \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (107)$$

Q_s being the quantity of steam in lbs. or kg. and L_s the latent heat of this heating steam (thus not of the vapours produced by this steam). The liquid heat of the heating steam has no value for the performance, as it will leave the evaporator with the hot condensate.

It should be recollected that there are always radiation losses, and the actual steam consumption will be 3 to 10 per cent. higher, depending upon the effectiveness of the insulation covering the evaporating bodies and the connecting steam, vapour and juice lines. For the first body of a quadruple effect, the following conditions may be assumed :—

t_i = 215°F.
t = 185°F.
C_j = 0·895.
L = 968 B.Th.U./lb. of 1 lb./sq. in. pressure.
L_s = 960 B.Th.U./lb. of 5 lbs./sq. in. pressure.
F = (1 — 15 ÷ 20) = 0·25.
H = 268·85 B.Th.U. per lb. juice.
Q_s = 0·28 lbs. steam per lb. juice.

As each lb. of juice is subject to an evaporation of 0·25 or 0·25 lbs. water, it requires 0·28 lbs. exhaust steam, without considering eventual radiation heat losses.

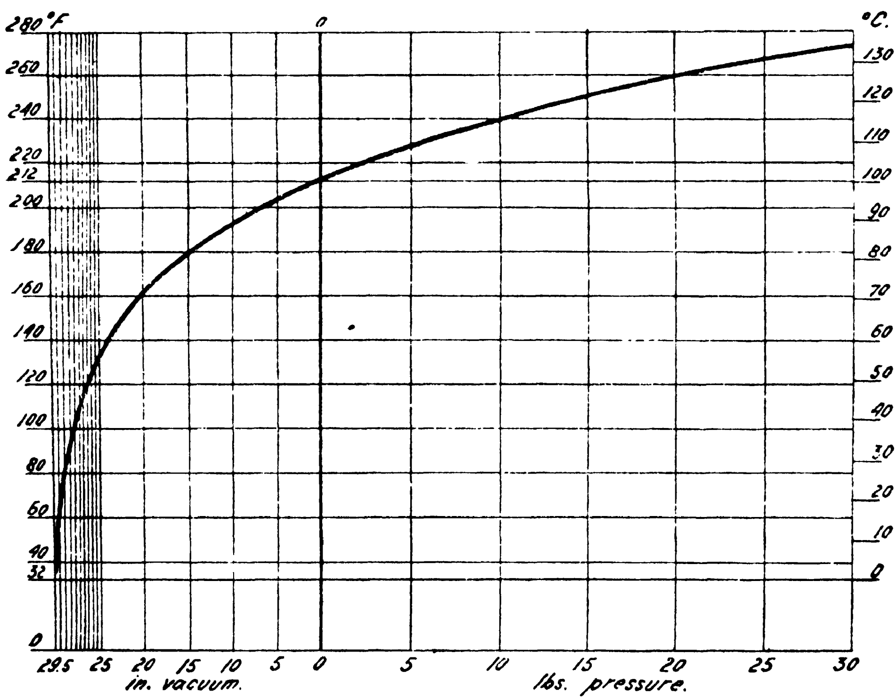


Fig. 419.—Pressure-Temperature Diagram.

In *Fig.* 419 is shown the *Pressure-temperature Diagram* for saturated steam from vacuum up to 30 lbs./sq. in. gauge pressure. As 27 in. vacuum may be considered the maximum obtainable in average cane sugar factories, the equivalent temperatures range from 115° to 274°F. The graph explains, also, why evaporation under vacuum has found such a wide application, as the greatest temperature difference is within this vacuum range. Pressure evaporation is only adopted to a limited extent.

As any solution boils at a higher temperature than water, the *Temperature Rise Graph* for pure sucrose solutions of varying densities is shown in ***Fig.* 420**, the boiling to be obtained under atmospheric pressure ranging between 10 and 92 per cent. sucrose in solution. **RAOULT** has found that this temperature rise applies equally to boiling under vacuum or under pressures above the atmospheric one, but more recent investigations have indicated small differences. As the juice is an impure sugar solution, slight differences may occur, but these are negligible.

In the evaporation performance only the boiling of the thick-juice or syrup concerns us, the temperature rise being 5·4°F. for a density of 60° Brix. It will be noted that the rise in temperature becomes of more importance at the boiling stage than during the evaporating performance.

The *Hydrostatic Pressure* of the juice column will also cause a rise in temperature at the lower juice levels. A column of water, 1 ft. high, exerts a pressure of 0·432 lbs. per sq. in. on its base ; and for syrup of 60° Brix, having a specific gravity of about 1·289, it will be 0·557 lbs. per sq. in. When this syrup is boiling under a vacuum of 26 in. mercury column, the vapour pressure

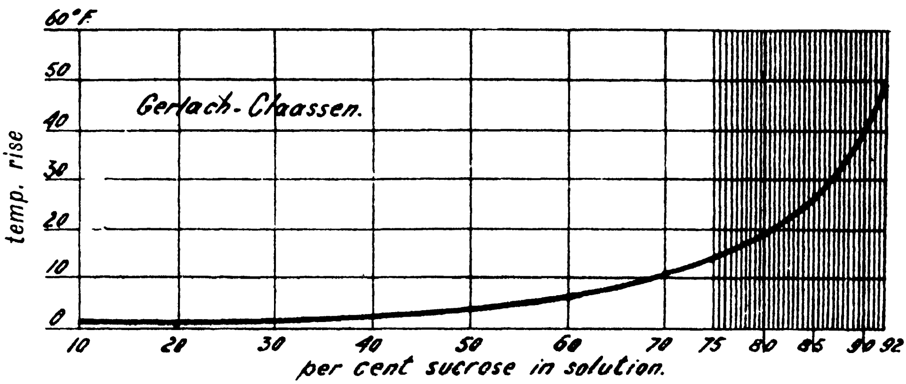


Fig. 420.—Temperature Rise Graph.

at the top of the syrup column amounts to 1·954 lbs./sq. in. absolute pressure, whereas at the base of the 1 ft. column, a pressure exists of 1·954 + 0·557 = 2·501 lbs./sq. in. The corresponding boiling temperatures for the top and base of the column are respectively 130·8 and 140·4°F. The danger of overheating is not paramount, but the evaporation and heat transmission at the lower levels are greatly reduced. When measuring the vapour temperatures it should be recollected that they are generally slightly superheated, so do not correspond exactly with the equivalent pressures for saturated steam.

The heat transmission in an evaporator is of varying magnitude, depending upon incrustations and incondensible gases, which render the calculation at fault. The increased density of the juice also reduces its value. From the author's experience, the following values have been found in practice for multiple evaporators :—

K = 200–400 B.Th.U./sq. ft. heating surface/hr./1°F. temp. difference.
K = 1000–2000 cal./sq. m./hr./1°C. temp. difference

and it should be noted that for the first bodies this value will be higher and for the last ones less. The lower values represent everyday average performance, but the higher ones are obtained with special attention to design and operation. A clean heating surface on both sides, proper juice circulation, and film formation are the deciding factors.

It is established practice to express this heat transmission also in lbs. water evaporated per sq. ft. H.S./hr., and from HAUSBRAND the following figures are quoted :—

Simple effect	14–16 lbs./sq. ft.	70–80 kg./sq. m.
Double effect	6–7·5 ,,	30–36 ,,
Triple effect	4–5 ,,	20–25 ,,
Quadruple effect ..	3·7–4·5 ,,	18–21 ,,
Quintuple effect....	3–3·7 ,,	15–18 ,,

For conversion the following equivalents are given :—

1 lb./sq ft. = 4·874 kg./sq. m.
1 kg./sq. m.= 0·206 lbs./sq. ft.

The above figures may be taken as very conservative and are obtainable under very stringent conditions. The author has installed vessels in quadruple effect, with an overall evaporation figure for the whole crop of 8·3 lbs./sq. ft. (40·4 kg. per sq. m.). In one instance, the author was called to investigate a 20,000 sq. ft. quadruple effect heated by exhaust steam, which actually had an average evaporation figure during the whole crop of over 10 lbs./sq. ft. (48·7 kg./sq. m.), this result being achieved by the factory engineer by closing the bottom end of the downtake and withdrawing the syrup and concentrated juice from there. From Java average figures have been reported (for 1926) as follows :—

Triple effect	31 kg./sq. metre	6·4 lbs./sq. ft.
Quadruple effect	22 kg./sq. metre	4·5 lbs./sq. ft.
Quintuple effect	18 kg./sq. metre	3·7 lbs. sq. ft.

Perhaps the most outstanding feature in steam economy in the sugar factory has been the invention of RILLIEUX in Louisiana about a century ago, the *Multiple Effect Evaporator,* which is now used all over the globe.

The underlying principle consists in a large heating surface with a small temperature difference. It will be obvious that with a smaller heating surface and a proportionately larger temperature difference, the same evaporation effect could be obtained, but RILLIEUX achieved his object in a unique way, by using the vapours from a previous vessel for heating the next one in series, thus greatly reducing the required steam consumption. As to the number of consecutive bodies employed, there are simple, double, triple, quadruple and quintuple effect evaporators. Sixtuple evaporators for use in cane sugar factories have not come to the author's knowledge. The quadruple effect is the one most widely used nowadays, and about 4 lbs. of water are evaporated per lb. of heating steam.

In *Fig.* 421 the *General Layout of a Quadruple Effect* of American design is shown. The first vessel *I* is heated by exhaust steam, which enters at *a*. The hot clarified juice is charged at *b*, to the four inlets on the bottom, thus the juice is fed in right below the vertical tubes of the calandria, the juice being on the inside of the tubes. The first vapours accumulate in the vapour belt of the first body and, after passing the catch-all or entrainment device *c*, are conducted by the vapour pipe *d* to the steam belt or calandria of the second body *II*. The juice circulates through the tubes and overflows over the upper tube plate towards the centre tube or downtake of the calandria, the concentrated juice leaving the body at *e*. An automatic juice level control apparatus *f*, having a telescopic adjusting sleeve, maintains a constant juice level in *f*.

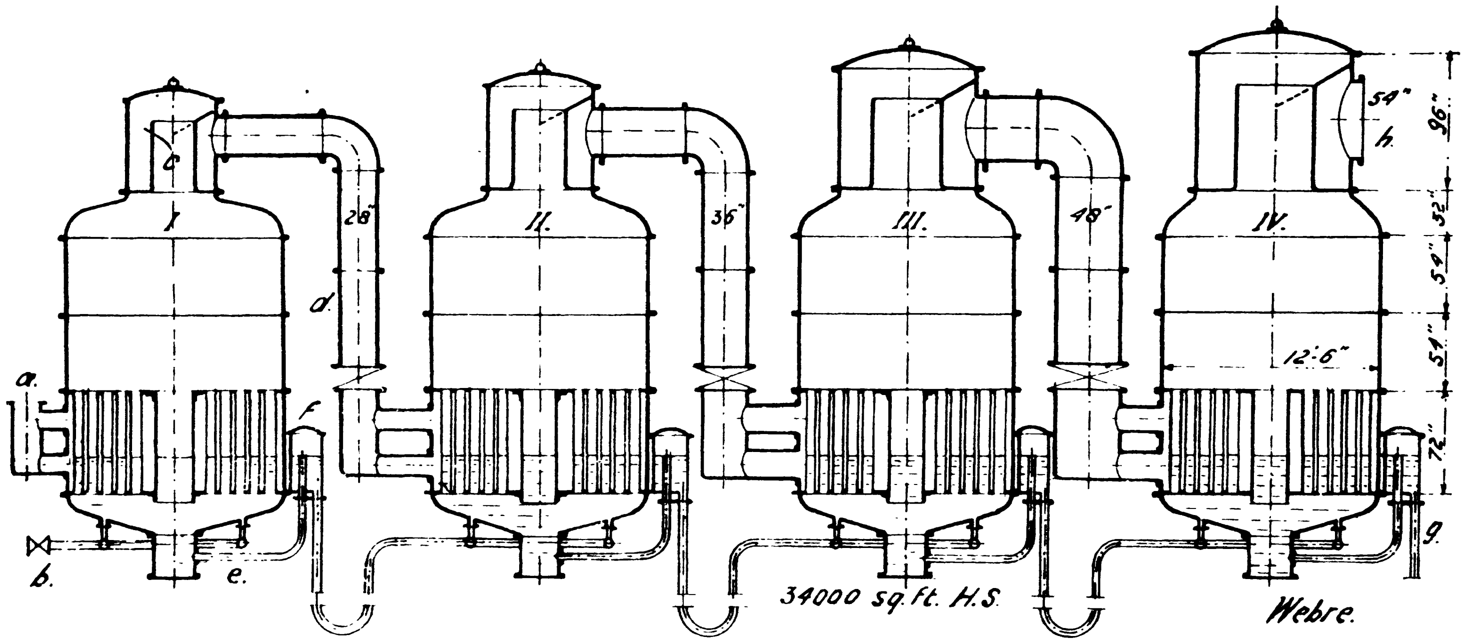


Fig. 421.—General Layout of a Quadruple Effect.

As the pressure in I is higher than the prevailing pressure in II, the juice will be pressed out from the first body, for which reason the vessel of the level control f is connected to the vapour space of this same body. The juice connexion between the first and the second body has to be provided with a syphon or a throttling valve, to give a continuous feed to the next body. The cycle is repeated four times and the syrup of the last body is withdrawn at g by a pump or by gravity, when the last body is placed at a barometric head at least equal to the vacuum prevailing in IV.

The vapours of the fourth body are conducted to a barometric condenser at h, so as to maintain a vacuum, which can only be done by a condensing performance, in which the vapour volume is reduced to the corresponding water volume.

The total range in pressure of a standard quadruple effect is between 3 lbs. per sq. in. and 26 in. as assumed in the *Pressure and Temperature Stepping Graph* at *Fig.* 422, and from practical observations it has been found that the total pressure drop is divided into nearly equal stages. The chart is divided into the ranges for both pressure and vacuum type evaporators, the total range being from 15 lbs. gauge pressure up to 26 in. vacuum. There have been a

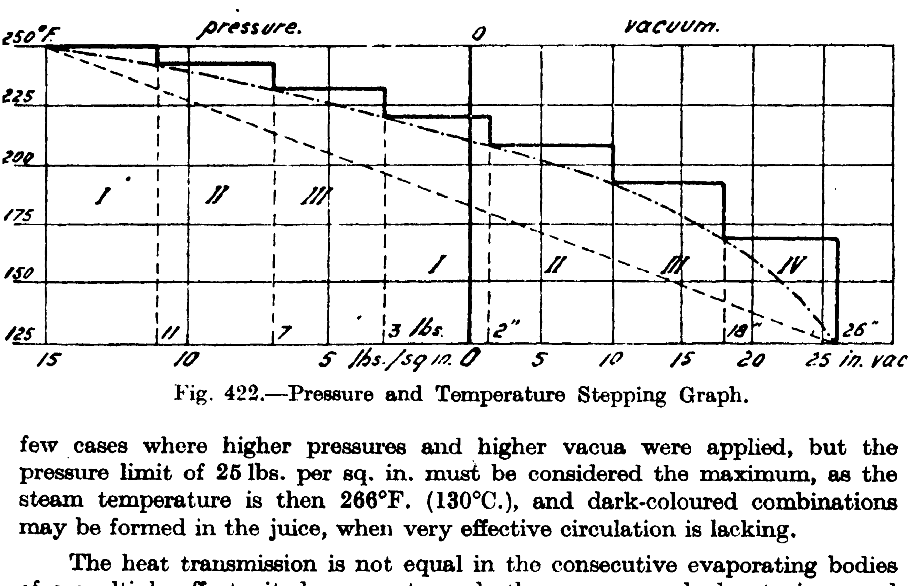


Fig. 422.—Pressure and Temperature Stepping Graph.

few cases where higher pressures and higher vacua were applied, but the pressure limit of 25 lbs. per sq. in. must be considered the maximum, as the steam temperature is then 266°F. (130°C.), and dark-coloured combinations may be formed in the juice, when very effective circulation is lacking.

The heat transmission is not equal in the consecutive evaporating bodies of a multiple effect; it decreases towards the vacuum end, due to increased density of the juice, which has reduced the heat absorption, and to the low pressure vapours of less density, which have proved less efficient a heating medium than the steam of higher pressure.

The heat drop, therefore, increases towards the last bodies, since it has become standard practice for manufacturing reasons to make all the bodies of equal size.

From tests carried out on factory equipment by different authors, the following average coefficients of heat transmission have been found :—

First body ..	$K = 450$	B.Th.U. sq. ft./hr./1°F. or	2250 cal./sq. m./hr./1°
Second body	$K = 350$	,, ,, ,, ,,	1750 ,, ,,
Third body ..	$K = 250$	,, ,, ,, ,,	1250 ,, ,,
Fourth body	$K = 150$	,, ,, ,, ,,	750 ,, ,,

these being average values for a straight quadruple effect, but it should be remembered that there is no fixed ratio between the heat transmission coefficients, as with increased incrustation or faulty circulation of an evaporator within the train the temperature difference for this body will automatically be increased. As the pressure also varies with the temperature required, any disproportionate stepping of the pressure will indicate which body is deficient.

From the foregoing it will be obvious that :—

$$450\Delta_i = 350\Delta_{ii} = 250\Delta_{iii} = 150\Delta_{iv}$$

when Δ_i is the heat drop in degrees F. for the first body, Δ_{ii} for the second, etc., and from the total heat drop we find the individual ones are, per cent. :—

	Per cent.
First body	14
Second body	18
Third body	25
Fourth body	43

In *Fig.* 422 a total heat drop of 221—125 = 96°F. is assumed, the individual ones thus being: 13, 17, 24 and 42°F. for a quadruple effect of the vacuum type. For a pressure triple effect, it will be seen that there is less difference between the individual heat drops.

An *Evaporation Chart* is shown in *Fig.* 423, indicating the evaporation curve from 15 to 65° Brix, corresponding to a 77 per cent. total evaporation.

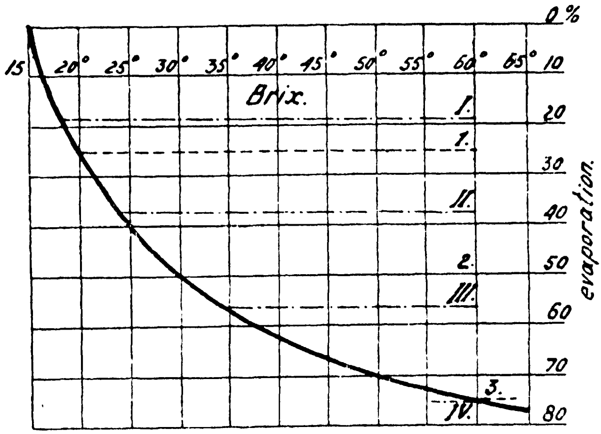


Fig. 423.—Evaporation Chart.

The bodies of a multiple effect will each evaporate nearly the same amount of water, as there is only a slight difference between the total heat input of each consecutive body. The intersection lines for a triple and a quadruple effect are thus drawn equidistant and the corresponding Brix obtained at each body can be easily read.

For any factory the size of the evaporating plant, viz., its heating surface, $H.S.$, can be derived simply from :—

$$H.S. = W_e \div E \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (108)$$

in which : $H.S.$ = heating surface in sq. ft. or sq. m.

W_e = total quantity of water to be evaporated in lbs. or kg./hr.

E = evaporation in lbs./sq. ft./hr. or kg./sq. m./hr. as previously tabulated in this chapter.

The evaporation figure should be taken low for new factories, to make provision for future increases of grinding capacity, which will generally be required. Where a vapour cell or pre-evaporator is provided, the total heating surface applies equally; such a pre-evaporator can of course be added to any existing multiple effect, so as to increase the total heating surface.

2.—Construction of Evaporators.

There are two types of evaporator in use, both being of the calandria type, viz. :—

(*a*) Vertical tube evaporators.
(*b*) Horizontal tube evaporators.

The totally submerged type of heating surface is now obsolete and the partially submerged type has become standard practice, as it ensures the formation of a juice film on the inside or the outside of the tubes.

In *Fig.* 421 is shown a quadruple effect of the vertical tube type, which is the type mostly used, the juice circulating within the tubes and the heating steam on the outside. Nevertheless, it is common practice to consider the heating surface as on the inside diameter of the tubes, and when comparing estimates, it should be ascertained *if inside or outside heating surface is meant*, the latter being about 10 per cent. in excess of the former.

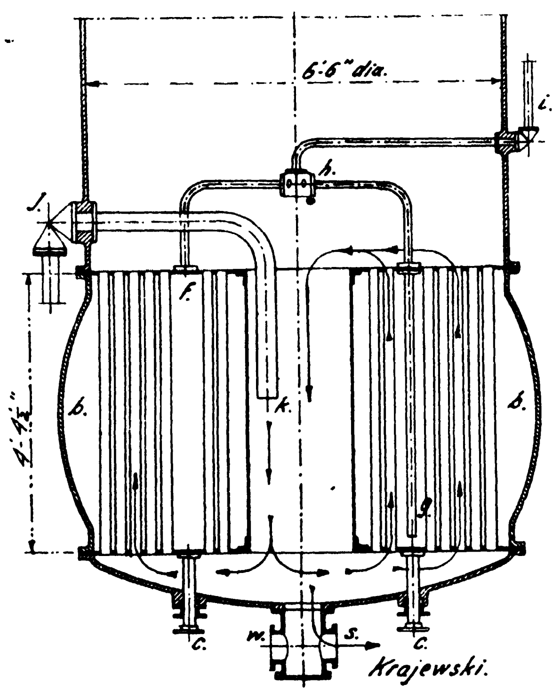


Fig. 424.—Steam Belt or Calandria.

In *Fig.* 424 the *Steam Belt or Calandria* of a vertical evaporator is shown. The steam belt as well as the upper one is made of cast iron, although steel plate is used by many designers. Cast iron is used in those instances where washing is done with hydrochloric acid, but steel plate bodies do not now suffer so much from this treatment, especially as inside countersunk rivet heads and welded constructions have opened a new field. For pressure vessels and especially their steam belts, steel has the advantage of greater strength, and the author has supplied with satisfactory results cast iron evaporators having a steel calandria, the juice not coming into contact at all with the steel. The steel plate used was from $\frac{3}{8}$ in. to $\frac{9}{16}$ in. thickness, whereas the cast iron was from $\frac{3}{4}$ in. to $1\frac{1}{4}$ in. thick, according to the size of the evaporators. The sizes of standard evaporators may be taken as from 500 up to 12,000 sq. ft. H.S. per body.

The calandria in *Fig.* 424 is of the fixed type, the floating type not being often used. The tube plates are made of cast bronze, rolled yellow metal, steel, special steel alloy or copper. Brass, yellow metal and steel tube plates are made to a thickness of $\frac{3}{4}$ in. to $1\frac{1}{4}$ in., giving good adherence with the tubes expanded into them. Copper tube plates are generally only $\frac{5}{8}$ in. thick, in particular in American designs. The weight of the tubes being considerable, the tube plates must be capable of supporting them, otherwise sagging in the centre will occur; and a proportionate thickness thus should be provided in large vessels. Sometimes the calandrias are suspended by phosphor bronze tie rods from the upper belt of the evaporator. Copper tube plates at times acquire a mattress-like surface through tube replacements, unequal expansion, or sagging.

The tubes are fixed in by means of a tube expander and for the bottom plate a long stem expander can be applied from the top plate, as it is not always a pleasant job to insert the expander from below in the restricted bottom space. It is also good practice to have the supporting footings on the steam belt, so as to be able to lower the bottom when the tubes have to be expanded.

The tops of the tubes need not necessarily be flush with the upper tube plate, as the circulation is from below towards the top and, generally, the tubes protrude about $\frac{1}{16}$ in. to $\frac{1}{8}$ in. Beading or countersinking is sometimes employed with evaporator tubes. The length of the tubes to be ordered should be $\frac{1}{4}$ in. more than the overall measurement between the outside of the tube plates. This measurement should be taken at different positions, as variations might exist.

The size of the tubes (of brass or copper, as steel should not be used—see Chapter XVIII) is usually 2 in. O.D. for American and British practice, but the Continental standard of about $1\frac{5}{16}$ in. to $1\frac{7}{16}$ in. (33/36 mm.) is also accepted in America as it will increase the H.S. per given tube plate area and achieve a better evaporation through more efficient heat transmission. The weight of a 2 in. copper tube, having 0·065 in. wall thickness is 1·53 lbs./ft. for copper and 1·46 lbs. for brass. The tubes are arranged in diamond pitch, as shown in *Fig.* 379, this giving the greatest H.S. per unit area of the tube plates. The wall between two adjacent tubes is about $\frac{3}{8}$ in. to $\frac{1}{2}$ in., so the tube pitch will amount to : $p = O.D. + \frac{3}{8}$ in. (or $\frac{1}{2}$ in.).

Deoxydized copper has been used to advantage for these tubes, the ends generally being annealed for a length of 3 in. Some operating engineers do not favour annealing and the author has seen copper tubes which corroded in a relatively short time at the line between the annealed and non-annealed parts. The holes for the tubes have to be drilled $\frac{1}{32}$ in. above outside tube diameter.

Incondensible gases may accumulate in the steam space of the calandria and will impair the heat transmission to a great extent. It is therefore important that the calandria be vented. As separation of gases is only possible when there exists a difference in specific gravity, connexions g and f from respectively the bottom and the top of the steam space are provided in the design of *Fig.* 424. The six connexions are joined in a union piece h and then connected to the valve i outside the body. Some designers provide perforated vertical tubes from the top to the bottom of the steam space, and equally good results have been reported as to the venting of these incondensible gases.

The individual connexions i are joined to a main line of normally 2 in. going to the condenser or to a subsequent body, but in both cases the valves i should be throttled, as otherwise considerable steam losses will occur. The connexions inside the vessel above the tube plate are made preferably of copper. Steel tubes will corrode within a crop time, when weekly cleaning with muriatic acid is applied.

Regarding juice circulation in the evaporator, there are two opinions, each having a number of adherents. The one relies on the supposed fact that the concentrated juice on account of its increased specific gravity will flow to the bottom, thus the thinnest juice always floats on top, more or less independent of the loci of the feed and discharge connexions. The objection to this theory is that the circulation is too strong to make this separation feasible, but anyhow a certain intermixture takes place, as is proved by practical performance. The design of *Fig.* 424 has been developed according to this view, the feed being led via the valve *j* through a bent tube inside the centre tube or downtake. It is, of course, supposed that the juice will not flow directly towards the discharge *s*, but will first circulate through the tubes. The author has records of an evaporation up to 6·5 lbs. per sq. ft. in quadruple effect with this kind of equipment.

The second opinion lies in forcing the juice flow in the direction of the prevailing circulation. The circulation is due to the fact that the juice comes rapidly to boiling within the tubes, and the vapour bubbles will carry it in a thin layer or film to the top plate. During its travel along the tube walls, heat will be readily absorbed and an increased evaporation achieved.

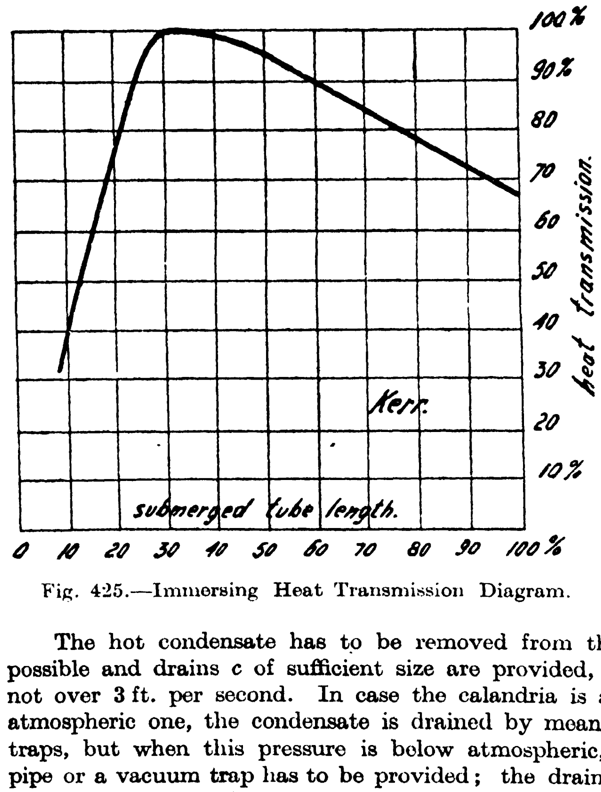


Fig. 425.—Immersing Heat Transmission Diagram.

The steam space *b* around the calandria serves the purpose of even distribution, so as to make it possible for the heating medium to penetrate between the tubes. The centre downtake as a rule has a diameter of about one-quarter to one-fifth of the inside body.

The hot condensate has to be removed from the calandria as soon as possible and drains *c* of sufficient size are provided, calculated on a flow of not over 3 ft. per second. In case the calandria is at a pressure above the atmospheric one, the condensate is drained by means of one or more steam traps, but when this pressure is below atmospheric, then a barometric leg pipe or a vacuum trap has to be provided; the drain can also be connected to the suction end of a pump, creating the necessary vacuum for condensate withdrawal. The condensate has to flow to the pump intake by gravity, and it is good practice to have a set of sight glasses in the condensate line, to be able to observe the rate of flow.

Tests have been made by KERR to find out how much of the tubes should be immersed in the juice, and the *Immersing Heat Transmission Diagram*, *Fig.* 425, has been drawn according to these tests, showing that the most favourable results are obtained by between 30 and 40 per cent. immersion of the vertical tube length. Below 30 per cent. the heat transmission drops rapidly and the juice level should be kept rather above than below the mark mentioned. A juice level gauge is provided on each body, but sometimes hydro-dynamic action of the juice will give an error in the indication, and the bottom connexion should be made at the locus of reduced juice flow to render this error as small as possible.

In *Fig.* 426 is shown a *Calandria with Sealed Downtake or Centre Tube,* the steam being admitted in an annular space *b* having baffle plates with slots on the inside periphery for even distribution of the heating medium. The juice is charged at *j* at the bottom of the evaporator and generally a circular perforated pipe is laid on the bottom for equal juice distribution. As the centre downtake is closed, the juice has to rise through the tubes and flows or spouts over the upper tube plate towards the downtake to discharge at *s*. It is to be noted that several designers keep midway between the open and the sealed downtake systems, the juice when entering the body being distributed below the calandria tubes and thus is forced to participate in the circulation. The concentrated juice is then discharged at the bottom centre. The highest evaporation figures the author has observed in practice have been obtained with sealed downtakes, which may indicate that this system has practical value.

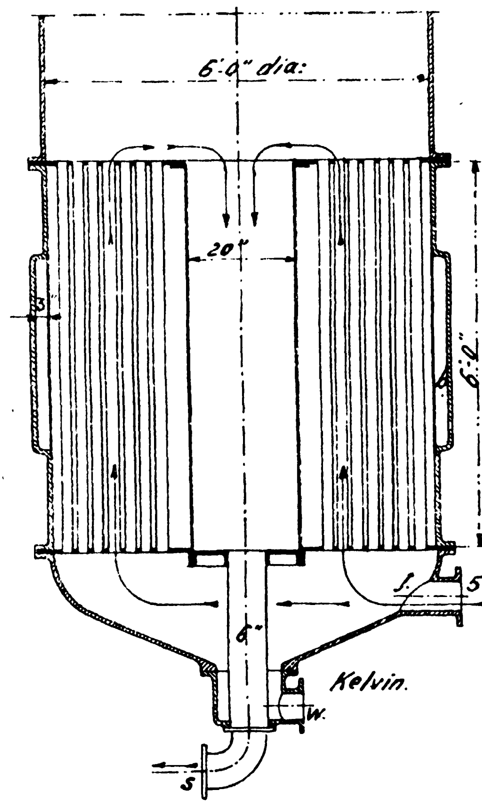


Fig. 426.—Calandria with Sealed Downtake.

The tube length of most evaporators in cane sugar factories is between 4 and 5 feet, but longer tubes, 6 to 8 feet, have been installed by the author with very good operating results and moreover with a reduction in space occupied. Several evaporators of the French design of KESTNER, with tubes up to 23 ft. long, have also been applied in the cane sugar industry with good results, especially as first bodies. In beet sugar factories these long-tube evaporators are used in multiple effects, but it should be remembered that beet juices are considerably purer than cane juices and incrustations are thus less to be feared.

The steam and juice flow in the evaporator has received attention from several designers and in *Fig.* 427 is shown a *Baffled Multi-Downtake Calandria* of American design, of which favourable operating results have been reported. The steam is guided by baffles *b* in two directions, reaching in zig-zag fashion the centre of the calandria. As will be noted, the steam path is reduced in area towards the calandria centre, thus allowing for a reduced steam volume through consecutive condensation.

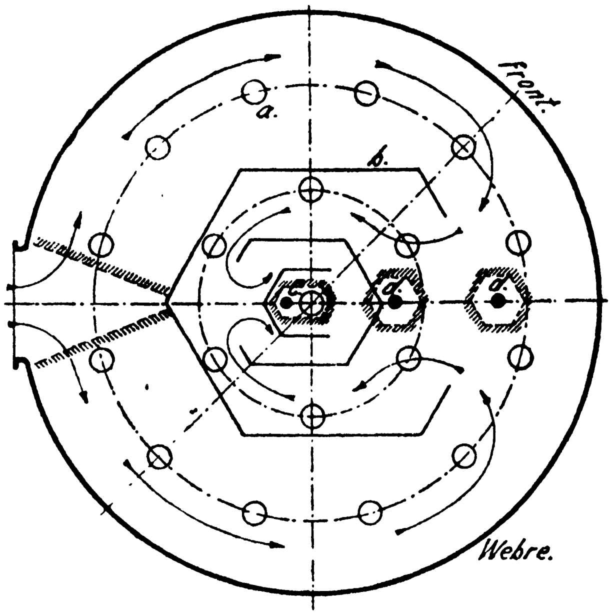


Fig. 427.—Baffled Multi-Downtake Calandria.

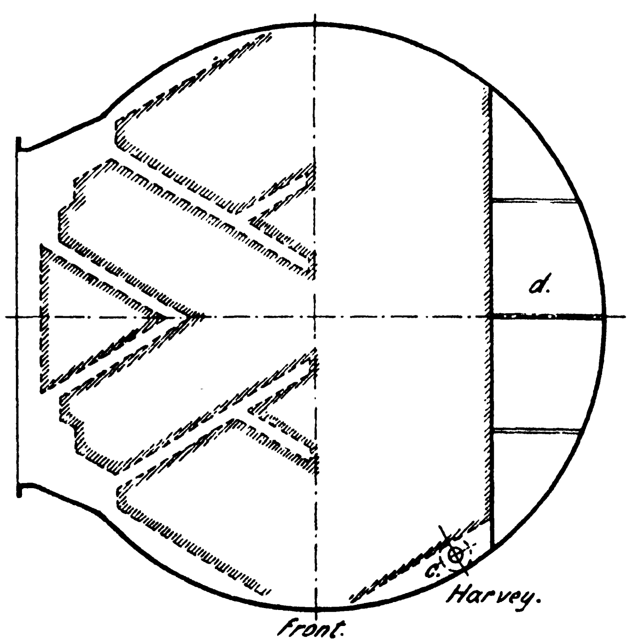


Fig. 428.—Segmental Downtake Calandria.

Another feature of this design is the number of larger tubes or downtakes of about 6 in. diameter, which clearly will shorten the juice travel over the upper tube plate. The condensate is drained by the outlets *d* and the incondensible gases will be forced to the end of the steam paths at *c*, where the steam flow is assumed to have ceased through complete condensation. The contour of the open spaces within the calandria is indicated by shaded dotted lines.

In a large Cuban sugar factory an average evaporation of about 5 lbs. per sq. ft. per hour has been achieved in quintuple effect arrangement with this type of evaporator, using exhaust steam as a heating medium, this being a favourable figure.

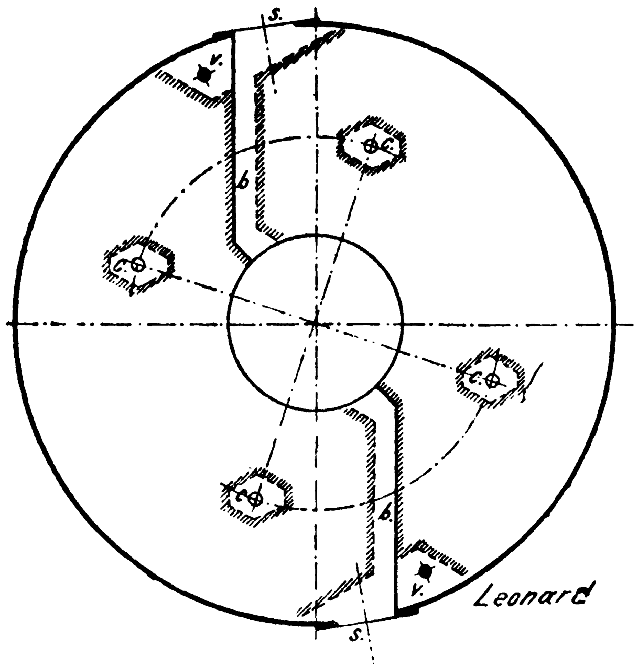


Fig. 429.—Calandria with Bilateral Steam Entrance.

Another design of British origin is the *Segmental Downtake Calandria* shown in *Fig.* 428, the steam being admitted by a broad inlet with special steam lanes to the calandria tubes for proper penetration of the steam. Through the shortness of the steam path a very effective heating is obtained. The condensate is drained at *c* and incondensible gases are vented at different loci, close to the straight wall of the downtake. The downtake *d* is of segmental area, reinforced by three plates between the flat wall and the belt periphery. As the downtake is located at the coolest spot of the calandria, it will materially add to a good circulation of the juice.

In *Fig.* 429 is shown a *Calandria with Bilateral Steam Entrance*, of American design, the steam entering at the diametrically opposed inlets *s*, guided by two short baffles *b*, which are generally made of copper plate of No. 16 S.W.G. The calandria is thus divided in two equal parts or sections, and the steam lanes give access over the full width of the steam paths. As will be seen, these steam paths have also a converging section towards the end, at which spot the incondensible gases are withdrawn at *v*, thus at the current-free part of the calandria. The condensate is drained at the loci of heaviest condensation, viz., at *c*.

With this type of calandria an average evaporation of 8 lbs. per sq. ft. H.S./hr. has been achieved with quadruple effect performance, using exhaust steam of 5 lbs. gauge as heating medium, and 26 in. vacuum in the last body.

In *Fig.* 430 is shown another British design, the *Inclined Type Calandria*, the tube plates being slightly inclined towards the sealed segmental downtake *a*. These plates are reinforced by throughgoing stay bolts (indicated by the chain dotted lines) at regular intervals. Owing to the sealed downtake, the juice is forced through its predetermined circulation, the concentrated juice being discharged at *b*. As already mentioned, the author has observed high rates of evaporation with sealed downtake evaporators. It will be obvious that the shortest hydrostatic head of juice will prevail in the tubes close to the steam inlet, and the most intensive ebullition will take place in these tubes, which suits the purpose of the long travel over the upper tube plate towards the downtake. The calandria is made of cast iron, its flat wall being of the same material, well reinforced by ribs integrally cast on.

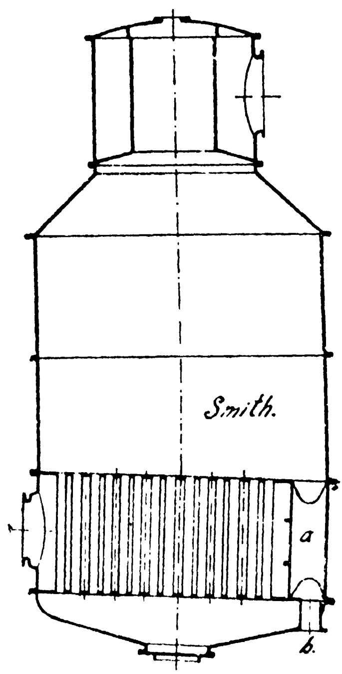
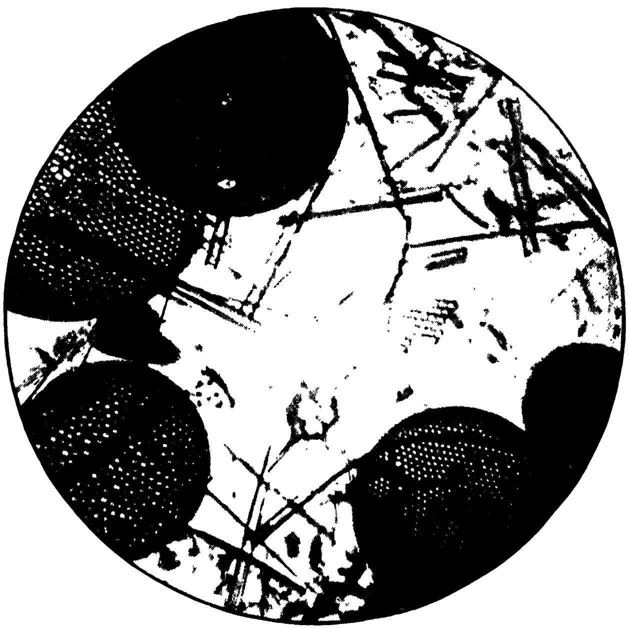


Fig. 430.—Inclined Type Calandria.

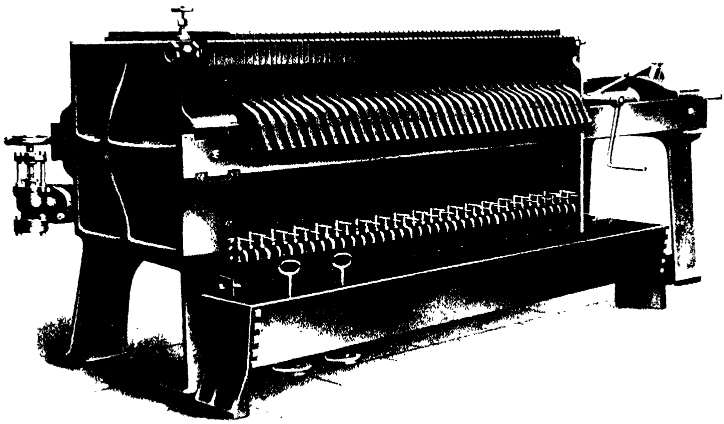
The *Vapour Space* of an evaporator has a height of about up to twice the tube length, it generally being about 1½ times this length. HAUSBRAND has developed an interesting theory about the height and volume of the vapour space required, the deciding factors being the tube length, the immersion height, the concentration, the pressure or vacuum inside the vapour space, and the size of the bubbles. As several of these data have only been ascertained for solutions other than cane sugar juice, sugar evaporator designers are guided by practical observation. From this it will be found that the bodies under a partial vacuum spout the juice particles up to the greatest height. HAUSBRAND soundly advises the provision of entrainment apparatus or catch-alls, where the theory might be at fault, and for sugar juice evaporators these catch-alls or entrainment devices should be included in the design of every evaporator or vacuum pan; a separate Chapter will be found further on, dealing with this kind of equipment.

A recently patented design is the *Two-fold Circulation Inclined Calandria* shown in *Fig.* 431, the tube sheets being sloped at 10° with the horizontal. The juice is charged at the left hand side at *a* and on reaching the upper tube plate will be guided by a bilateral set of downtake pipes *b*-*c* of ample size to the bottom of the right hand section, the evaporation performance being repeated and the concentrated juice discharged at *d*. A novel feature with this construction is the discharge of the concentrated juice from above the upper tube plate.

MICROGRAPH OF SPICULAR DIATOMS
(Diatomaceous Silica).
(*Dicalite Co.*)



MICROGRAPH OF COSCINODISCUS DIATOMS
(Diatomaceous Silica)
(*Dicalite Co.*)



Johnson Plate and Frame Press.
(S. H. Johnson & Co., Ltd.)

Hand Quarrying of Diatomaceous Silica after Top Soil has been Removed.
(Dicalite Co.)

There is also a patented design where the whole evaporator is placed in an inclined position of about 10° with the vertical. Inclined evaporators have been used in process work of the chemical industries to a large extent.

As the first body of a multiple effect generally has a vapour pressure slightly above the atmospheric one, the juice has to be charged also under a hydrostatic head, which may be produced by a centrifugal pump, regulated by a throttling gate valve in the discharge line. When using piston pumps it will generally be more practicable to have a *High Level Evaporator Supply Tank* with a gravity head well above the equivalent pressure in the evaporator.

Given a minimum assumed density of 1·06, a column of 1 ft. high will exert a hydrostatic pressure of 0·458 lbs./sq. in. or practically 0·5 lbs./sq. in. and a vapour pressure of 5 lbs. will require 10 ft. hydrostatic head above the prevailing juice level in the particular first body. Designers sometimes place this juice charging or supply tank on top of the catch-all of the evaporator. An overflow returns excessive juice feed into this tank back to the clarified juice tank. Tank and hot juice lines should be covered with heat insulating material.

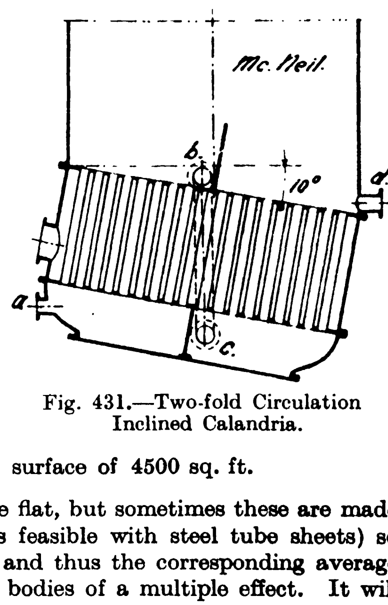


Fig. 431.—Two-fold Circulation Inclined Calandria.

The general arrangement of a *Vapour Cell* or *Pre-Evaporator* (as supplied by the author) is shown in *Fig.* 432, this having 6000 sq. ft. heating surface, composed of copper tubes of $1\frac{5}{16}$ in. × $1\frac{7}{16}$ in. diameter and 8 ft. length. The steam is charged to the calandria by three inlets on the periphery and the circular downtake is placed at one side of the calandria, close to the outside periphery. When working with an exhaust pressure of 6 lbs./sq. in. and 2·5 lbs./sq. in. vapour pressure, an average evaporation of 8·3 lbs. water per sq. ft./hr. has been obtained. The vapours are used for heating the total juice up to 212°F. in five juice heaters of an aggregate heating surface of 4500 sq. ft.

The tube plates of this vapour cell are flat, but sometimes these are made to curve spherically downwards (which is feasible with steel tube sheets) so as to reduce the juice space in the body and thus the corresponding average time limit the juice stays in the body or bodies of a multiple effect. It will be obvious that with higher steam temperatures a briefer stay of the juice in the evaporators will be required, to avoid inversion losses.

In any evaporator in single or multiple effect arrangement, the average amount of juice going through it will be, according to formula (105) :—

$$W_{average} = (W_j + W_{ji}) \div 2$$

and the average time the juice remains in it :—

$$t = (V \times 60) \div W_{aver./hr.} \quad \ldots\ldots\ldots\ldots \quad (109)$$

in which : t = average time in minutes.
V = weight of the juice volume in the evaporator.
$W_{av./hr.}$ = average weight of juice going through the evaporator per hour.

For a quadruple effect V is the weight of the total juice volume in the four bodies, and, for each body separately, obviously the fourth part of that quantity, it being assumed that the bodies are of equal size. The time limit t for each body in pressure evaporators, where the danger of colouring might exist, is generally kept within 3 minutes.

When the heating surface has been determined by formula (108) the number of tubes required can be easily calculated, when the inside diameter and the tube length have been decided on. Furthermore, D being the inside diameter in inches of the evaporator, about 6 per cent. of the area has to be deducted for the downtake, thus leaving $0{\cdot}94 \times \pi \times D^2 \div 4$. As the total tube plate area cannot be occupied with tubes, as space has to be provided for the connexions of the condensate drains, the tubes for incondensible gases and the steam lanes, a further 10 per cent. of the net area has to be deducted, thus leaving $0{\cdot}84 \times \pi \times D^2 \div 4 = 0{\cdot}693\ D^2 \simeq 0{\cdot}7\ D^2$.

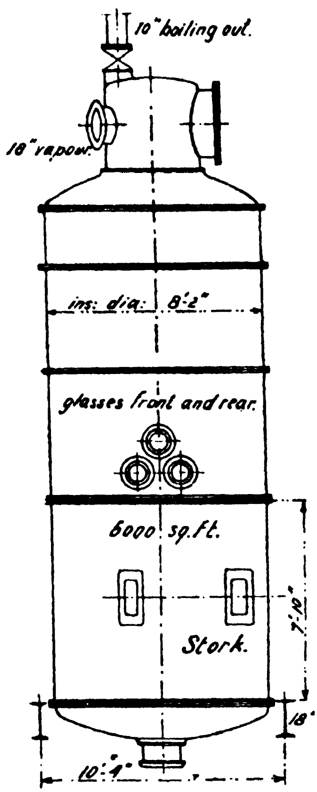


Fig. 432.—General Arrangement of a Vapour Cell.

The tubes are assumed to be spaced in diamond or rhombic arrangement, having a pitch p, the rhombic top angle as a rule being 60°; and the area of each rhombus, which is equivalent to the space occupied by one tube, amounts to : $p^2 \div 2 \times \sqrt{3} = 0{\cdot}866\,p^2$.

The number N of tubes thus has to fulfil the equation :—

$$N = 0{\cdot}7\ D^2 \div 0{\cdot}866\ p^2 \quad \text{or}$$

$$D^2 = N \times 0{\cdot}866\ p^2 \div 0{\cdot}7 = 1{\cdot}237 \times N \times p^2$$

from which is derived :—

$$D = 1{\cdot}11 \times p \times \sqrt{N} \quad .. \ (110)$$

for a tube pitch of $1\frac{7}{8}$ in. (outside tube dia. $1\frac{7}{16}$ in.)

$$D = 2{\cdot}08\ \sqrt{N} \ \ldots\ldots\ldots \ (110a)$$

and for $2\frac{1}{2}$ in. pitch (outside tube dia. 2 in.)

$$D = 2{\cdot}775\ \sqrt{N} \ \ldots\ldots\ldots \ (110b)$$

In those instances where additional heating surface is required, but where the installation of a vapour cell or pre-evaporator is not feasible or economical, the heating surface of existing evaporators can be increased by *Attached Auxiliary Evaporating Bodies*, as shown in *Fig.* 433.

The main body a is of the segmental downtake type and the auxiliary body b has between 25 and 50 per cent. of the heating surface of the former. The juice enters the auxiliary body and is then led to the main vessel, although provision has been made for divided juice distribution. The tube length of both main and auxiliary bodies has been designed differently, and it will be noted that the hydrostatic height H_1 in the auxiliary body is greater than the one prevailing in the main body, indicated by the dimension H. This, nevertheless, is quite practicable, when the heating steam is admitted into the auxiliary body and a slightly higher temperature prevails in this body. Moreover, the juice level can be kept different in the two bodies.

Before attempting any such scheme of enlargement, it should be ascertained in advance whether the existing vapour pipes between the bodies or to the condenser are of sufficient size, as not to hamper the evaporation performance, since pipe friction in the vapour lines will require a higher pressure in the producing body, which is equivalent to a higher temperature, thus less temperature difference between the heating steam and the juice, and a correspondingly reduced heat transmission.

As already mentioned, KESTNER designed evaporators with very long tubes, but as these proved inconvenient for cane juice evaporators in some instances, the *Semi-Kestner Evaporator* (*Fig.* 434) came into use, having tube lengths up to 12 ft. The juice is charged at *a* at the bottom through a circular

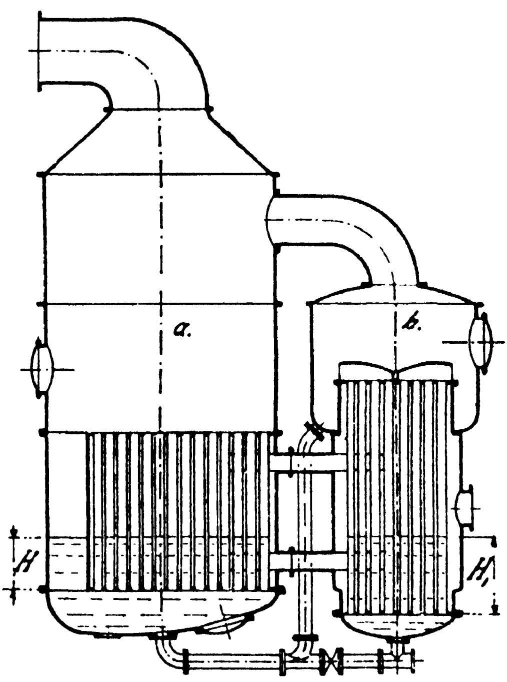


Fig. 433.—Attached Auxiliary Evaporating Body.

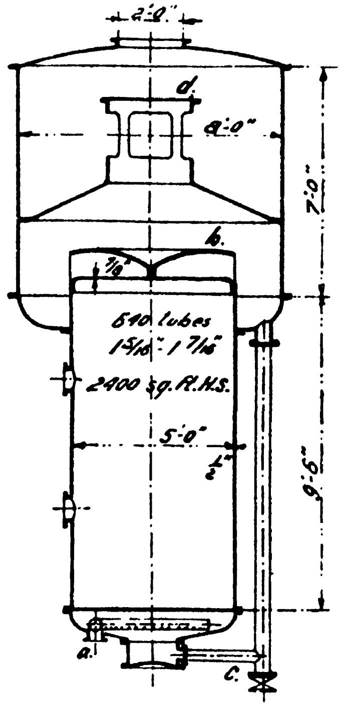


Fig. 434.—Semi-Kestner Evaporator.

perforated pipe. As the spouting effect increases with the tube length, an impeller-like device *b* is mounted on top of the upper tube plate, ensuring that the juice drops are diverted towards the inside periphery of the vapour space. A juice catcher *d* is also provided, to prevent drops of juice being drawn into the vapour pipe. The concentrated juice is withdrawn from the annular bottom of the vapour belt.

The *Horizontal Tube Spray Evaporator* shown in *Fig.* 435 has been widely used for seawater evaporation on board ship and also in the sugar industry. The steam is admitted on the inside of the sealed tubes, whereas the juice is sprayed over the outside. There is no hydrostatic head and film formation results. A circulating pump *c.p.* is required to maintain this spraying performance. The concentrated juice is withdrawn also from the lowest part. In practice a high evaporation has been obtained and a heat transmission coefficient reported of 675 B.Th.U./sq. ft./1°F./hr. for a pressure evaporation of 5 lbs. gauge, dropping to 200 B.Th.U. for 14 in. vacuum.[1]

The removal of incrustations is nevertheless not so easily undertaken here as in the case of vertical tubes with inside juice circulation. The tubes are slightly inclined, the hot condensate flowing towards the front end, where it is drained by means of a float-operated valve.

A problem that has aroused interest lately is whether the evaporating and condensing performances could not be carried on closer together, avoiding the large and costly vapour spaces and the connecting pipe lines. In *Fig.* 436 is shown such a *Compound Evaporator*, which has been used for laboratory tests at the University of Delft (Holland).[2] The apparatus is composed of an outside shell having about 7 in. outside diameter with a steam jacket *a*. The juice is charged at the top of the apparatus by means of a perforated tube *b*, thus descending in film fashion on the inside wall of the steam jacket. The concentrated juice is re-collected in a rim *c*, and is discharged outside the apparatus. The produced vapours are almost instantly condensed on the wall of the inner tube *d*, which is cooled by water supplied by a tube *e*, the condensate being drained at *f* and the heated cooling water discharged at *g*. Although the distance between the heating and the condensing surfaces has been as little as 1½ in. no trace of sugar has been found in the condensate, the tests being made with a 5 per cent. sugar solution. An average evaporation of 0·45 lbs./sq. ft./1°F./hr. (4 kg./m²/1°C./hr.), has been obtained. A more technical solution for factory performance on sugar juices is now sought.

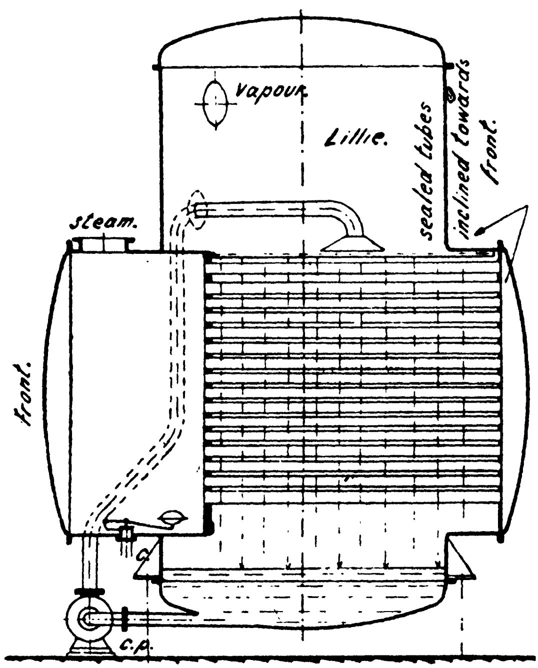


Fig. 435.—Horizontal Tube Spray Evaporator.

When boiling out the evaporators with caustic soda solution, steam is not supplied to the calandria, as it will give too intense a boiling, due to the large heating surface. A perforated steam pipe is therefore arranged at the bottom of the evaporator. The gases emerging from this or a hydrochloric acid solution should not be allowed to enter into the next calandria or the condenser, as they might cause corrosion, and each body should have a 10 in. connexion with a gate valve out through the roof to relieve the boiling house of these vapours.

[1] See article by Prof. E. W. KERR, *Am. Soc. Mech Engrs.*, 1916.

[2] See the abstract of an article of H. I. WATERMAN and A. DAZEBT in *Int. Sugar Jl.*, 1936, p. 19.

3.—Evaporator Details.

A few details should be considered more closely, as these will enhance the operating performance. In *Fig.* 437 is shown a *Flush Tube Plate* arrangement, the plate being held between the flanges of the steam and the vapour belt, which will allow for any unequal expansion of tube plate and belt material. As two independent joints have to be made, the bolts are provided with a shoulder of cylindrical or conical shape, which secures the tube plate firmly to the steam belt. The bolt holes should be $\frac{1}{16}$ in. over the bolt diameter.

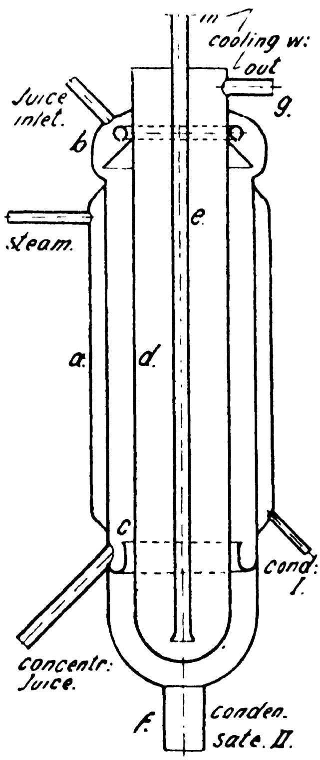


Fig. 436.—Compound Evaporator.

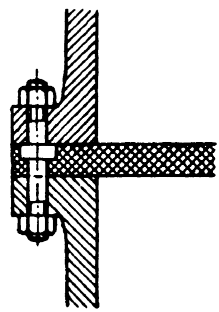


Fig. 437. Flush Tube Sheet.

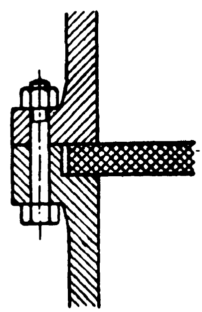


Fig. 438. Recessed Tube Sheet.

Another construction is a *Recessed Tube Sheet*, as shown in *Fig.* 438. This has the disadvantage that leaks between the juice and the steam space cannot be so readily detected and as the annular space between the tube plate and the lower flange recess may fill up with the red lead jointing material, which becomes very hard, the expansion of the tube sheet cannot take place freely and the author has seen steam belts cracked as a consequence, when made of cast iron.

A very important detail ot an evaporator and also of a vacuum pan is the *Sight Glass* and, unfortunately, there are many evaporators which are sent out inadequately equipped, so far as ability to observe the contents is concerned. A good design of Continental make is shown in *Fig.* 439, the glass disc being ½ in. thick and of highly polished Pyrex glass, which has great transparency. The free diameter is 8 in. and the glass is held between two rubber gaskets, pressed together by a large threaded brass nipple, which has to be tightened by a special wrench. The outer mounting is also made of brass and can be easily inserted in the vapour belt opening. Some designers prefer rectangular glasses, but the joints are not so air-tight nor the pressure on the glass so even as with those in *Fig.* 439. Sometimes a brass ring is laid on the outer rubber gasket, to prevent the nipple bedding into the latter.

Fig. 439.—Sight Glass.

Sight glasses generally become stained or cloudy when not highly polished, and they should be cleaned of their thin deposit of lime-salts by washing with a little weak muriatic acid.

Sight glasses have to be provided at both front and rear of the evaporator, as otherwise the latter is termed " blind." With an electric bulb hung before the rear sight glass, the interior of the body will be sufficiently illuminated for proper observation.

As the interior of an evaporator has to be accessible for the purpose of scraping the tubes to remove scale, or else to repair leaky tubes, *manholes* have to be provided in the vapour belt, as well as in the bottom. For easy access a diameter of 18 in. is to be preferred, and the author has supplied this size frequently.

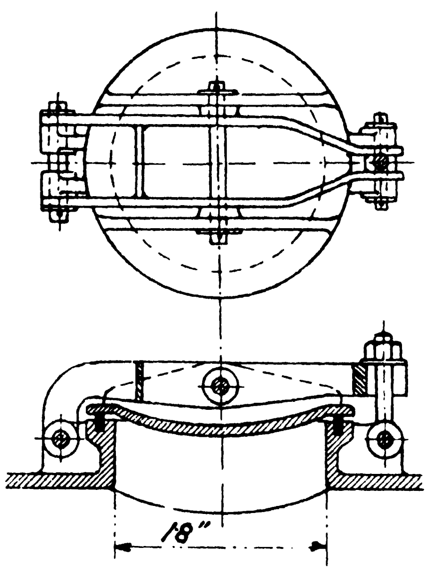


Fig. 440.—Manhole.

The manhole has to be designed for vertical as well as for horizontal position, and the hinging levers must be so attached that the cover will not sag when in open position. The double lever construction shown in *Fig.* 440 serves this object, the distance pieces being welded. A fixed rubber gasket is laid in a corresponding groove of the manhole opening. The lugs for the lever and the eye-bolt are cast symmetrically on the manhole opening, thus making a left and right hand arrangement equally possible. As will be seen, only one bolt has to be tightened.

Juice level regulators are shown in *Fig.* 421 fitted to all bodies of the multiple effect. In many instances, only the juice level of the first body is regulated by an automatic device, the other bodies being controlled by throttling gate valves in the juice lines between these bodies. In *Fig.* 441 such a *Full Automatic Juice Level Control* apparatus is shown, which has been applied on the first bodies of the multiple evaporators in beet sugar factories. It will be obvious that a constant juice level will achieve an improved evaporating performance as the hydrostatic head remains constant, and generally an increased heat transmission, thus giving an increased capacity to the evaporator.

The juice level acts on a float in the vessel *a* and operates a double-seated valve, which gives access to the steam relay *b*, which in turn controls the access of live steam to the regulating valve *c* in the juice charging line to the first body. The live steam relay enters into the circuit, as the vapour pressure of the first body generally will not be sufficient to operate the juice control valve and thus a medium of higher pressure is applied.

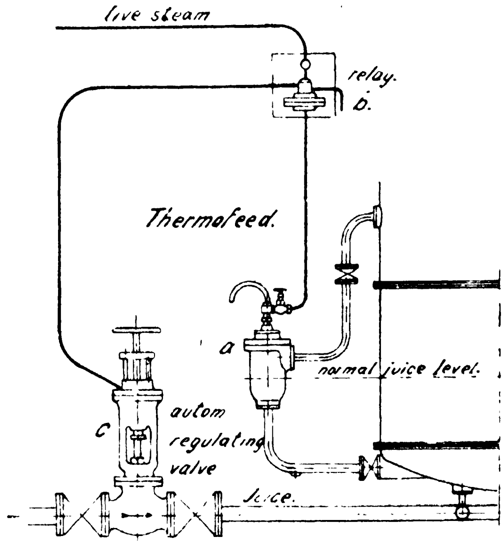


Fig. 441.—Full Automatic Juice Level Control.

A low juice level in the evaporator will speed up evaporation, whereas too high a level will retard it and it will therefore be obvious that, for optimum evaporating results, a controlled juice level will show paramount advantages.

The condensate of the evaporating bodies is generally not measured, but as it will indicate the performance obtained, means have been provided for the purpose in some instances. A *V-notch Condensate Meter*, based on a well-known principle, is shown in *Fig.* 442, this being a horizontal vessel, in which the condensate enters at *C.I.* To avoid turbulence a tranquillizing baffle *a* will assure an even flow to the sharp-edged V-notch, made of copper plate, marked *b*. The condensate flows off at *C.O.* and at *V* a connexion is made with the calandria, from which the condensate has been drained, for equalizing the vapour pressure inside the V-notch meter. Two sight glasses *l*, diametrically opposed, indicate the height *H* of the overflow level.

For a 90° sharp-edged V-notch, the quantity of effluent, according to KING, amounts to :—

$$Q = 2{\cdot}52 \times H^{2{\cdot}47} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (111)$$

Q being measured in cub. ft./sec., when *H* is given in feet. The computation has to be done by means of logarithms.

By marking a line on the sight glasses, it can easily be ascertained when the evaporator is falling behind in performance.

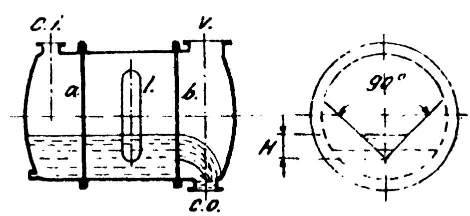


Fig. 442.—V-notch Condensate Meter.

In *Fig.* 443 is shown a system of *Barometric Condensate Drainage* as installed by the author on three bodies of a quadruple effect, each having 3000 sq. ft. heating surface and an evaporation of about 6·5 lbs. per sq. ft./hr. The condensate of the second body is released by a 4 in. copper syphon *b*, 14 ft. long, into a flashing vessel *a*, having diametrically opposed sight glasses for proper observation. The vessel is connected by a 6 in. pipe to the vapour space in the third calandria, to allow flashing with the benefit of the latent heat of the vapour produced. The third vessel is drained the same way and the condensate of the fourth vessel is discharged, together with the condensate of the previous ones, to the intake of an electrically-driven centrifugal pump c, having an output of about 400 gals./min., or over 300 per cent. of the

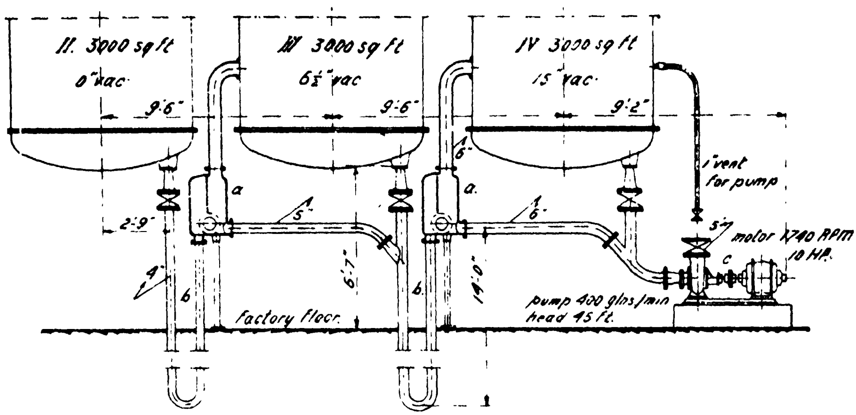


Fig. 443.—Barometric Condensate Drainage.

normal quantity of water to be handled. The pump delivers against a head of about 45 ft., of which 17 ft. is required to overcome the prevailing vacuum. The discharge of the pump is regulated by a throttling gate valve.

As 1 in. difference in vacuum requires only 1·14 ft. hydrostatic head in the syphon, it will be obvious that a large safety factor has been provided, so as to guarantee a steady flow of the condensate. The condensate of a previous body is not drawn into the next one, as is sometimes arranged, but the available heat of this condensate is released by the flashing.

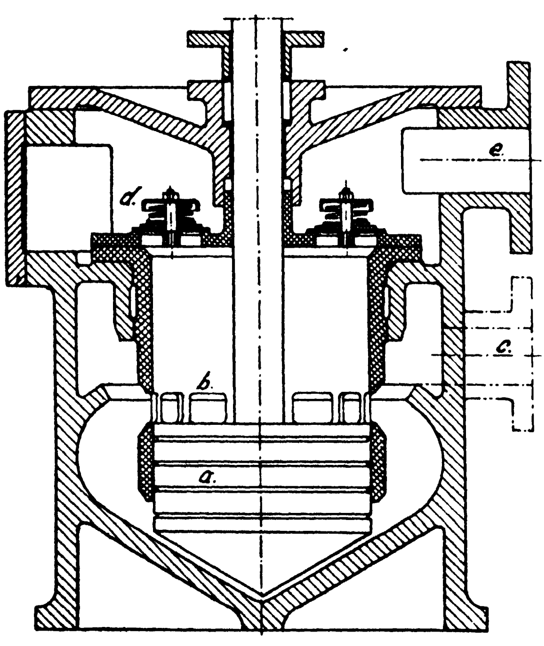


Fig. 444.—Edwards Wet Vacuum Pump.

A 1 in. vent connects the pump body to the calandria of the fourth vessel, so that one can prime the pump. One packing box, having a water seal for irregular air entrance is required, and for this reason the unilateral intake has been used.

The arrangement has worked satisfactorily in practice.

Instead of barometric condensate drainage of the calandrias the most common alternative is to use small duplex steam pumps or electrical centrifugal pumps. But as the pump cylinders may evacuate during operation, the uncontrolled speeding and priming may result in breakages, and with these pumps complete drainage of the calandrias is not feasible, the lower tube plate generally being covered with condensate for a few inches, thus reducing the effectiveness of the heating surface.

For this reason the *Edwards Wet Vacuum Pump*, shown in *Fig.* 444, has been used to advantage, as not only is the condensate, but also possible air or incondensible gases are removed from the calandrias. The volumetric capacity of the pump should be calculated at about 300–400 per cent. of the condensate volume. The Edwards pump plant generally comprises three pump bodies for removal of the condensate of the three last bodies of a quadruple effect and one pump of smaller diameter for pumping the syrup from the last body, all driven by one attached steam cylinder or electric motor.

The *modus operandi* is as follows : The piston *a*, having a bronze bushing with labyrinth grooves, moves in a bronze liner, having slots *b*, which cause a corresponding reduction in the effectiveness of the full pump stroke. As the conical bottom of the piston coincides with the bottom of the pump body, it will be clear that the condensate which has entered the pump body through the inlet *c* will be thrown into the pump cylinder at the downward stroke of the piston and it will be expelled through the valves *d* at the upward stroke of the piston. With the use of slots *b*, suction valves are not required. The discharge valves are of the single or multiple disc type of metal (Kinghorn), the pump discharge being at *e*. Air vessels are provided on the suction as well as on the discharge side and the former have sight glasses to observe the condensate flow. Snuffling valves are placed on the pump body to allow a small air suction for cushioning purposes.

The piston rod is also made of bronze or covered with a bronze sleeve, and the piston rod packing box prevents the entry of air, which would impair the efficiency of the pump.

These pumps are built for vertical, as well as for horizontal, arrangement and the operating performance, from the author's experience with a dozen of these pumps, has been satisfactory.

4.—General Arrangements of Evaporators.

Multiple evaporators are generally laid out in a straight line, the vapour lines sometimes at the axis of the component bodies. More common is the *Zig-zag Vapour Pipe Arrangement* shown in *Fig.* 445, by which reduced floor space is ensured. The evaporators are built on platforms with sufficient headway underneath for the juice and condensate connexions, as well as for the necessary pumping plant. For barometric connexions a height of 20 ft. from the floor to the top of the supporting beams will be sufficient.

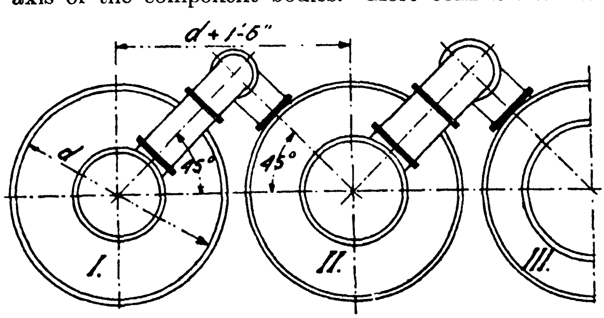


Fig. 445.—Zig-zag Vapour Pipe Arrangement.

The vapour pipes are of increasing area towards the last body. The weight of vapours to be carried will be more or less equal for all the bodies, but the volume expands, owing to the increasing vacuum. For a quadruple effect, the pressures (vacua) and the corresponding volumes are as follows :—

1st body	steam belt,	3 lbs. gauge,	22·53 cub. ft./lb.,		proportion	= 1·00
1st body	vap. space,	2 in. vac.,	28·57 cub.ft./lb.		,,	= 1·27
2nd body	,,	10 in. vac.,	39·13	,,	,,	= 1·76
3rd body	,,	18 in. vac.,	63·10	,,	,,	= 2·80
4th body	,,	26 in. vac.,	176·7	,,	,,	= 7·80

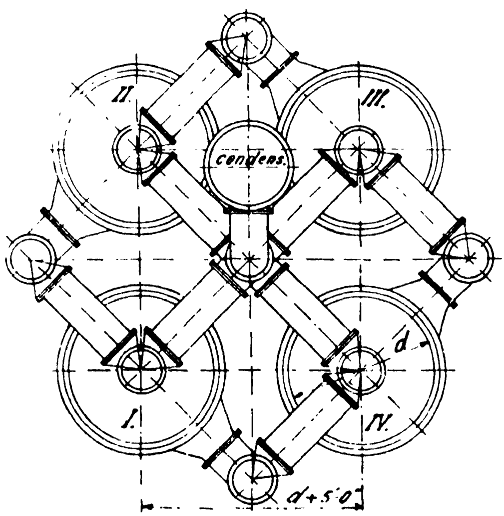


Fig. 446.—Quadrant Arrangement.

The velocity of the exhaust steam to the first body should not be taken with a flow above 150 ft./sec., whereas the maximum for the vapours to the condenser may be well over 200 ft./sec.; but the author has records of heavy entrainment losses with 160 ft./sec. for the last body of a quadruple effect, so velocities should be kept as low as possible. Moreover high vapour velocities will cause higher resistance, and a part of the pressure difference between the evaporating bodies will be hampered and thus limit the evaporative capacity.

In *Fig.* 446 a *Quadrant Arrangement* of a quadruple effect is shown, occupying a very reduced floor space. The condenser is also arranged within this quadrant and it will be seen that each of the bodies can be connected to the condenser by corresponding valves, but for high evaporation rates, the vapour pipes have to be calculated for the largest vapour volumes, which is costly.

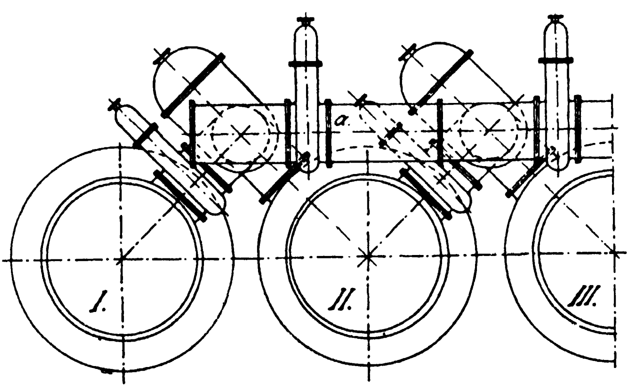


Fig. 447.—Through Connected Multiple Evaporator.

The angled valves have cast iron valve discs with a hard rubber seal. The vapour valves are operated by chain wheels from the evaporator floor.

A *Through Connected Multiple Evaporator* is shown in *Fig.* 447 where the interconnecting vapour line *a* gradually increases in area towards the last evaporating body. Each evaporator requires three large gate valves, which make this arrangement costly, and it is only practicable in those instances where a weekly cleaning of the evaporators is not possible and thus a body has to be cut out during operation for cleaning purposes.

It may be useful to mention that after evaporators have been boiled out with hydrochloric acid, they should not be entered with an open light, as explosive gases might have formed.

As already stated, a straight multiple effect should have bodies of equal heating surface, as the evaporator generally will reach a capacity proportionate to the smallest body in the train.

In cases where the first body or the first two bodies are supplying vapours to juice heaters or vacuum pans, thus when using so-called *pre-evaporators* or *vapour cells*, the bodies which supply vapours for outside demand have to possess the requisite additional heating surface.

As heat losses are considerable when the evaporators are not protected by insulation, all bodies should be covered with insulating material and, obviously, the pressure bodies with a heavier layer than the ones under partial vacuum, the temperature difference with the surrounding air being the deciding factor.

The materials mostly used for heat insulation of evaporators are hair felt and asbestos-magnesia, the first being supplied in sheets of about 1 in. thickness, whereas asbestos-magnesia is mixed with water and put on 2 in. thick, encased in wire netting so that it will not easily fall off. A *wood lagging* of oak boards of about 2 in. × ½ in. is generally attached on top of the insulating material. The boards are varnished alternately light and dark for esthetic reasons, but for economic ones, the wood lagging is not always applied. Brass bands with a screw tightening attachment hold the boards (which are tongued and grooved) firmly together after they have been previously nailed to wooden battens of circular form attached around the cast iron or steel belts of the evaporator. The lagging must be dry and should not be laid on unprotected surfaces, as it will warp under too high a temperature.

For a 2⅜ in. insulation of kieselguhr, SANDERA[1] has found a heat loss of 30 B.Th.U./sq. ft./hr. (83 cal/m²/hr.) exposed surface on evaporators, the heat drop between the vapour and the surrounding air being 126°F. (52°C.).

Vapour lines to the condenser, have not, of course, to be insulated but other uncovered surfaces will give rise to heavy heat losses, and overall practical data noted by the author, with insulated bodies but uncovered vapour lines, gave a loss between 3 and 10 per cent. of the corresponding heat exchange in the body. When high rates of vapour flow prevail, these heat losses increase.

For the total heat distribution in a multiple evaporator, a heat balance of the incoming and outgoing heat in the steam, juice, vapours and condensate should be computed. In Chapter XXXIII these heat balances are dealt with.

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1 See *Zeitschr. Zuckerind. Czechoslov.* 1935, abstracted *Int. Sugar Jl.*, 1935, p. 227

CHAPTER XXII.

VACUUM PANS.

The final and intermittent stage of evaporation is carried out in vacuum pans, during which process the supersaturation of the sugar solution develops to such an extent that crystallization of the sucrose takes place. Boiling in the vacuum pans is conducted below atmospheric pressure, as indicated by the name of the equipment in which it is performed. The purpose of boiling under vacuum is to use as low a temperature as possible in the pan, and to avoid dark colouring of the massecuite through excessive heat. In the previous chapter an explanation has been given of the influence of the rise in boiling point caused by the concentration and the hydrostatic head of the column of juice or massecuite. These two factors are of great importance, especially in the case of the vacuum pan.

1.—Principles of Boiling.

As with the evaporator performance, the densities of the massecuites in the vacuum pans are given in degrees Brix in the laboratory reports for factory control, and a slight error will occur when mechanical calculations are based upon these, as the Brix measure is a hydrometer for pure sucrose solutions and its graduation does not give a true indication when the solutions are impure, as is the case in a sugar factory. Nevertheless, the difference is fortunately of no practical value.

The concentrated juice, called syrup, thick-juice or meladura, having normally a density of about 60° Brix when it leaves the evaporator, has now to be concentrated further up to 96° Brix or more. The quantity of water to be evaporated can be easily derived from formula (105), when the weight of the syrup is known.

In the previous Chapter a 75 per cent. evaporation in the evaporators has been assumed and it is obvious that the remaining 25 per cent. will be syrup. The normal evaporation in the vacuum pan being considered from 60 to 96° Brix, the quantity of water evaporated amounts to :—

$$W_{ep} = W_s \times (1 - 60 \div 96) = 0{\cdot}375\ W_s$$

in which W_{ep} is the quantity of water evaporated in the vacuum pan and W_s the weight of the syrup. The formula serves equally for the British as for the metric system.

Expressed on diluted juice, the percentage is considerably less, it being :—

$$W_{ep} = 0{\cdot}375\ W_s = 0{\cdot}375 \times 0{\cdot}25\ W_j = 0{\cdot}09375\ W_j$$

in which W_j is the weight of the diluted juice entering the evaporator.

It will be seen, later, that all the sucrose present in the massecuite cannot be crystallized out ; and to obtain a commercial yield of sugar, consecutive boilings are required, the latter ones being built up partly from molasses of a previous strike. These molasses have been separated from the sugar crystals in the centrifugals at a density normally of between 85 and 90° Brix. As a thorough mixture of these molasses with the syrup or massecuite in the pan is of paramount importance, it needs diluting since reducing the density will cause a reduction in the degree of viscosity, and dilution of the aforementioned molasses is therefore established practice. This dilution water, as well as the wash-water W_w for eventually dissolving false grain, has to be evaporated and the general formula will read :—

$$W_{ep} = W_s \times (1 - B_1 \div B_2) + W_m \times (1 - B_m \div B_2) + W_w \quad .. \ (112)$$

The signification of the symbols is (as far as not previously mentioned) :—

B_1 = Brix of syrup.
B_2 = Brix of massecuite, when the pan is discharged.
B_m = Brix of molasses as charged into the pan.
W_m = Weight of molasses as charged into the pan.

It will be obvious that when the molasses is diluted to the same Brix as the syrup, which represents the most favourable condition for proper mixing in the pan, thus $B_m = B_1$, then the formula can be abbreviated to :—

$$W_{ep} = (W_s + W_m) \times (1 - B_1 \div B_2) + W_w \ \ldots\ldots \ (112a)$$

The specific weight of the sugar solutions increases with concentration and a *Specific Gravity Chart of Sugar Solutions* in pure water is shown in *Fig.* 448. At 0° Brix, the specific gravity is 1 (pure water), whereas pure sucrose

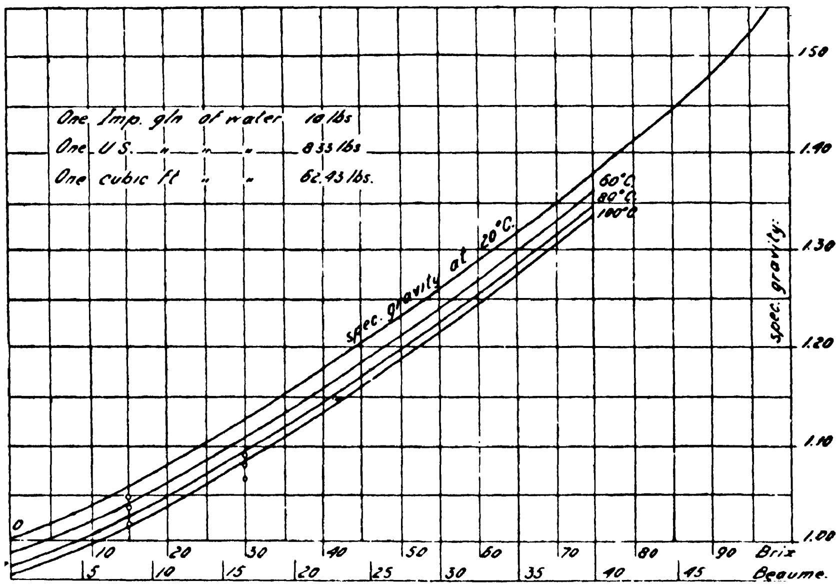


Fig. 448.—Specific Gravity Chart of Sugar Solutions.

has a specific gravity of 1·585 and all the possible sugar solutions necessarily have a specific weight lying between these limits. The specific gravities are given for the standard temperature of 20°C. (68°F.), but as this temperature does not rule in a vacuum pan, the additional specific weights for temperatures of 60, 80 and 100°C. (140, 176 and 212°F.) are given. Below the Brix scale, the corresponding values according to the scale of Baumé have been inserted ; moreover, for conversion of the specific weight into the weights per Imp. and U.S. gallon, as well as per cub. ft., the corresponding weights of pure water are mentioned.

The specific weights of the sugar solutions are required for calculating the weight of the massecuite in a pan of a given size, and also for calculating the hydrostatic head, acting above the heating surface.

Crystallization of sucrose is only possible when the sugar solution becomes supersaturated, and the degree of supersaturation is generally kept between

1 and 1·2, as impure sugar solutions are being treated. It should be added that crystallization is not a spontaneous phenomenon, but requires what in chemistry would be called a catalyzer, this being an embryo, which is present in the air or inside the pan. "Chock graining," which is the introduction of a small quantity of powdered sugar into a vacuum pan in which supersaturation has been brought well above the saturation point, is also to be explained by this phenomenon. In an absolutely sterile enclosure, a pure sugar solution can be concentrated considerably above the saturation point without giving rise to crystallization, as has been observed by several Russian technologists, although this criterion is not universally accepted. Fortunately in practical pan boiling, such a sterile condition does not exist.

The *Saturation Curve* depicted in *Fig.* 449 (sucrose per cent. solution drawn in chain dotted line in the diagram) shows that the temperature of the solution has a marked influence on the saturation limit, a higher temperature corresponding with a higher quantity of sucrose dissolved per cent. solution. In the same figure the *Dissolvent Curve* (sucrose per cent. water, drawn in full

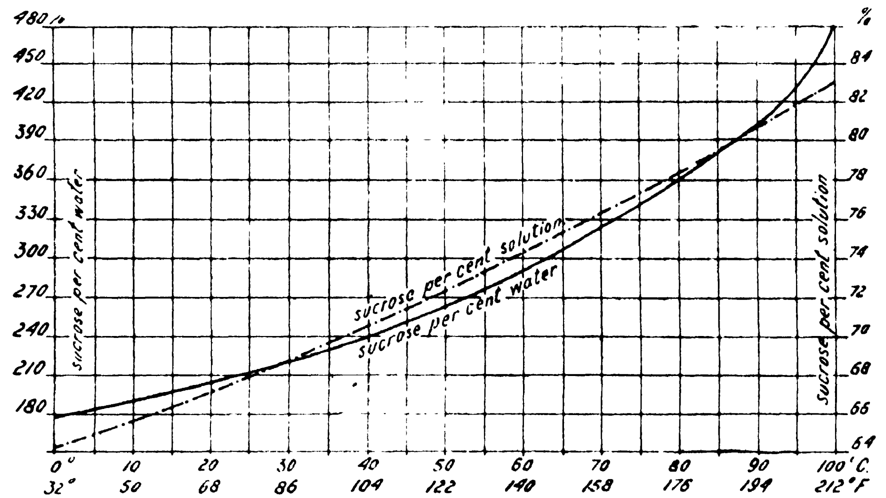


Fig. 449.—Saturation Curves.

line in the diagram) synonymous for saturation curve per cent. water, has also been drawn. Given a prevailing temperature of 70°C. (158°F.) in a vacuum pan, the saturation point lies just above 76 sucrose per cent. solution or slightly above 76° Brix. It will also be observed that when cooling down a massecuite, as is done in the crystallizers (to be dealt with in a later Chapter), the saturation point falls too.

When a saturated sugar solution of, e.g., 78° Brix (on the right hand horizontal line of the diagram at 78), having a temperature of about 80°C. (176°F.) is cooled down to 40°C. (104°F.), the corresponding saturation point (more to the left in the diagram) will be 70·5 per cent. sucrose in solution; thus under the optimum conditions assumed, there will crystallize 78 — 70·5 = 7·5 per cent. sugar per 100 solution.

A sucrose solution occupies less volume than the volumes of water and solid sucrose combined. This contraction is highest at a concentration of 56 per cent., it being then 1 per cent. in volume but is of no practical value as far as it refers to the design of the vacuum pan.

Another important factor for the boiling of sugar solutions is the *Quotient of Purity*, as on it the yield of commercial sugar will largely depend. In the analytical reports supplied by factory chemists the true purity coefficients are seldom mentioned, and generally only the apparent purity quotient, or the "apparent purity" as it is called, will be found. The difference between the true or absolute purity and the apparent one is due to the imperfections of the apparatus used for measuring, this being the Brix spindle already referred to, and the polariscope. The latter instrument is based on the fact that a beam of light sent through a sugar solution will be rotated and this rotation is measured, according to a scale in percentages of sugar. A polariscopic reading of 90 indicates 90 per cent. sucrose in pure solution. The non-sugars in solution which lead to small errors in the Brix readings will also cause a faulty rotation of the beam of light and thus make the polariscopic readings of only approximate value.

The apparent purity is calculated according to the formula :—

$$Pur_{app.} = \text{Pol.} \div \text{Brix} \times 100 \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (113)$$

It will be obvious that the true or absolute purity has to fulfil the equation :

$$Pur_{abs.} = \text{Sucrose in solution} \div \text{Total solids in solution}$$

or :

$$Pur_{abs.} = \text{Su} \div \text{So} \times 100 \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (113a)$$

The absolute purities are of course more valuable for technical calculations than the apparent ones, but the factory engineer has to accept the figures given him by the laboratory staff and small errors in the calculations may therefore occur ; but generally these are of no importance to the engineering side of the design or operation of vacuum pans.

A high concentration of the massecuite in the vacuum pan will give a high yield of sucrose. The weight of the massecuite will always be the sum of the weight of the sugar crystals plus the weight of the molasses, thus, with insertion of the Brix densities :—

$$B_{mas} = [100x + B_{mol}\,(100 - x)] \div 100$$

in which : B_{mas} = Brix of massecuite.

B_{mol} = Brix of molasses, derived from this massecuite.

x = Per cent. of sugar in crystal form present in massecuite.

From this formula is easily derived the per cent. crystals by weight when the Brix of the massecuite and the molasses derived therefrom are given, thus :

$$x = \frac{B_{mas} - B_{mol}}{100 - B_{mol}} \times 100 \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (114)$$

There is no fixed relation between the density or Brix of the massecuite and the Brix or density of the molasses produced therefrom, as this relation depends upon how much sucrose has been crystallized out, the crystallization sometimes being hampered by faulty circulation and high viscosity. But even when crystallization can proceed undisturbed, there is a practical limit, which can be explained as follows :—

In textbooks on sugar manufacture, the specific weight of sucrose is given as 1·585, but in crystal form, as it is obtained in the sugar factories, the specific weight is considerably less. The author has ascertained on different occasions the specific weight of grained sugar, which of course varies with the size and regularity of the crystals produced. In general, it can be accepted that the average value is 0·8 ; fine granulated sugar weighing about 5 to 8 per cent. more, and coarse sugar less, so about 50 per cent. of the volume is taken up by voids.

It will now be apparent that in a massecuite only about 50 per cent. of the volume can be occupied by crystals, the rest being molasses. This nevertheless is an optimum condition, as the crystals are already touching each other, so a circulation of the massecuite in the pan will thus no longer obtain and overheating of the sugar may result.

The purity of the molasses produced from a massecuite of a given purity can also be easily determined, when the per cent. of crystals is known. PRINSEN GEERLIGS long ago established the following formula, which is identical with (114) :—

$$100\ \mathrm{Pur}_{mas} = \mathrm{Pur}_{s} \times x + \mathrm{Pur}_{mol}\ (100 - x) \ldots\ldots (115)$$

and the formula can also be written :—

$$x = \frac{\mathrm{Pur}_{mas} - \mathrm{Pur}_{mol}}{\mathrm{Pur}_{sug.} - \mathrm{Pur}_{mol}} \times 100 \ldots\ldots\ldots\ldots\ldots\ldots (115a)$$

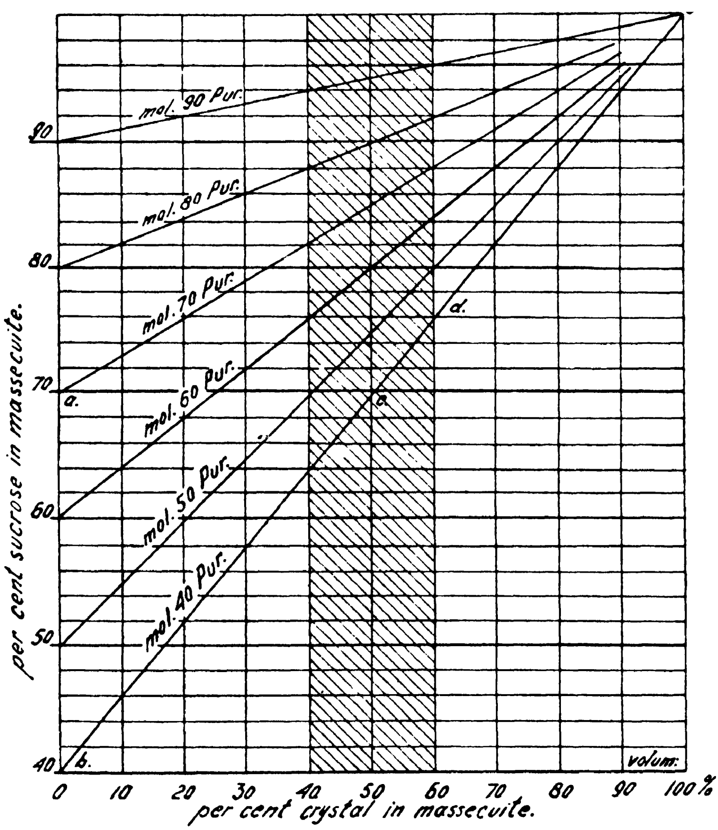
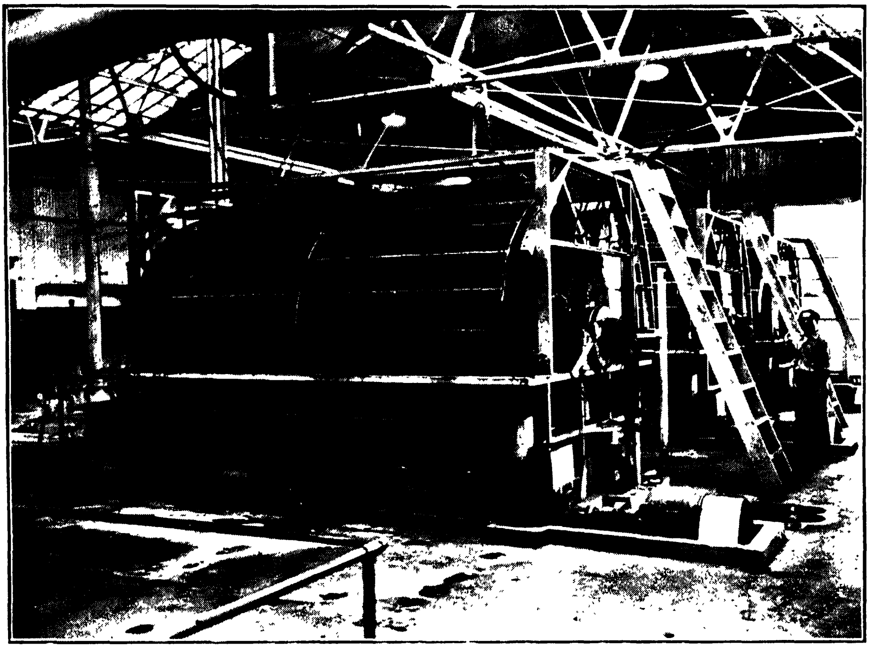
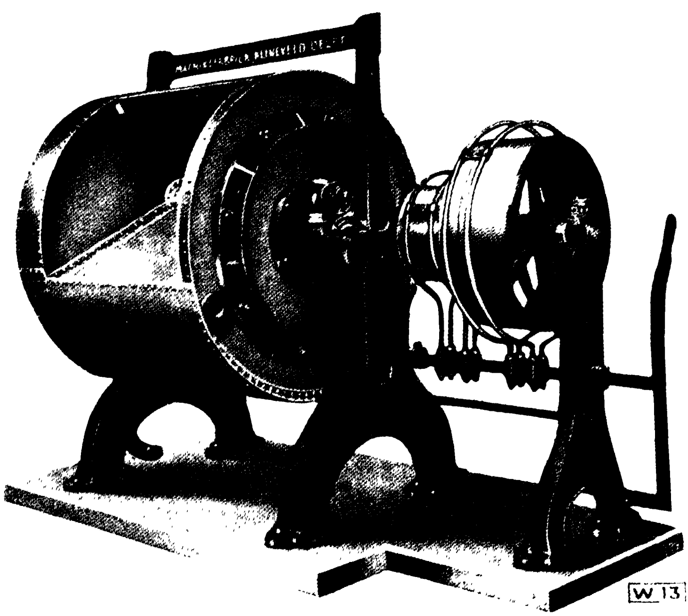


Fig. 450.—Crystal Yield Diagram.

In *Fig.* 450 a *Crystal Yield Diagram* is given, the yield being shown in per cent. of the volume of the massecuite. Taking, e.g., a massecuite with a purity of 70 at *a* and following the horizontal line until 50 per cent. crystal at c, the intersection is on the line *bo*, representing the purity line of 40 for molasses. It will be obvious that from a massecuite of 40 assumed purity, with 0 per cent. crystal, the molasses will have this same purity, and with

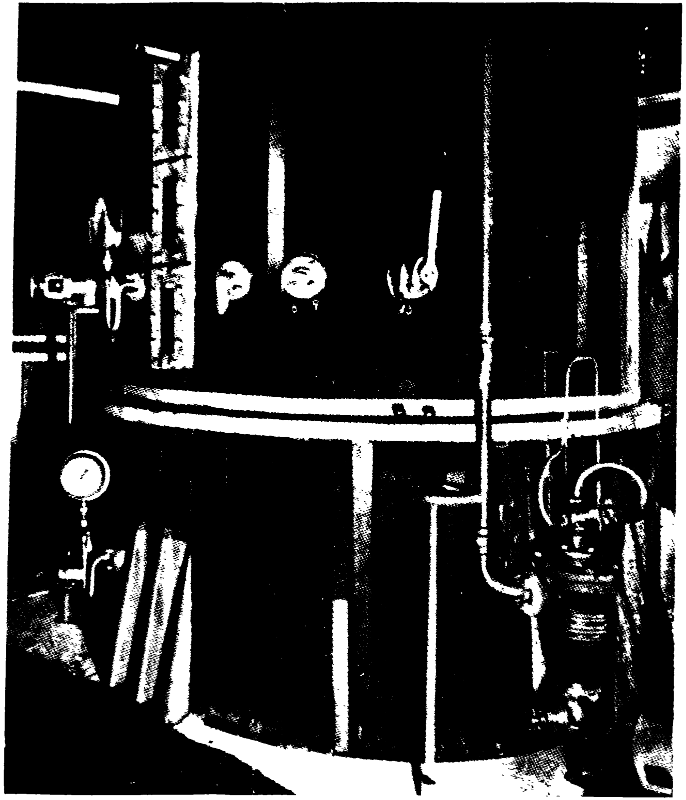


INSTALLATION OF OLIVER-CAMPBELL FILTERS FOR CACHAZA IN HAWAII.
(Oliver-United Filters, Inc.)



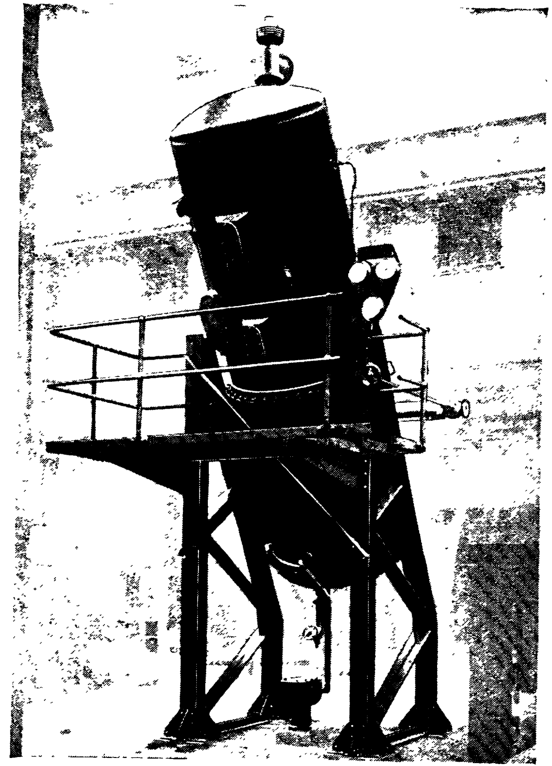
SPECIAL WASHING MACHINE FOR FILTER CLOTH.
(Reineveld N.V.)

PLATES 85 & 86.



THERMO-FEED REGULATOR FOR JUICE LEVEL CONTROL ON A PRE-EVAPORATOR.

(Ronald Trist & Co., Ltd.)



PATENT INCLINED EVAPORATOR WITH STAGING.

100 per cent. crystal assumed, there would be no molasses produced at all, and the massecuite must contain 100 per cent. sucrose. With a very fine grain 60 per cent. of the volume of the massecuite can be obtained in crystal form, and the diagram indicates clearly that a massecuite of 76 pur. at *d* will yield a molasses of 40 pur. Absolute purities of course are meant, as apparent purities are only approximate figures.

The shaded area represents the practical field in which the per cent. crystal per volume will generally range.

The *purity drop* will be less for high grades than for low ones, and this drop, i.e. the difference between the purities of the massecuite and of the molasses produced therefrom, is the deciding factor as to how many boilings will be required to exhaust the molasses until the practical limit of about 36 absolute purity.

Taking 50 per cent. on volume as the average quantity of crystals to be obtained, formula (114) can be written :—

$$100\ B_{mas} = 100 \times 50 + 50\ B_{mol}$$

$$B_{mol} = \frac{100\ B_{mas} - 5000}{50}$$

thus : $B_{mol} = 2 \times B_{mas} - 100$ .... (116)

Similarly formula (115) will yield :—

$\mathrm{Pur}_{mol} = 2 \times \mathrm{Pur}_{mas} - 100$ .... (116a)

For overall calculations, the following Brix tables can therefore be compiled :—

A massecuite of	96°	Brix	yields	molasses of	92° Brix.
,,	95°	,,	,,		90° ,,
,,	94°	,,	,,		88° ,,
,,	93°	,,	,,		86° ,,
,,	92°	,,	,,		84° ,,
,,	91°	,,	,,		82° ,,
,,	90°	,,	,,		80° ,,
,,	89°	,,	,,		78° ,,
,,	88°	,,	,,		76° ,,

The absolute purities and the purity drops according to (116a) can be tabulated as follows :—

A massecuite of	90	Abs. Pur.	gives	molasses of	80	Abs. Pur.,	drop 10
,,	88	,,	,,	,,	76	,,	,, 12
,,	85	,,	,,	,,	70	,,	,, 15
,,	82	,,	,,	,,	64	,,	,, 18
,,	80	,,	,,	,,	60	,,	,, 20
,,	70	,,	,,	,,	40	,,	,, 30
,,	65	,,	,,	,,	30	,,	,, 35

From practical observations PRINSEN GEERLIGS has found[1] the following results without cooling the massecuites, and indicated by *apparent* purities :—

A massecuite of	90	App. Pur.	yields	molasses of	75	App.	Pur.,	drop 15
,,	88	,,	,,	,,	71	,,	,,	,, 16
,,	85	,,	,,	,,	65	,,	,,	,, 20
,,	82	,,	,,	,,	62	,,	,,	,, 20
,,	80	,,	,,	,,	60	,,	,,	,, 20
,,	70	,,	,,	,,	50	,,	,,	,, 20
,,	65	,,	,,	,,	45	,,	,,	,, 20

1 See "Cane Sugar and its Manufacture," p. 229.

For practical calculations, where the apparent purities are given, the last-mentioned data should be followed. A higher drop in apparent purity than 20 is not given, but nowadays with improved circulation in the vacuum pans, apparent purity drops of 30 to 35 for the low grades have been obtained.

The pan capacity must obviously be calculated from the total quantity of massecuite produced per 24 hours and this point has to be considered carefully. A certain boiling scheme, as outlined below, has been assumed, although any other scheme will not affect the trend of the calculation :—

Boiling scheme : First massecuites built up from syrup alone.
Second massecuites built up from syrup and first mol.
Third massecuites built up from syrup and second mol.

The syrup may be charged direct into the pans or be already grained in another pan, this performance being termed *seeding* or overcutting. The purities are assumed as follows :—

First and second sugars	96	App.	Pur.
Third sugar	90	,,	,,
Syrup	85	,,	,,
First boilings	85	,,	,,
Second boilings	72	,,	,,
Third boilings	60	,,	,,
First molasses	65	,,	,,
Second molasses	50	,,	,,
Final molasses	30	,,	,,

First boilings : Using formula (115) it will be found that the first boilings furnish : $(85 - 65) \div (96 - 65) \times 100 =$ 64·5 per cent. sugar
and 35·5 per cent. first molasses

Second boilings : Are built up from :—
$(72 - 65) \div (85 - 65) \times 100 =$ 35 per cent. syrup
and 65 per cent. first molasses

and will furnish :—
$(72 - 50) \div (96 - 50) \times 100 =$ 48 per cent. sugar
and 52 per cent second molasses

Third boilings : Are built up from :—
$(60 - 50) \div (90 - 50) \times 100 =$ 25 per cent syrup
and 75 per cent. second molasses

and will furnish :—
$(60 - 30) \div (90 - 30) \times 100 =$ 50 per cent. sugar
and 50 per cent. final molasses

The quantitative proportions in these boilings are :—

First boilings : 100 syrup furnishes .... 100·0 massecuite

Second boilings : the available 35·5 parts first molasses require $35{\cdot}5 \times (35 \div 65) = 19$ parts syrup ; total .... 54·5 ,,

Third boilings : available $54{\cdot}5 \times 0{\cdot}52 = 28{\cdot}3$ parts second molasses, which require $28{\cdot}3 \times (25 \div 75) = 9{\cdot}4$ parts syrup ; total .... 37·7 ,,

Total 128·4 syrup + 35·5 first mol.
+ 28·3 second mol. .... 192·2 massecuite

As final molasses, $37{\cdot}7 \times (50 \div 100) = 18{\cdot}9$ parts will be produced, and per 100 parts virgin syrup, the figures are :—

Syrup	100 parts
First molasses	27·6 parts
Second molasses	22 parts
Total	149·6 parts massecuite
Final molasses	14·7 parts

According to this boiling scheme and the assumed purities, about 150 parts by weight of massecuite will be produced per 100 parts syrup. But it should not be overlooked that the syrup has a density around 60° Brix and the massecuites about 94° Brix, thus the volume and weight are greatly reduced. For the pan capacity, the calculation should be based on about 85° Brix, thus :

$$W_{mass} = W_s \times (60 \div 85) \times C = 0{\cdot}7\ W_s \times C$$

in which W_s is the weight on syrup and C the coefficient of multiplication for the multiple boiling system, thus 1·5 (150 ÷ 100) as in the above-mentioned example ; but this might be increased to 2·5 when other purities prevail.

The specific gravity of massecuite at 60°C. (140°F.), and 85° Brix is about 1·43 and the weight of 1 cub. ft. will thus be 1·43 × 62·43 = 89·2 lbs. ≃ 90 lbs. and the volume in cubic feet of the massecuite V_{mass} will amount to :—

$$V_{mass} = M_{mass} \div 90 = (W_s \times C) \div 130 \quad \ldots\ldots \quad (117)$$

It being assumed that a ton of cane produces 2150 lbs. diluted juice, which after evaporation yields 540 lbs. syrup of 60° Brix, and taking C between 1·5 and 2·5, it is easily seen that 6 *to* 10 *cub. ft. of massecuite will be produced per ton of cane* (1·7–2·8 hl. per ton of cane and 85° Brix).

The next factor in vacuum pan dimensioning will be the time t, in which a strike can be completed, i.e., the time from when the vacuum pan starts charging until it is again ready for charging after the previous strike has been discharged, including time for cleaning the pan interior with steam.

High purity syrups or liquors, as used in refineries and for direct consumption sugar manufacture, require only a short strike, the author having seen strikes finished within 1½ hours. The first boilings of raw sugar houses in Cuba require 2½ to 4 hours, second boilings 4 to 6 hours, and third boilings 6 to 8 hours as an average, with rapid circulating pans having sufficient heating surface. In Java about twice this time is considered as standard for completing a strike. The factors of a mechanical nature, which have a bearing upon the time of strike, are :—

(*a*) Heating surface.
(*b*) Purity of the massecuite.
(*c*) Viscosity of the massecuite.
(*d*) The circulation inside the pan.
(*e*) The vacuum and temperature difference between the massecuite and the heating steam.
(*f*) The prevailing hydrostatic head under which the massecuite is boiling.

Taking P as the total pan charging volume, it will be seen that for a total volume V_{mass} per 24 hours (both in cub. ft. and in hl.) the following equation holds : $V_{mass} = P \times 24 \div t$
and the required pan capacity in cub. ft. is :—

$$P = V_{mass} \times t \div 24 \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (118)$$

Whenever possible, the pan capacity should be apportioned according to the number of boilings to be made, and average figures for this division are :—

For 3 boilings : Proportion I : II : III = 10 : 6 : 8 total 24
For 4 boilings : Proportion I : II : III : IV = 8 : 7 : 5 : 4 total 24

The coefficient of heat transmission in the vacuum pan is low, due to viscosity, a small temperature difference at the lower levels and insufficient circulation, all of which may be derived from the *Vacuum Pan Evaporating Chart* shown in *Fig.* 451; the tests were taken on 12 ft. calandria pans, having 2100 sq. ft. heating surface and 1200 cub. ft. capacity, the heating medium being exhaust steam of about 5 lbs. per sq. in. The highest evaporation rate has been obtained by the seed strike *S* of virgin syrup, the consecutive strikes *A*, *B* and *C* being from syrup with molasses. The highest evaporation takes place at the lowest density and it soon drops rapidly. A maximum of 20 lbs. evaporation per sq. ft. H.S./hr. has been reported for calandria pans, coil pans having generally 50 to 100 per cent. more evaporation per sq. ft. due to the use of reduced live steam or to the reduced hydrostatic head.

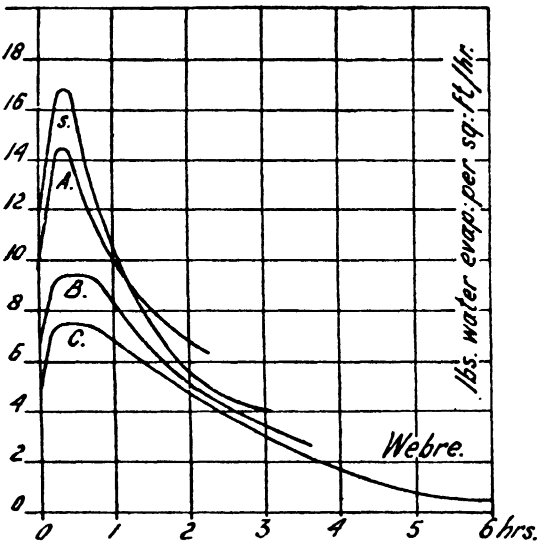


Fig. 451.—Vacuum Pan Evaporating Chart.

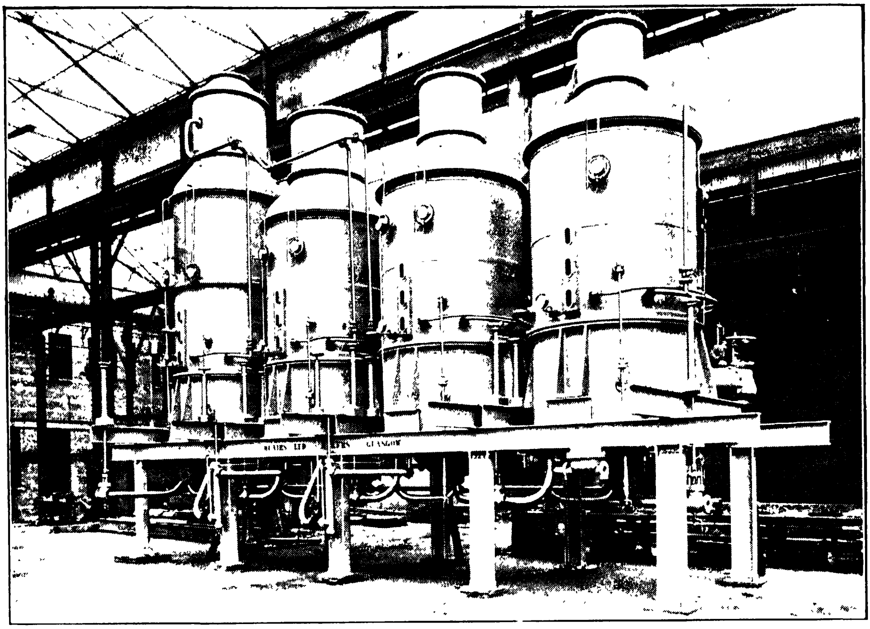
The heating medium has a direct bearing on the heating surface required, as may be gleaned from the following table :—

Steam pressure	30 lbs.	7 lbs.	0 lbs./sq. in.
Steam temperature	274°F.	232°F.	212°F.
Massecuite temp. at 26 in. vacuum with 20°F. temp. rise	145°F.	145°F.	145°F.
Temperature difference	129°F.	88°F.	67°F.
Proportion	1·9	1·3	1

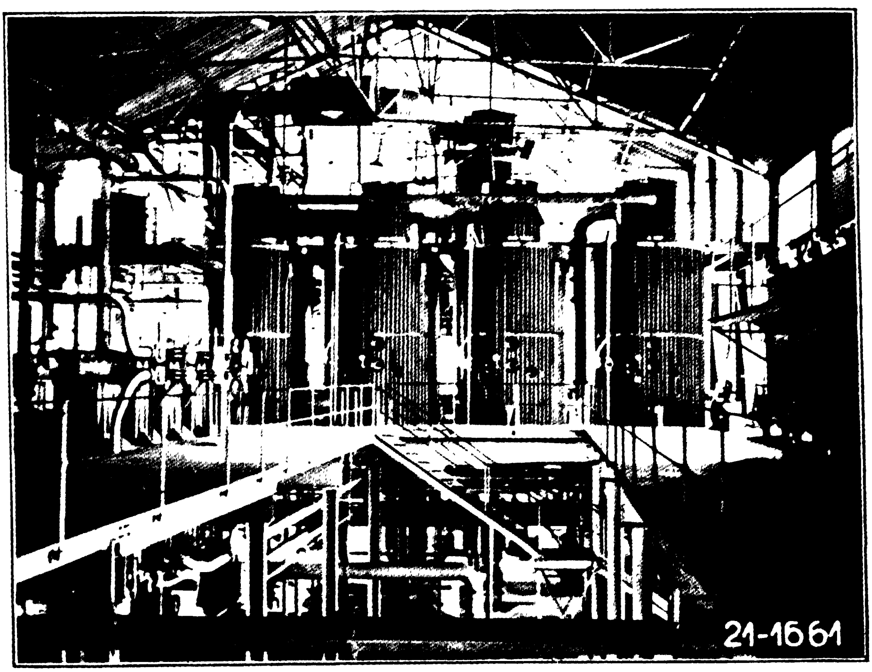
Reduced live steam can be used in coils, whereas in calandrias exhaust steam or vapours from a pre-evaporator are applied.

The *hydrostatic head* of the massecuite above the heating surface is of paramount importance compared with the performance in the evaporators. Needless to say, there is no film formation, and the massecuite level is kept in calandria pans 6 to 7 ft. above the upper tube plate. At 90° Brix, if 7 ft. hydrostatic head and a vacuum of 26 in. prevail above the massecuite level, the following temperature has to obtain to make evaporation possible :—

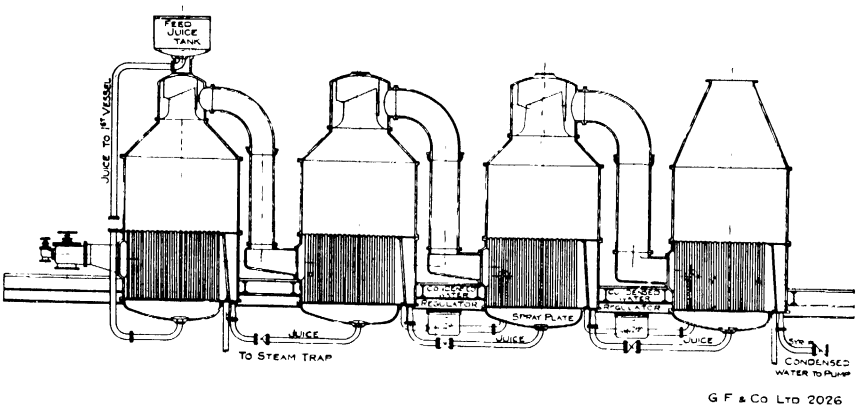
Temperature rise at 90° Brix		40°F.
Hydrostatic pressure $0{\cdot}432 \times 1{\cdot}45 \times 7 =$	4·380 lbs./sq. in.	
Vapour pressure at 26 in. vacuum	1·954 lbs./sq. in.	
Total absolute pressure	6·334 lbs./sq. in.	
Vapour temperature at this pressure		173°F.
Required temperature for evaporation		213°F.



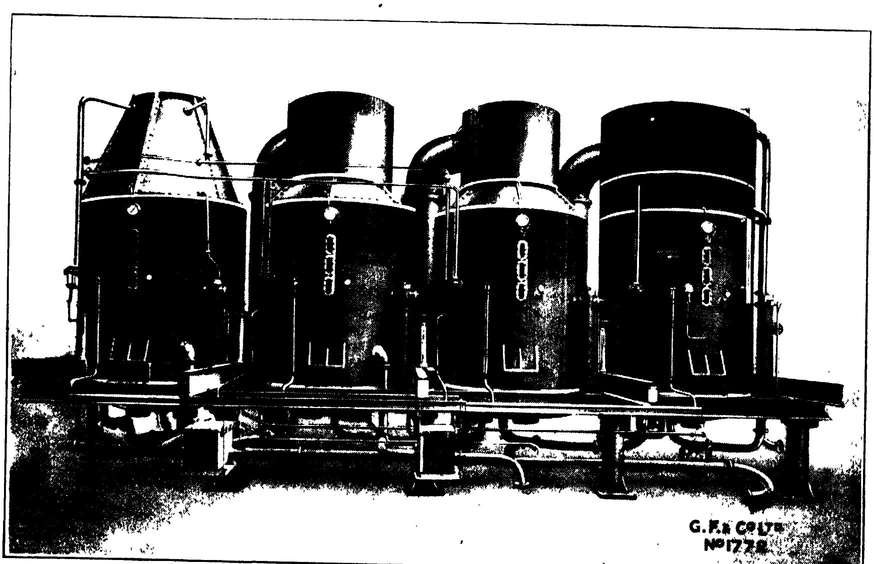
QUADRUPLE EFFECT EVAPORATOR.
(*Blairs, Ltd.*)



EVAPORATING PLANT, AND VACUUM PANS ABOVE, IN CHINESE SUGAR FACTORY.
(*Skoda Works, Ltd.*)



QUADRUPLE EFFECT EVAPORATOR WITH SEALED DOWNTAKES.
(Geo. Fletcher & Co., Ltd.)



QUADRUPLE EFFECT EVAPORATOR WITH SEALED DOWNTAKES.
(Geo. Fletcher & Co., Ltd.)

Evaporation obviously is only possible when the hot massecuite goes to a higher level under reduced hydrostatic, i.e., total pressure, and this explains the primary importance of good circulation in a vacuum pan. A passage has to be provided for both the upgoing and the downgoing streams, the first taking place between the coils or through the tubes of the calandria and the latter through the centre well or downtake. Both should provide a more or less equal passage.

The *steam consumption* of a vacuum pan will vary according to the evaporation in the pan, and formula (107) of the previous Chapter can be applied.

The *heating surface* is proportionate to the pan capacity, up to the following rates :—

For coil pans :
- 1·3 sq. ft. H.S. per cub. ft. pan capacity
- 0·45 m^2 H.S. per hl. pan capacity.

With calandria pans, specially designed for the use of exhaust steam and vapours, these rates rise to :—
- 2·3 sq. ft. H.S. per cub. ft. pan capacity
- 0·8 m^2 H.S. per hl. pan capacity.

The constructional designs of vacuum pans can be divided into three groups, viz. :

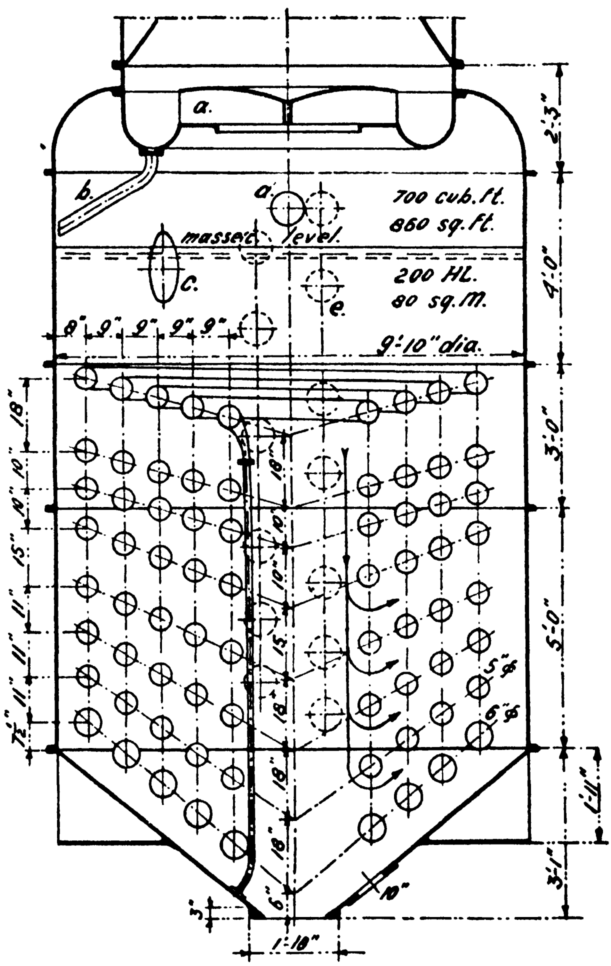


Fig. 452.—Coil Vacuum Pan.

(*a*) Coil vacuum pans.
(*b*) Calandria vacuum pans.
(*c*) Vacuum pans with mechanical circulation.

These three groups will be explained in consecutive sub-headings.

2.—Coil Vacuum Pans.

The heating surface of coil pans is arranged as a spirally wound coil or set of coils, made of copper. In *Fig.* 452 such a *Coil Vacuum Pan* designed by the author is shown, having 6 in. and 5 in. outside diameter coils.[1] The coils are laid in such a way that there is a continuous descent towards the drain for the condensate, so the latter will not accumulate in the coils. The vertical distance between the coils at the centre well is made greater than between those close to the cast iron pan belts, to encourage a good circulation, as indicated by the arrows.

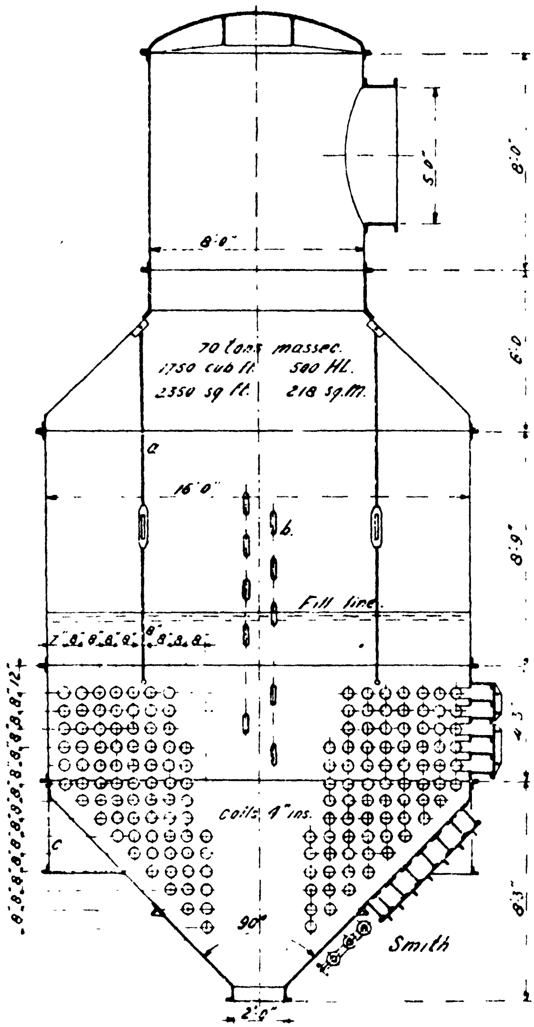


Fig. 453.—Lyre Coil Pan.

The sight glasses *e* at the front of the pan are 8 in. in diameter, being similar to those used in evaporators and only one, marked *d*, is fixed at the rear, above the massecuite level. These sight glasses are so arranged that when the level disappears in one glass, it will appear in the next one above.

A juice catcher *a* is arranged in the upper part of the pan, which discharges through a pipe *b*, and at *c* a manhole is provided, 16 in. in diameter. The bottom discharge valve is not shown, but a 10 in. outlet at the bottom end is for connecting to a so-called graining pan. The vacuum pan therefore starts its operation with a "*footing*," i.e., sufficient fine-grained massecuite to build up crystallization for the complete pan fill.

Each coil has a separate steam trap, a steam and an exhaust valve, as well as a pressure gauge, so it can be cut out independently from the others.

The principal advantage of the coil pan is that the heating surface can be cut in, according to the pan fill and moreover the heating surface can be arranged high up in the pan, thus under a small hydrostatic head of massecuite, which will favour the heat transmission. In some countries the coil pan has been replaced by the calandria pan, which will allow a higher rate of heating surface.

---

[1] When not otherwise mentioned, the U.S. standard of outside diameters is implied.

The pan belts and bottom are of cast iron, the belts covered with hair felt and oak lagging. The bottom generally is not insulated as it is more difficult to do this.

This pan has been in satisfactory operation, and exhaust steam is mostly used, only the lower coils receiving reduced live steam, when the pan has been filled up to its highest level.

An enamelled juice scale is attached to the front of the pan to indicate the pan capacity at different filling levels. An electric bulb is fixed to a slide, which can be raised or lowered on a vertical rod in front of the sight glasses for night work. Above the proof-stick there is also an electric bulb to assist the sugar boiler when he checks the crystallization by spreading the sample from the proof-stick on a piece of glass.

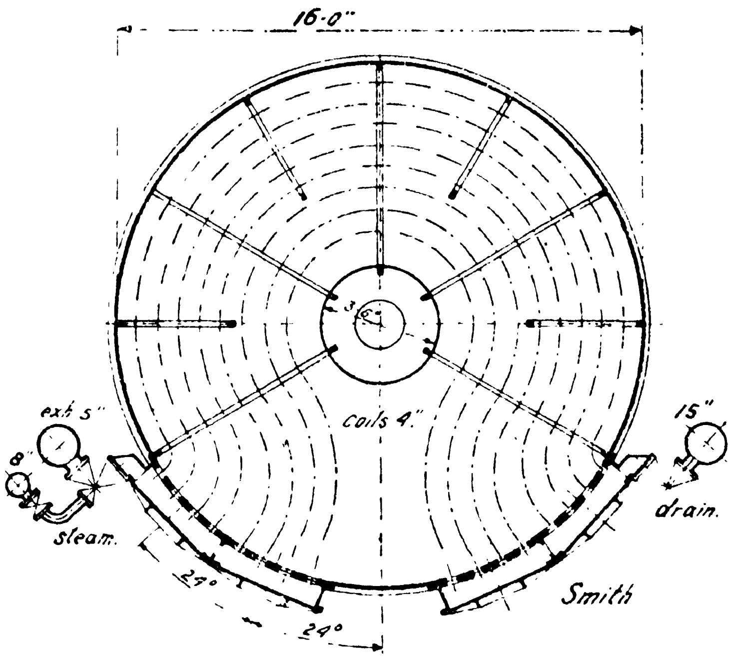


Fig. 454.—Lyre Coils.

Another coil pan, which the author has seen in satisfactory operation, is the *Lyre Coil Pan*, shown in *Fig.* 453. The coils are lyre-shaped and laid in a horizontal plane. *Fig.* 454 shows how the lyre coils are laid, these being charged from the steam headers and discharged into the drain headers. Each set of coils in the same plane can be cut out by closing the steam and drain valves.

A feature of this pan is the low hydrostatic head for its normal rated capacity. Coil pans can start with a small footing, which generally amounts to 25 to 30 per cent. of the complete pan filling.

Another *Arrangement of Coils* is shown in *Fig.* 455, in which each set of coils is split into eight branches of short length. For exhaust steam and vapours, these short coils are of greater effectiveness than long ones. This pan also has proved to be satisfactory under actual operating conditions.

The proportion between coil length and diameter is raised by some designers up to 200—250, but the author has been asked by more than one operating engineer, whether such a coil length is really efficient, as the end may not get heated. A shorter coil length, therefore, is favoured by other designers and it will be obvious that for exhaust steam or vapours above atmospheric pressure the proportion should not be over, say 100 ; and there are now designs with a figure as low as 75, which the author has seen in satisfactory operation for low pressure heating steam.

In those cases where excess of exhaust steam and vapours is pronounced and a saving of live steam is desired, the short coil will prove of great advantage.

The cross section of a large coil pan, 16 ft. in diameter, is shown in *Fig.* **456.** The individual length of the coil branches is from 70 to 60 ft., thus about 160 times the diameter of the tube. The thickness of the coils normally

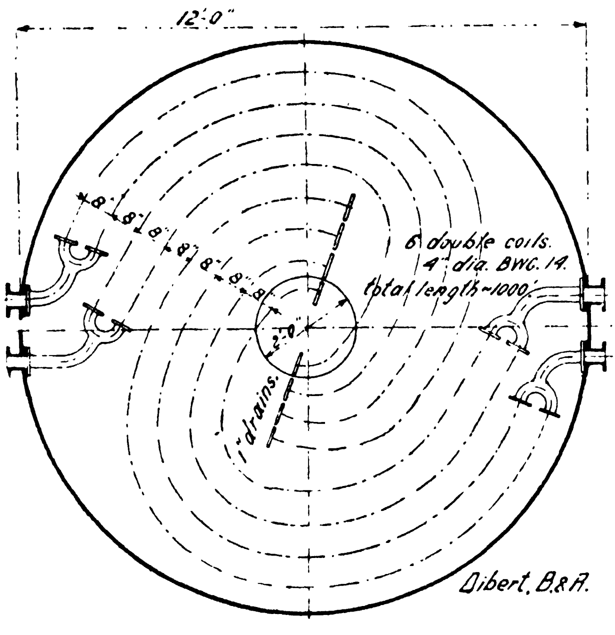


Fig. 455.—Arrangement of Short Coils.

is between Nos. 12 and 14 B.W.G. (0·109 in. to 0·083 in. or 2·76 to 2·10 mm.). As the thickness of the coil for all practical purposes does not impair the heat transmission, a thicker coil will have a longer life and should be preferred, as coils are subject to wear. The author has come across coils which had been for 55 years in uninterrupted operation during crop time, but the wall thickness had decreased to about one-fourth of the original dimension.

Vacuum pans are cleaned just as are evaporating vessels, and the use of caustic soda and muriatic acid has become standard practice in many countries. A copper bend with a valve is connected to the pan belt, so that the muriatic acid can be aspirated by a few inches of vacuum when the pan is partly filled with water.

The material of the pan belts consists of cast iron, steel or copper, the last-named being used only in isolated cases. The coils are always of copper, and the flanges of brass. The coil supports are preferably made of brass, as mild steel is subject to rapid corrosion when strong acids are used for boiling out. The inside of the coils is difficult to clean, this being a disadvantage of the heating coil; and an oil-free heating medium, such as reduced live steam or exhaust steam from a turbine, will not cause inside incrustations. For cleaning, the coils are filled with a caustic soda solution, the pan filled with water and boiling started by means of a perforated steam pipe inside the pan. The author has known cases where the condensate drain of a coil was choked by carbonized lubricating oil present in the exhaust steam.

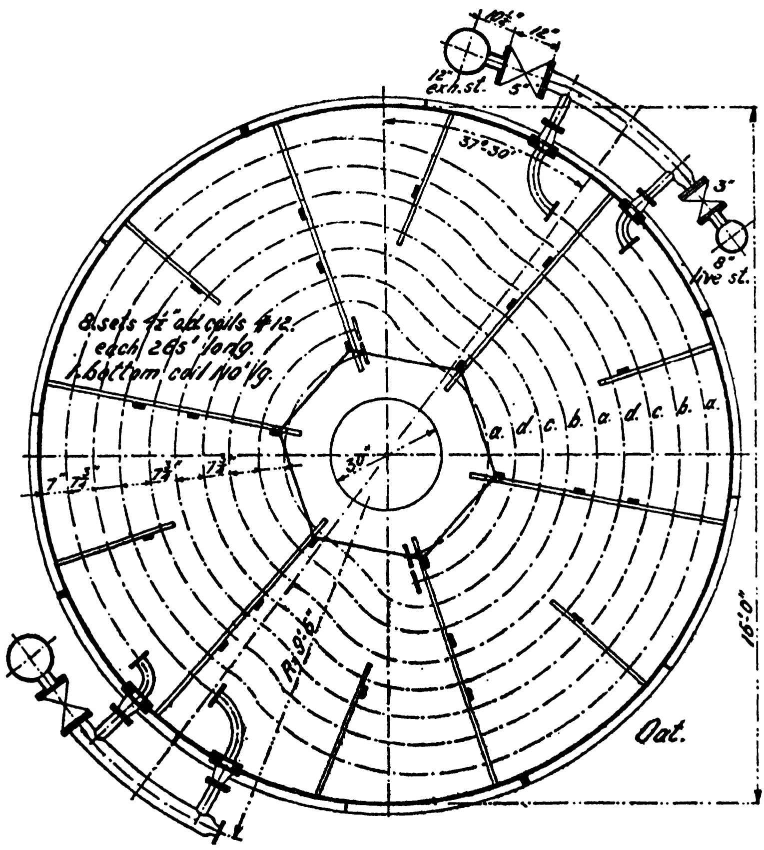


Fig. 456.—Coil Arrangement in Large Vacuum Pan.

Brass *Coil Supports* are shown in *Fig.* 457. These supports are mounted on mild steel bars 4 in. × 1 in., but sometimes brass bars are used. They have to be well secured, as vibrations within the pan are not always avoidable, especially during periods of heavy evaporation. The individual straps are hinged at the centre line and can be easily opened.

An *Integral Brass Coil Support* is shown in *Fig.* 458, which can be readily mounted on the pan bottom and be attached to the pan belts. The coils are not clamped firmly between the different sections, but are allowed some play. Sometimes these supports are made of cast iron provided with a copper ferrule at the point of the coil support.

As the standard length of copper pipes is about 20 ft., coils are generally made up of different sections, connected by flanges of brass, or by a *Double Beaded Coil Joint*, as shown in *Fig.* 459, which occupies less space, but is, of course, a non-detachable union. This joint has to be sealed with hard solder.

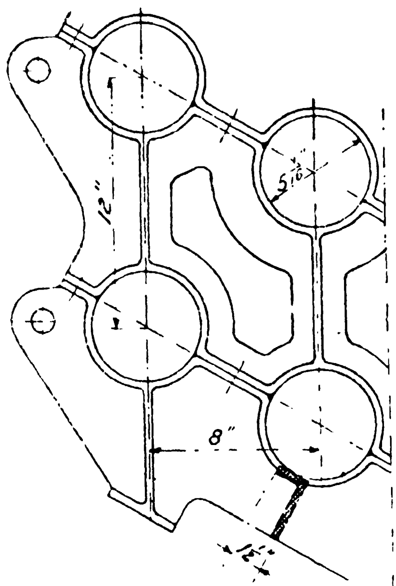


Fig. 458.—Integral Brass Coil Support.

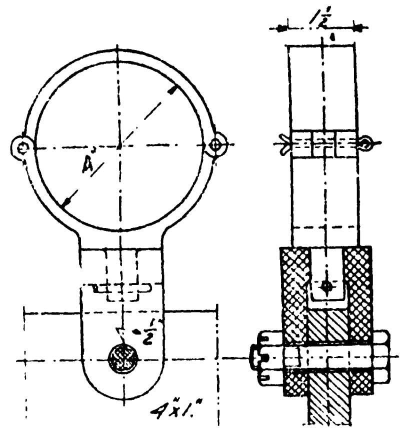


Fig. 457.—Brass Coil Support.

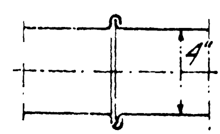


Fig. 459.—Double Beaded Coil Joint.

Bending copper coils is generally done by filling them with rosin or pitch, the ends being closed by wooden plugs. The filling is to prevent flattening or buckling. Copper coils have been bent in the cold while filled with water at 60-75 lbs. pressure, the ends being closed by soldered caps. This method has not always given the desired results, as water is free-flowing, and buckling or flattening has been known to occur. Sand-filling or special pipe-bending machines, for bending pipes in the cold, are now used with good results. The bending force, except in the case of the bending machine, is produced by a tackle or chain hoist on a floor plate.

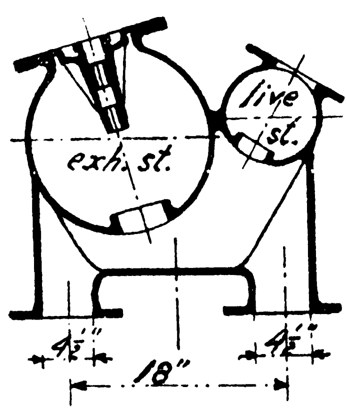


Fig. 460.—Integral Manifold with Incorporated Valves.

The coils are usually heated with either live or exhaust steam, and manifolds with the necessary valves are to be found on each coil vacuum pan. With the object of simplifying the equipment an *Integral Manifold with Incorporated Valves* as shown in *Fig.* 460 has been designed. Between two adjacent coils a division wall is cast in the collector of the manifold.

For heating purposes in vacuum pans, a *Double Bottom*, as shown in *Fig.* 461, has found wide application in Java, this being heated with live steam. As the bottom sheets are of rather thin material, staybolts have to be provided between them. The material is steel, although the inside sheet has been made

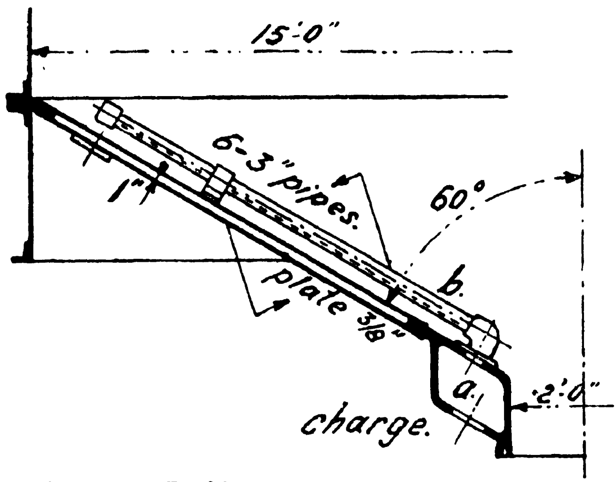


Fig. 461.—Double Bottom and Charging Pipes.

of copper. Insulation should be provided on the outside, as otherwise heavy heat losses will occur. The syrup is charged into a ring-shaped space *a*, from which six perforated pipes give an even distribution over the pan bottom.

3.—Calandria Vacuum Pans.

This type of pan is built on lines similar to those of the vertical evaporating bodies, the belts being of cast iron or steel, as is the case with coil pans. A very large heating surface can be arranged in a comparatively small space and a short strike thus achieved. Some operators are afraid of overheating in calandria pans, but this risk is groundless when exhaust steam is used and it should be more dreaded with coil pans, heated with live steam. A *Straight Fixed Calandria Pan* is shown in *Fig.* 462, this being a favourite design; with a first boiling of about 85 purity the strike will be finished in from 2½ to 3½ hours, exhaust steam being the heating medium. The circulation goes down through a centre well of 4 ft. 2 in. diameter and the six 11 in. downtakes, distributed between the tubes. These last are of copper, as generally is the case, and the tube plates are of "ingot iron" in this instance. Brass or copper tube plates are also frequently employed; a flush upper tube plate will make cleaning easy and ensure less accumulation of caked sugar.

After the pan has been discharged, hot water or steam, or both, are sprayed into the pan to dissolve and dislodge any massecuite adhering to the inside of the pan. The discharge valve is sometimes closed for this operation and as the vapour valve is so, of course, a pressure may develop inside the pan. Cast iron vacuum pans in particular are not designed for inside pressure and therefore a large spring-loaded safety valve should be provided at the top of the pan, which will discharge under a small overload. The author knows a case of a vacuum pan exploding, which resulted in some casualties and great damage to the rest of the equipment around.

In the bottom of the pan of *Fig.* 462 a copper coil is arranged, which can be used also for boiling out the pan with caustic soda or muriatic acid solution. A special spray pipe for cleaning purposes will be useful below the calandria, at the highest spot of the bottom cone.

As to the *circulation of the massecuite,* little is known, although it has been recognized as of paramount importance for the pan performance proper. WEBBE has made extensive investigations[1] by measuring temperatures inside the vacuum pan and by plain thermic calculation he has found a speed of circulation varying between 0·012 and 2·97 ft. per sec., the average being 0·387 ft./sec. for heavy concentration, as the evaporation rate, at the moment of the tests, was only 1·5 lbs. per sq. ft. H.S /hr.

The calandria pan on which the tests were made had an inside diameter of 12 ft., 1200 cub. ft. capacity, 2100 sq. ft. H.S., 418 tubes 5 in. outside dia. × 4 ft. 0 in. long, and a central downtake of 36 in. dia. The upgoing current through the tubes gave a free area of about 7600 sq. in., whereas the downgoing current through the centre well encountered only a free area of about 1017 sq in. It will be apparent that the downgoing current had to be seven times as fast as the upgoing, which would greatly diminish the efficiency of the circulation.

Several operating sugar men have pointed out to the author this defect in many calandria pans, and he has changed the calandrias of several pans, providing for a much larger downtake; for a 15 ft. pan this downtake was increased to 6 ft. with very satisfactory results, as it shortened the time required for the striking of low grades, undoubtedly due to the better massecuite circulation.

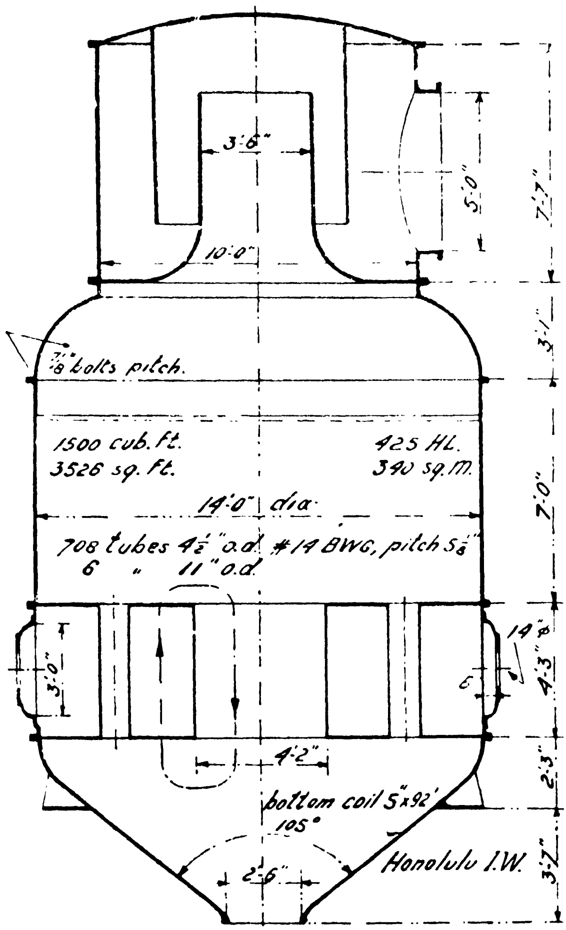
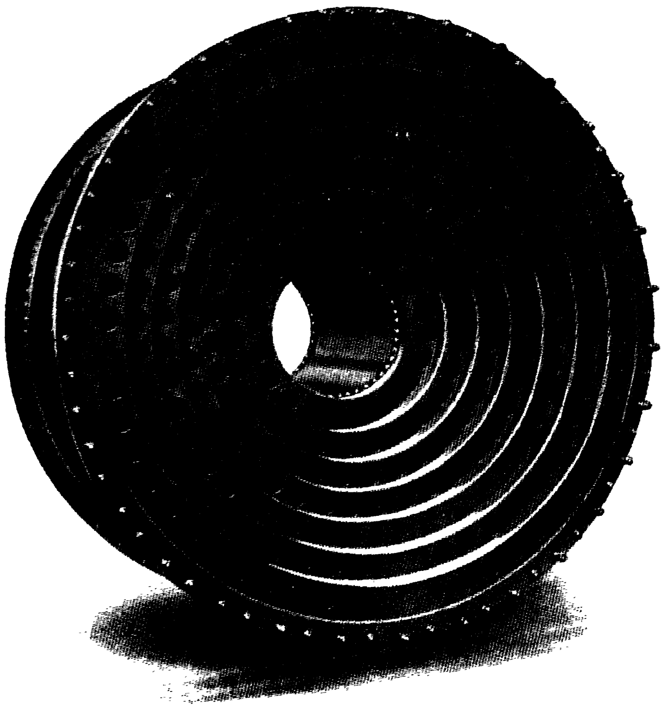
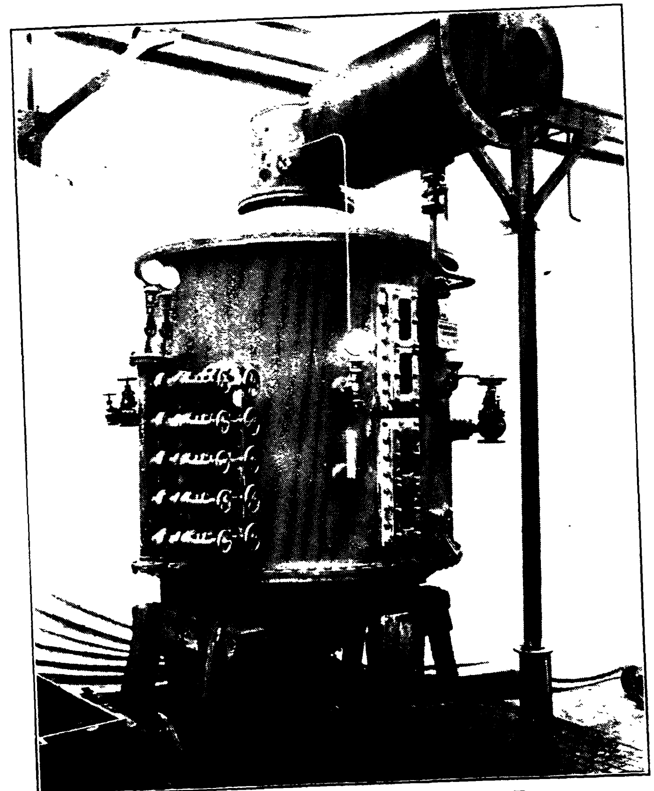


Fig. 462.—Straight Fixed Calandria Vacuum Pan.

[1] See the article of A. L. WEBBE in "Proceedings, Association Cuban Sugar Technologists," 1932.

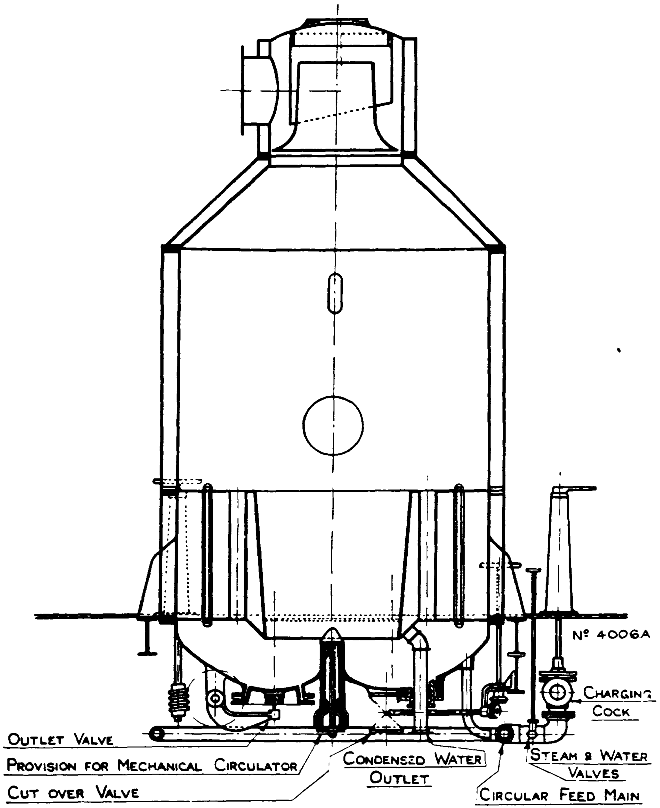


Stepped Top Tube Plate for Calandria Vacuum Pan.
(*Fawcett, Preston & Co., Ltd.*)



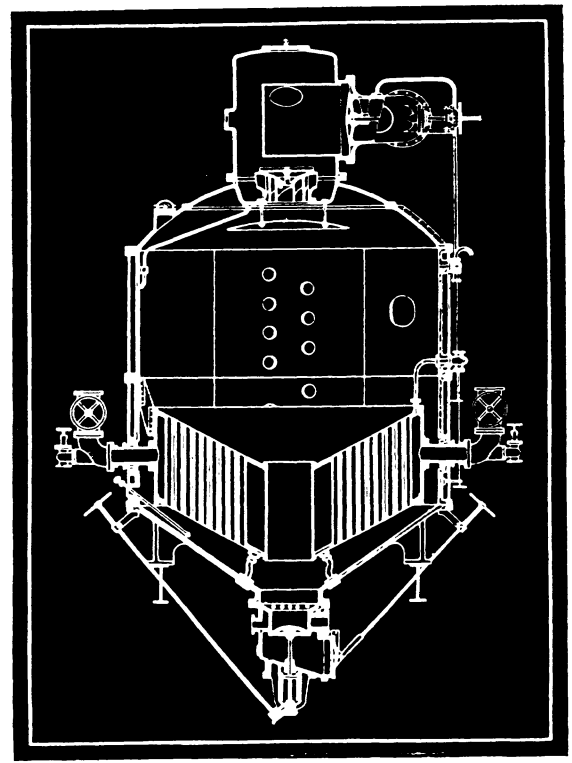
Small Cast Iron Coil Vacuum Pan.
(*Maschinenfabrik Sangerhausen*)

PLATES 93 & 94.



Sectional View of "Centre Flow" Type Vacuum Pan.

(*Geo. Fletcher & Co., Ltd.*)



Cross Section of Vacuum Pan with Inclined Calandria.

(*U.C.M.A.S.*)

The newest designs of calandria pans embody a large downtake, as may be seen from *Fig.* 463, which depicts a *Stream Flow Calandria Vacuum Pan*, recently furnished to the British colonies. The centre well is conical, having 6 to 7 ft. diameter, so the massecuite will easily flow down towards a streamlined bottom of unique design, which guides the massecuite back to the heating surface again.[1] The design allows for the installation of a mechanical circulator, if such an apparatus be required for very viscous massecuites of low purity. The bottom proper has an inclination of 20°, which will assist the pan to discharge easily. The calandria has three condensate drains *C.w.* and the charging syrup and molasses are well distributed, by six inlets, into the flow of the massecuite.

The incondensible gases are released by three perforated pipes inside the calandria, connected by a main line (outside the pan) with the vapour belt.

An 18 in. manhole will give easy access to the interior of the pan. The footing or graining capacity, just covering the calandria, is only 28 per cent. of the total pan capacity, which figure may be considered a favourable percentage.

In this same pan the proportion between the free tube area of the calandria and the area of the downtake amounts to about 2·1 : 1, thus showing a considerable improvement as compared with the previously mentioned calandria pan.

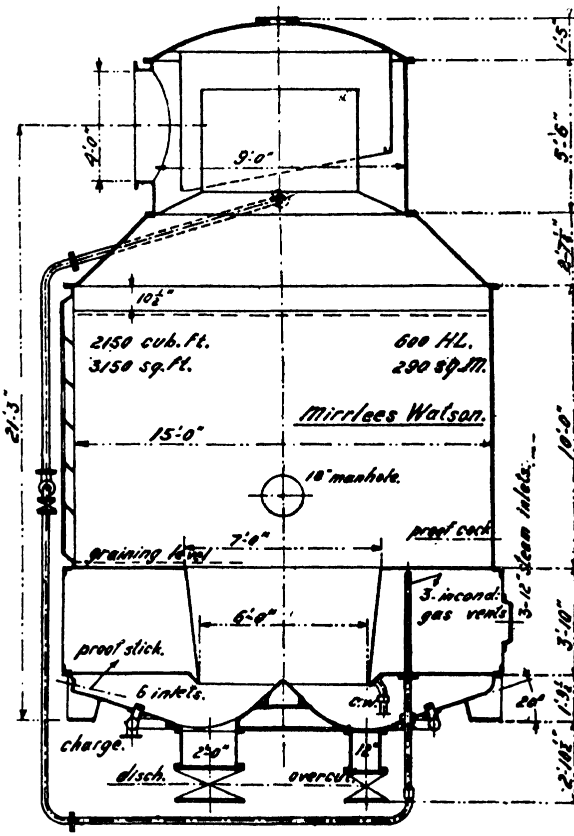


Fig. 463.—Stream Flow Calandria Vacuum Pan.

The *floating type of calandria* has been favoured in the past by many designers, although it is now disappearing. Its operating performance generally has been very good, as the downtake area was increased and thus an efficient circulation was established. The heating surface generally is less, due to the smaller calandria diameter, which cannot be offset by the heated calandria evolute. The connexions between the calandria and the pan belts have proved a source of trouble through leaking, and these connexions have to be made in such a way that free expansion can take place.

[1] See Netherlands Patent, No. 24537, September, 1929.

To reduce the bottom space in a calandria vacuum pan, the lower plate of the calandria has been made conical or spherical, but this has the inconvenience of greatly varying the tube length, and so has led to a *Pan with Conical Calandria*, of which a recent design is shown in *Fig.* 464. The syrup charging pipe has been put in its proper place at the bottom of the pan. It will be apparent that a more or less stream-line effect will be obtained with this construction.

Double conical calandrias have been used in American designs, but have not been generally adopted. The advantages are two-fold :—

(1) Reduced graining capacity of about 25 per cent. of the pan volume.

(2) Reduction of the hydrostatic head of the upper calandria, thus ensuring increased heat transmission.

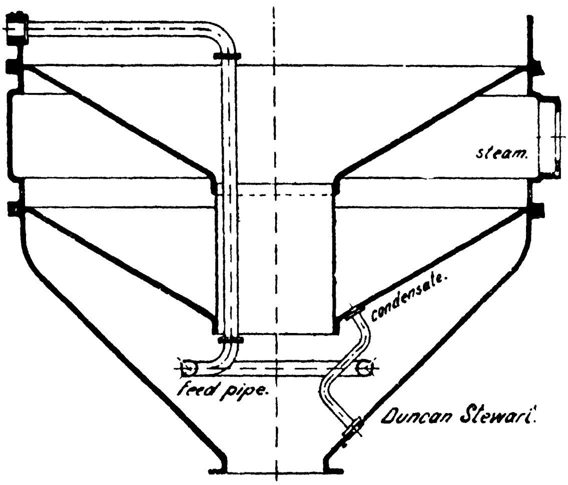


Fig. 464.—Pan with Conical Calandria.

The tubes in these conical calandrias are sometimes arranged in a vertical position or otherwise perpendicularly to the tube plates, thus forming an angle of about 60° with the horizontal. For manufacturing reasons, the latter design is more convenient, as the plates have to be drilled perpendicularly to the surface, the tracing of the hole centres being different for lower and upper plates, as well as the circular pitch. Moreover, the task of expanding the tubes will be easy, as equal adhesion is to be expected all round the tube. As regards circulation of the massecuite in the pan, the author prefers vertical tubes, as this circulation is difficult at high concentration, owing to the increased viscosity and it should not be hampered wherever possible.

To have the tubes vertically arranged, without the inconvenience of inclined drilling or unequal tube expanding, *Stepped Conical Tube Plates* have come into vogue, generally being cast in brass. They can be stamped in copper or steel, but this procedure is very expensive on account of the costly dies.

The radial tube pitch for stepped tube plates apparently amounts to :—

$$p_{rad} = d + t + \tfrac{1}{8}\text{ in.} \qquad (119)$$

in which : p_{rad} = minimum radial pitch of tubes in inches.

d = diameter of holes in tube plates, generally equal to the outside tube diameter plus $\frac{1}{32}$ in.

t = vertical thickness of tube plate between the tubes.

When cast tube plates are used, different thicknesses can be applied for the vertical and the horizontal parts of the plate. The addition of $\frac{1}{8}$ in. serves to keep clear of the steps while drilling.

The sight glasses of a vacuum pan have to be cleaned regularly, especially after a strike has been dropped. For circular glasses a *Sight Glass Cleaning Spray* using hot water or low pressure steam is shown in *Fig.* 465, as designed by the author. The nozzle *ab* inside the pan is made of brass, and joined by ½ in. pipe connexions *c* to the outside, where a double branch for each spray is provided, each having a stop valve for hot water and low pressure steam. Exhaust steam of 5 to 10 lbs. gauge pressure is sufficient for the purpose, when the hot water will not remove the adhering crystals. The apparatus have worked entirely satisfactorily.

The discharge valve of a vacuum pan is variously designed to correspond with the following types :—

(*a*) Open slide valve.
(*b*) Toggle inclined disc valve.
(*c*) Angle valve with closed body.
(*d*) Straightway gate or sluice valve.

The first type (*a*) is well known and has found extensive application, as its working can be readily inspected. The valve disc is seated against a brass, copper or rubber joint and the disc has first to come clear of the seat before it can be withdrawn. The closing device, therefore, consists of two actions, the lifting one and the sliding, which usually are both accomplished by turning a single hand-wheel.

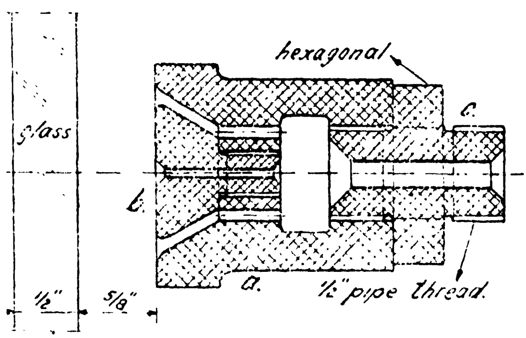


Fig. 465.—Sight Glass Cleaning Spray.

The inconvenience of the *sliding discharge valve* of horizontal arrangement is the fan-like discharge of massecuite, when the valve is opened, and a good re-collecting chute or trough with a sloping bottom of at least 9° and vertical walls, has to be fitted to prevent the spilling of the massecuite. The guides and the spindle are generally coated with massecuite during the discharge, and this may incrust on these fittings, growing to a mass which has to be removed by hand from time to time, so as not to impair the working of the valve.

The *Inclined Toggle Discharge Valve* with a 45° elbow piece, as shown in *Fig.* 466, effects a better stream flow of the massecuite. The disc is actuated by a toggle gear, composed of the two rods *d* and the lever *k*, which press the disc *a* on the pan outlet. The disc is hinged by a double lever *b* around the pivot *c*. The thread spindle *f* is stopped in its travel by the arrester *h* inside the tube *g*, and in this position the centre lines of *d* and *k* are not yet in line.

The handwheel *j* is for operation from below the pan platform. The tube *g* is well supported in a pivoted bearing attached to the pan body above the platform. This bearing has to withstand the thrust caused in closing the toggle gear. A second handwheel, operated from above the pan platform, is provided on the tube *g*. The closing adjustment of the disc *a* is easily effected by tightening the nuts *e* on the rods *d* and thus a positive closing is assured in a unique manner.

The sealing of the disc is accomplished by means of an inserted rubber ring, which gives a tight joint, although the rubber will wear, especially when all sugar crystals are not washed off the disc, after a strike has been dropped and the valve is closed again.

A discharge valve, fortunately, need not be absolutely air-tight, as the massecuite on top of it will act as a good seal. A discharge valve which proves leaky (within reasonable limits of course) when the pan is empty, may become tight when the pan is charged.

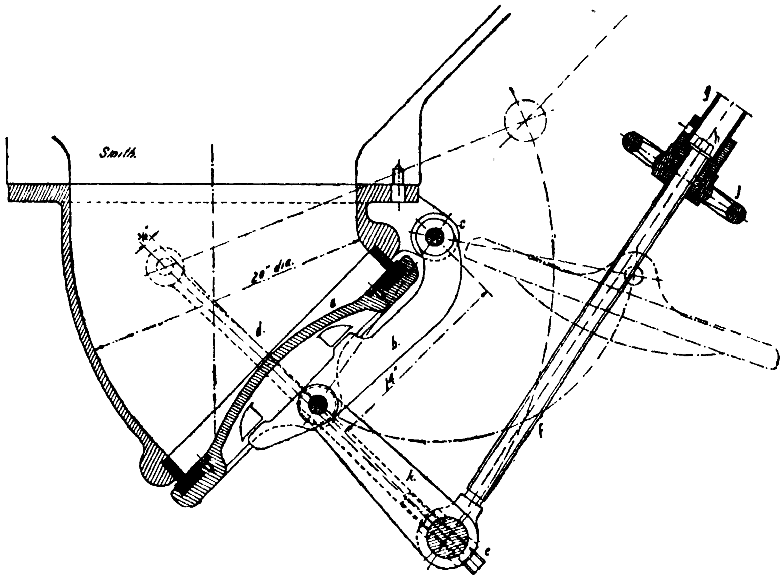


Fig. 466.—Inclined Toggle Discharge Valve.

With the pan under vacuum, an outside pressure exists on the valve, amounting to (in lbs. per sq. in.) :—

$$p_{ext.} = 14{\cdot}7 - p_{vac.} - (h \times 0{\cdot}648) \quad \ldots\ldots\ldots\ldots\ldots \quad (120)$$

being : $p_{ext.}$ = Outside pressure on disc in lbs./sq. in.
14·7 = The normal atmospheric pressure at sea level in lbs/.sq. in.
$p_{vac.}$ = The absolute pressure inside the pan in lbs./sq. in.
h = The head of massecuite above the discharge valve in feet.
0·648 = The hydrostatic pressure of a massecuite column, 1 ft. high, having a specific gravity of 1·50.

At 27 in. vacuum and 12 ft. hydrostatic head of massecuite, the outside pressure amounts to :—

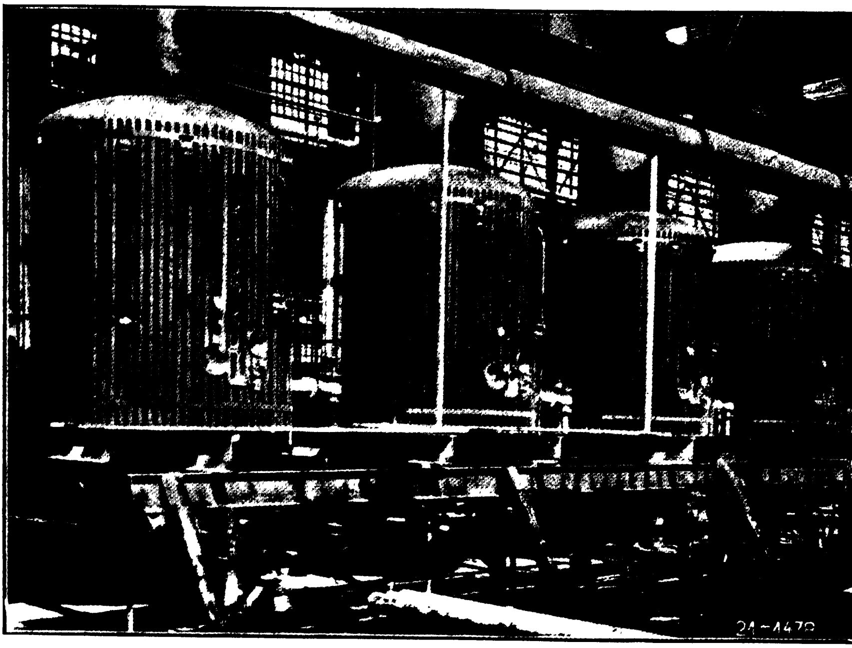
$$14{\cdot}7 - 1{\cdot}465 - (12 \times 0{\cdot}648) = 5{\cdot}459 \text{ lbs./sq. in.}$$

When the vacuum is broken, the inside pressure on the discharge valve amounts to :—

$$12 \times 0{\cdot}648 = 7{\cdot}776 \text{ lbs./sq. in.}$$

For different vacua, as read on the vacuum gauge and 29·8 in. absolute vacuum, the absolute pressures are :—

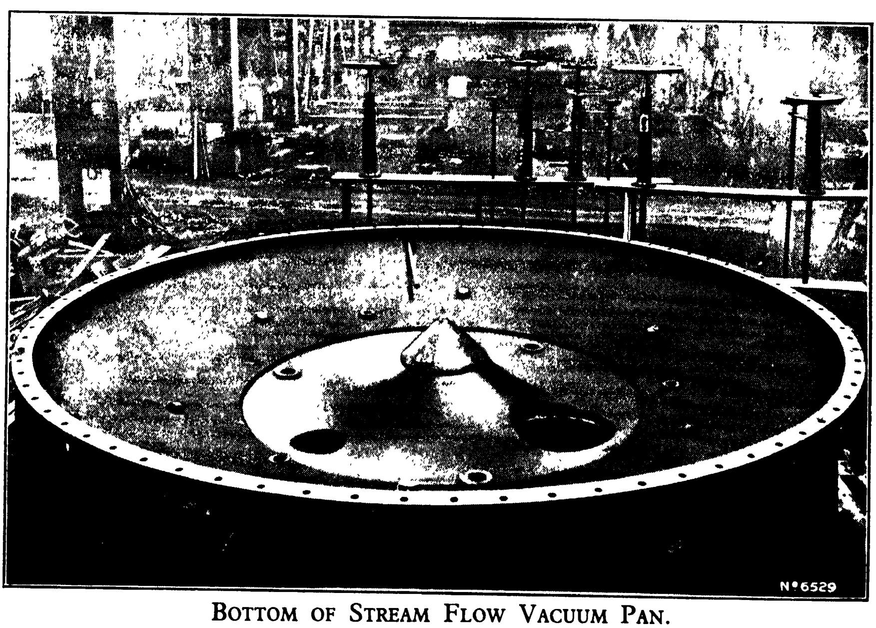
$p_{vac.}$ at	24 in. vacuum	2·93 lbs./sq. in.
,,	25 in. vacuum	2·44 ,,
,,	26 in. vacuum	1·954 ,,
,,	27 in. vacuum	1·465 ,,
,,	28 in. vacuum	0·977 ,,



FOUR CALANDRIA VACUUM PANS.

Vapour Valves are not yet attached.

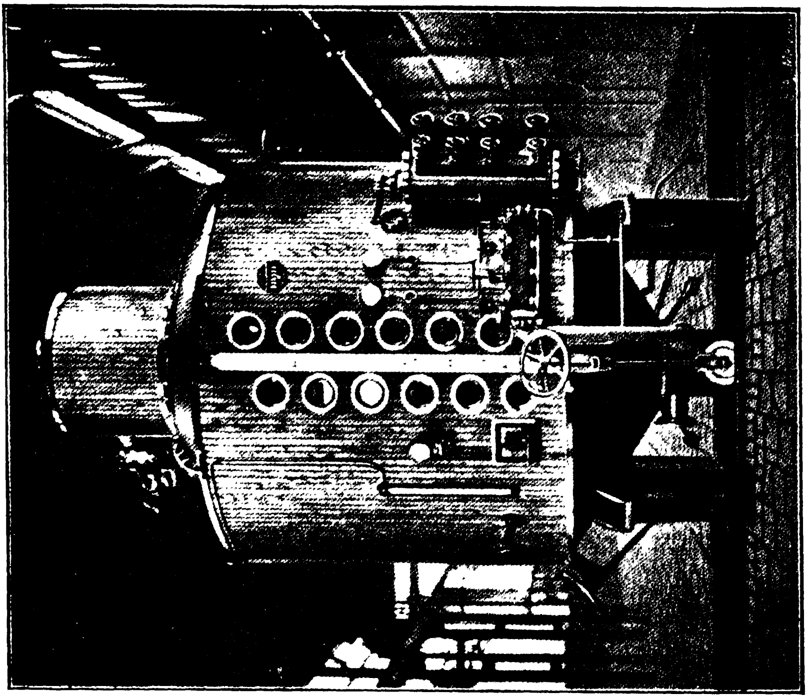
(*Skoda Works, Ltd.*)



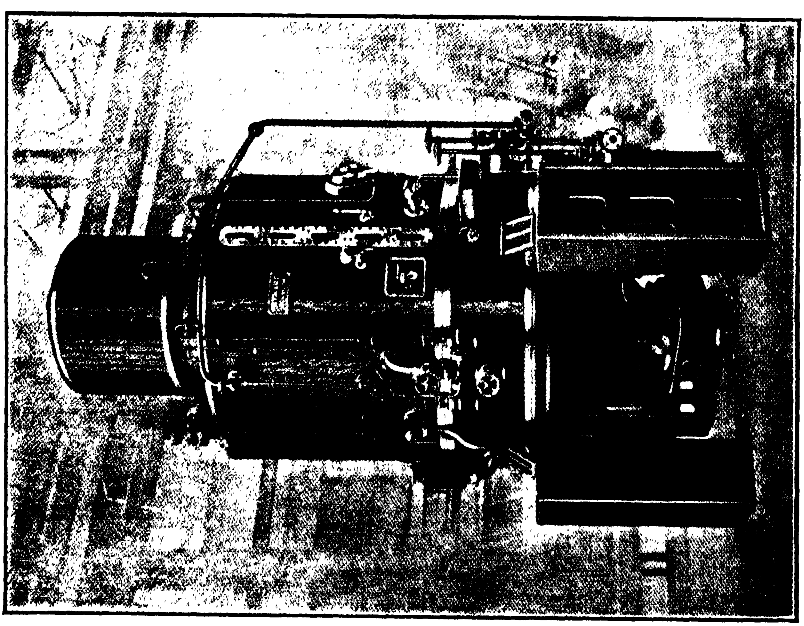
BOTTOM OF STREAM FLOW VACUUM PAN.

(*The Mirrlees Watson Co., Ltd.*)

PLATES 97 & 98.



COIL VACUUM PAN.
(U.C.M.A.S.)



CALANDRIA VACUUM PAN WITH ANGLE DISCHARGE VALVE.
(Duncan Stewart & Co. Ltd.)

To assume for the disc and operating gear a pressure of 15 lbs./sq. in., inside as well as outside, will give safe results.

A design of *Angle Discharge Valve* is shown in *Fig.* 467, the disc *a* being seated against a rubber or soft copper joint. The valve is operated by a brass screw spindle by means of the bevel gear *b*, which can be manipulated from any convenient place under or on top of the pan platform. The discharge slopes at about 11°, which is sufficient for easy flow of the massecuite. Gutters for massecuite should have a minimum inclination of 9° for proper effect.

The enclosed valve as shown has the advantage that there will be no spilling of massecuite and a closed pipe line, having the above-mentioned slope, and provided with a good steaming-out arrangement, can be used for conveying the massecuite to the mixers or crystallizers. The two steam connexions at *c* are for thorough cleansing of the valve and body after a strike has been discharged.

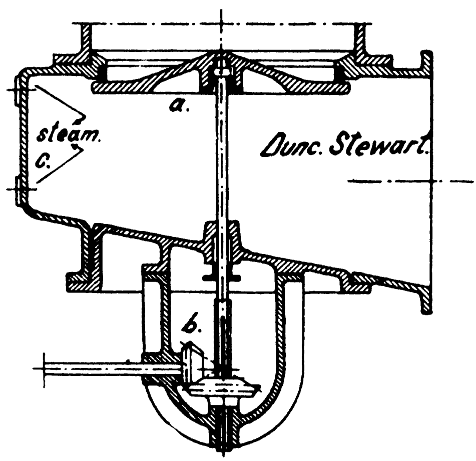


Fig. 467.—Enclosed Angle Discharge Valve.

The *Sluice or Gate Valve* is also used for this purpose, a T-piece being generally mounted under the pan outlet and thus two branches can be connected. It is essential that the dead space where there is no circulation of massecuite should be kept as small as possible, as massecuite hardening at these points is not illusory and may upset the quick discharge of a pan. The sluice valve proper needs a steaming-out nozzle at the bottom of the valve casing; this prevents accumulated massecuite being the cause for the valve not closing well.

4.—Vacuum Pans with Mechanical Circulation.

As already mentioned, the circulation of the massecuite in a vacuum pan is of paramount importance for good heat transmission and even growth of the crystals, as well as for high concentration and the corresponding high yield in crystals. Long ago, several designers, among whom FREITAG and GROSSE are well known, provided vertical stirrers or screw conveyors in their vacuum pans, which have been and still are used in the beet sugar industry. In cane sugar factories, these devices, not being fully perfected as a rule, have been discarded by sugar boilers for lack of desired results.

As the downtake area, especially in our present-day calandria pans, has been sacrificed to a great extent to provide a larger heating surface, the stage has apparently been reached where, with increased heating surface, the time to complete a strike cannot be shortened for lack of circulation, thus giving rise to overheating, so-called "froth fermentation." Attention, therefore, has been called to the matter by several experts,who have studied the phenomenon, and this has resulted in the rehabilitation of mechanical pan circulation.

In *Fig.* 468 is shown a *Calandria Pan with Mechanical Circulation,* which has been applied in the beet as well as the cane sugar industry in U.S.A. and Latin America. The circulator proper is composed of a shaft at the vertical pan centre, which is driven through a variable speed gear by an electric motor, mounted on top of the pan dome. There are four impellers *a* of the screw or helicoidal type with interjacent baffles or rectifiers *b*, to stop the massecuite whirling. At *c* the lower shaft bearing is provided with three wearing blocks, which can easily be replaced. The upper three propellers have no casings, whereas the lower one is arranged in the upper part of the downtake.

The syrup and molasses are fed through a pipe *d* inside the downtake underneath the lowest impeller and the circulation is indicated by the arrow *h*. The calandria is drained by the condensate pipes *g* in the usual way.

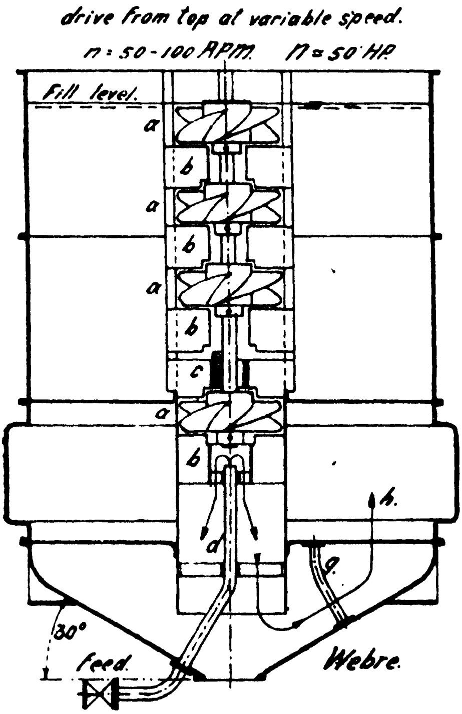


Fig. 468.—Calandria Pan with Mechanical Circulator.

It will be obvious that any higher speed of the circulator will produce a higher rate of circulation; the power consumption increases nearly as the square of the speed according to the slip. This consumption is high, as the massecuite is very viscous at high concentration, when circulation is mostly needed, and slip may occur, reducing the overall efficiency.

As previously stated, the highest evaporation takes place when the natural circulation in the pan is at its highest rate, and the purpose of the circulator is thus to retain this highest rate during the whole strike. Let it be assumed that the impellers have 30 in. pitch, the slip being taken at an average of 25 per cent. and a speed of circulation of 2 ft. per second is desired, then the speed of rotation has to be :—

$$2 \times 12 \times 60 : (30 \times 0{\cdot}75) = 64 \text{ r.p.m.}$$

In a 15 ft. calandria pan with a large downtake and 6 ft. hydrostatic head of massecuite, a circulator as shown in *Fig.* 468 has been fitted, driven by a 100 h.p. motor, having 100 r.p.m. max. speed of the main shaft, but normally run at 50 r.p.m. under reduced power. The low grade strikes in this pan were formerly finished in about 8 hours, but after installing the circulator the boiling time was reduced to about 4½ hours, the Brix of the massecuite at the moment of discharge being raised from an average of 93 to well over 95.

An ammeter in one of the phases of the three-phase alternating current motor will indicate the fluctuation of the power output and hence also the degree of fluidity.

Another design of circulator is shown in *Fig.* 469 ; this has been developed in Hawaii with very good results. It will be apparent that the design of the pan is a stream flow one, and the propeller *c* of a marine type with three blades has been arranged at the most suitable spot in the pan, the driving taking place from underneath through an electric motor *d* having constant power output at different speeds of 600, 900, 1200 and 1800 r.p.m., the reduction gear stepping down in the proportion 1 : 15.

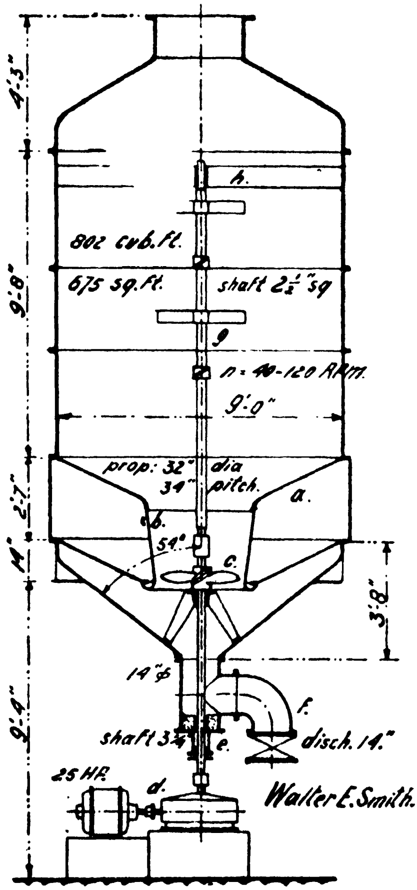


Fig. 469.
Single Propeller Circulator Design.

From extensive tests with this pan,[1] the following results can be attributed to the use of this circulator, at a speed in practice of about 4 ft./sec. for first boilings and about 3 ft./sec. for low grades, allowing about 25 per cent. slip for the impeller. These results are :—

(*a*) An average of 25 per cent. increase in heat transmission for first boilings.

(*b*) An average of 50 per cent. increase in heat transmission for low grade massecuites.

It is clear that low grade massecuites with a higher viscosity require much more mechanical circulation than do the freer boiling first massecuites.

The average power input at different speeds of the circulator shaft has been recorded as follows :—

40	60	80	120 r.p.m.
7·8	11·9	17·3	33·3 h.p.

The power consumption therefore is not proportional to the speed, nor to the square of it, as theoretically should be the case, if there were no slip or friction.

Under frictionless and slipless conditions, the weight W_m of massecuite circulated per second amounts to :—

$$W_m = \frac{\pi}{4}(d^2 - d_i^2) \times V \times W$$

in which : d = outside diameter of impeller in feet.
d_i = diameter of hub in feet.
V = speed of massecuite in ft./sec.
W = weight of massecuite per cub. ft. (93·6 lbs.).

or : $W_m = 73{\cdot}5\ (d^2 - d_i^2) \times V$ in lbs./sec.

$$\text{now} \quad V = \frac{p \times n}{60}$$

in which : p = the propeller pitch in feet.
n = the number of revolutions of the impeller per minute.

[1] See report of WALTER E. SMITH, of Raw Sugar Tech. Committee, Hawaiian Sug. Plant. Ass., 1932.

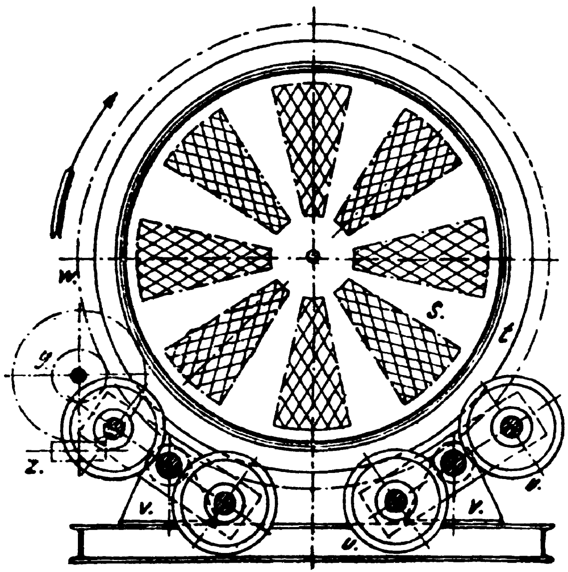


Fig. 470.—Rotary Horizontal Vacuum Pan
(Cross Section)

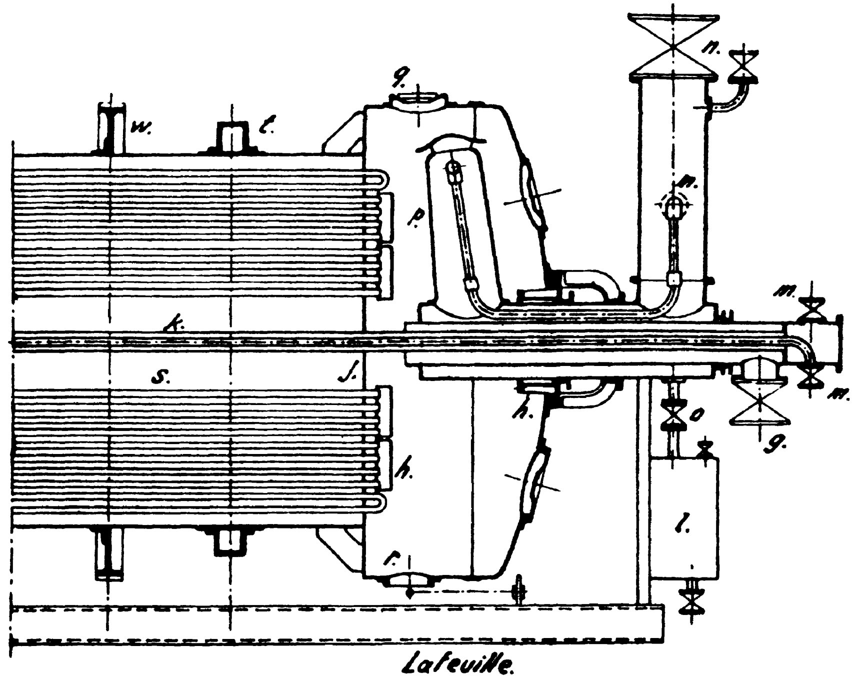


Fig. 471.—Rotary Horizontal Vacuum Pan
(Longitudinal Section).

For respectively 60 and 120 r.p.m., V amounts to p and $2p$ feet per second, and taking the latter, the formula can be written :—

$$W_m = 73{\cdot}5\ (d^2 - d_i^2) \times 2p.$$

The power consumption is the product of weight and speed ; it being considered that 1 h.p. = 550 ft./lbs./sec., then :—

$$N = \frac{W_m \times V}{550} = \frac{73{\cdot}5\ (d^2 - d_i^2) \times 4p^2}{550} = 0{\cdot}534\ (d^2 - d_i^2) \times p^2 \ \ldots . \ (121)$$

In the case above mentioned, the ideal power input therefore amounts to :—

$$N = 0{\cdot}534\ (2{\cdot}67^2 - 0{\cdot}75^2) \times 2{\cdot}83^2 = 28 \text{ h.p.}$$

It has now been found that this power consumption varies between 27 and 48 h.p., the overall efficiency varying between 1 and 0·58. For 40 r.p.m. the efficiency varies between 0·46 and 0·32.

Apparently the highest efficiencies are obtained at high speed, but the fact should not be overlooked that a natural circulation prevails in the pan, and therefore the overall efficiency of the impeller is still lower. As there will be less slip at low speed, the true friction resistance may be assumed to correspond with the power input, whereas at high speeds the slip is greater, requiring less power for the propulsion of a decreased quantity of massecuite, but under increased friction.

Quite different from the standard vertical pan type is the *Rotary Horizontal Vacuum Pan*, of unique design, shown in *Figs.* 470 and 471. It had originally been invented as a crystallizer in a beet sugar factory in France, but soon it was found that a larger scope of application was feasible and now the apparatus has combined both boiling and cooling operations. In several cane sugar countries, installations of these rotary pans have been made during the last decade, and a reduction in space occupied in the factory is a principal result.

The whole drum is rotated and therefore only small circulation velocities of the massecuite are required for good heat transmission. There is only a very small hydrostatic head and the gravity force is the one causing the circulation.

The construction comprises a cylindrical drum of about 7 ft. diameter and 30 ft. length for the larger sizes, supported by two tyres *t*, which bear upon two brackets *v* on each side, these having two supports each with two rollers mounted in roller bearings. It is the common construction as used for sugar dryers, rotary kilns and kindred equipment. The driving takes place by means of two toothed rims *w*, through spur gears *y*, which are mounted on a common transmission shaft, driven by the worm drive *z*.

The fixed connexions on the revolving drum can only be attached at the centre, and in *Fig.* 471 is shown the vacuum head of the rotary pan. The other end of the drum has attached to it the connexions for steam and cooling water, as well as the exit for incondensible gases. The vacuum head has a larger diameter than the rest of the drum, to hold the fixed vacuum pipe *p*, which is of stream-lined cross-section and covered by a cap. The resistance to the massecuite during rotation of the drum is therefore kept within reasonable limits and drippings cannot fall inside this pipe.

The feeding of the rotary pan is accomplished by the gate valve *g* acting as grain or massecuite inlet ; the three connexions *m* are for syrup and molasses which have interior conduits to above the massecuite level or by means of the interjacent tubes *k* to midway and the steam head end of the drum. As there is little or no lengthwise circulation in the drum, any charge of a more diluted feed, like syrup and molasses, has to be distributed over the full length of the drum.

The vacuum valve *n* has a bye-pass for equalizing the pressure on both sides of the valve; this will make the operation easier and reduce wear. At *o* a cleaning-out valve is provided, which will relieve the vacuum pipe from any massecuite or molasses that may have entered it. The closed vessel *l* is arranged to make this cleansing feasible while under vacuum.

The contents of the rotary pan are emptied by a number of discharge gates, all in the same longitudinal centre line of the drum, one being shown at *r*. At *q* a safety valve is provided, in case any pressure should develop in the pan during the steaming-out operation.

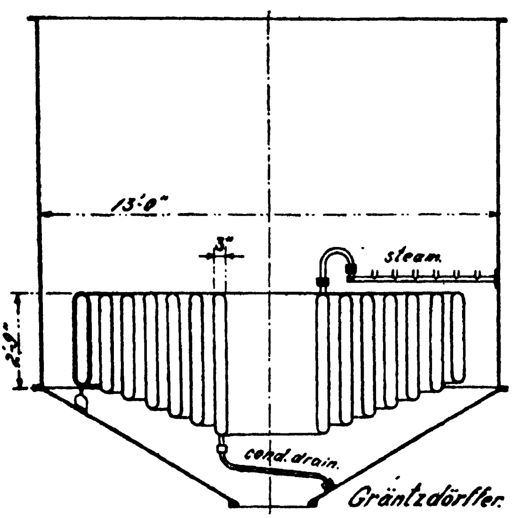


Fig. 472.—Ring-shaped Hollow Heating Bodies.

The tubes are arranged in eight nests *s*, the size of tube being about 2 in. ext. diameter. It will be seen that free passages are left between the tube nests for free circulation of the massecuite. The tubes are expanded in headers *h*, to allow for free expansion or contraction. The plates *j* are only for tube support and have openings for passage of the massecuite in longitudinal direction.

From the operation of these rotary pans it has been learnt that they will concentrate at a very high Brix, up to 98 and over; the rotation speed of the drum varies, it being about 1 r.p.m. when boiling and ½ r.p.m. during the cooling performance. A common merit of the boiling equipment with mechanical circulation is that vapours of 3 to 5 lbs./sq. in. can be used to advantage as a heating medium, which fact may favourably influence the heat balance of the factory.

The samples of massecuite are taken from a plug cock, which revolves slowly with the drum. The height of the massecuite is noted from a measuring stick, when the level can be observed through one of the two sight glasses in the vacuum end.

The power consumed in rotating the pan is small; for a 1760 cub. ft. pan only a 10 h.p. motor need be provided, the actual power input being below 2 h.p. During the cooling the massecuite may "freeze," in which case about three times the average power is consumed. At this stage a previous run-off is charged into the pan for dilution. The discharge of the pan is easy, and for washing out this run-off is also used. The pan fill is about 0·7 of the gross capacity.

The total boiling and cooling cycle takes about 6 hours for high grades and 10 to 16 hours for low grades.

These rotary pans are built in three sizes of 1166, 1533 and 1766 cub. ft. total capacity ; the diameter for all three sizes is equal, only the length varying.

There are a few patents, dealing with production of sugar direct from the syrup by spraying it under high pressure of about 3000 lbs./sq. in. in a stream of hot air, or by rapid cooling of the sprayed mass, so as to cause supersaturation. The author does not know of any practical data covering these systems.

Finally, a heating system by means of *Ring-shaped Hollow Heating Bodies* has been applied for vacuum pans, as shown in *Fig.* 472, and has been put to use in the beet sugar industry in continental Europe.

A very good circulation and heat transmission have been obtained and a large heating surface can be arranged with a low footing level in the pan. The steam connexions, as well as the condensate drains, have to be designed so that free expansion or contraction of the heating bodies is rendered possible. Stiff connexions are liable to break and careful attention has to be given to this detail, so as to obtain reliable results. Thanks to the flush exterior, the heating surface will remain clean.

CHAPTER XXIII.

THERMO-COMPRESSORS.

In the design of a modern sugar factory, it is desirable to have a shortage, a pronounced shortage, of exhaust steam available, so that it may be possible to install in the future more live steam consumers, such as are involved in adding to the equipment shredders, revolving knives or the like. The prime-movers for these apparatus consume live steam and thus produce exhaust steam. If at the outset there is already an excess of exhaust steam it is obvious that the heat balance gets upset and may give rise to additional fuel consumption.

Any pronounced shortage of exhaust steam thus leads to a better steam balance and will avoid in many instances the consumption of additional fuel.

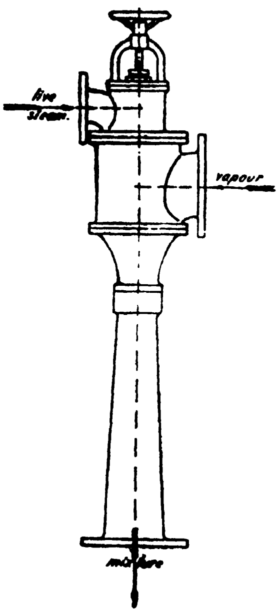


Fig. 473.
Thermo-Compressor.

In normally designed factories, the boiling-house will consume all the exhaust steam produced, and about 10 to 20 per cent. of the live steam, generally reduced by means of a reduction valve to a lower pressure than that prevailing in the boilers. This expansion of the live steam for the boiling-house can nevertheless be achieved with delivery of mechanical energy by compressing low pressure vapours from a first evaporating body or vapour cell, so that they will attain a higher pressure and an equivalent higher temperature.

The apparatus for compressing these low pressure vapours is termed a *Thermo-Compressor*, and its general appearance is shown in *Fig.* 473. Invented about 60 years ago, it has been applied in the sugar industry some 25 years. The live steam enters at the top inlet and it can be regulated or shut off by a hand-operated valve. The low pressure vapours are drawn in at the right hand orifice, whereas the compressed mixture is discharged at the bottom. The thermo-compressor can be arranged in any desired position, horizontal, vertical, inclined or reversed.

The steam passes a nozzle, converting its pressure or potential energy into a kinetic one and through surface friction or adhesion of this steam jet, the low pressure vapours, surrounding the jet, are drawn along and carried into the receiving nozzle for discharge.

It will be obvious that such a steam jet, especially when of larger size, might be conveniently split into several of smaller diameter, as the contact surface of the jet will thus increase with a corresponding increase in the efficiency of the thermo-compressor. Instead of one large thermo-compressor, therefore, several of smaller size will be more efficient in certain instances. There are also thermo-compressors having several small size nozzles built into one housing, and this multi-jet system will prove more economical than several separate compressor units. In every case a careful study has to be made of what is required.

The thermo-compressor is inexpensive, does not need any lubrication, and there is no wear on the nozzles when pure steam is used. Its light weight makes its installation easy and no foundations are required, as it may be supported from the connecting pipe lines.

The nozzles of the thermo-compressor are designed for certain live steam and vapour pressures, and any variation will lead to reduced efficiency. But as this phenomenon of reduced nozzle efficiency also arises to some extent with the nozzles of a steam turbine, the disadvantage accruing is of no greater importance.

In those cases where a big variation in capacity is required, several smaller sized thermo-compressors should be used, so that some can be shut off as may be desired.

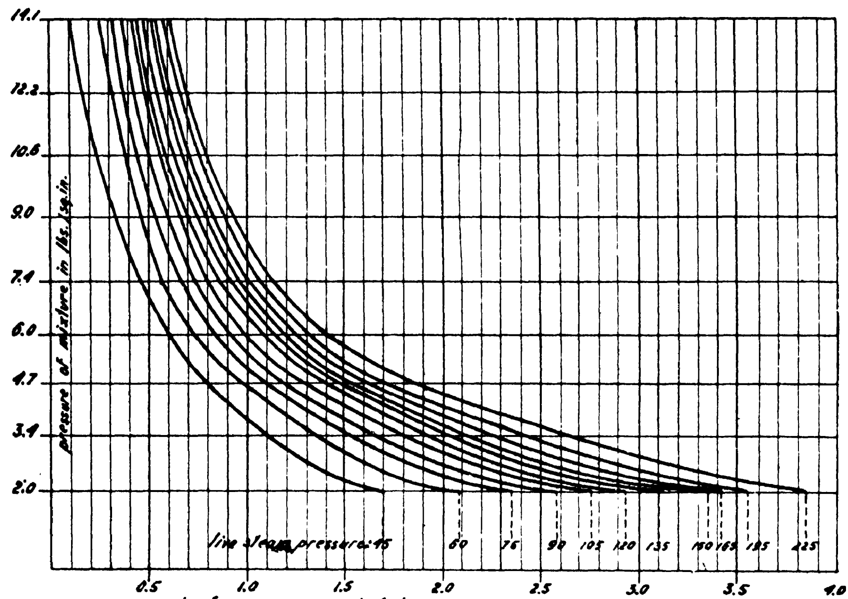


Fig. 474.—Thermo-Compressor Diagram.

In *Fig.* 474 is shown a diagram, according to data compiled by ERCLANCHER in 1911. The curves indicate the amount of vapour compressed per pound of live steam and from the atmospheric pressure to the gauge pressures mentioned at the left side of the diagram.

One lb. of live steam, e.g. of 105 lbs. gauge pressure, will compress 1 lb. of vapour of atmospheric pressure up to 6 lbs./sq. in. back pressure. By checking the amounts of B.Th.U.'s which enter the thermo-compressor, it will be ascertained that the discharge mixture becomes slightly superheated.

Although the diagram is for practical use, it nevertheless should be remembered that under optimum conditions and with well-designed thermo-compressors, higher efficiencies still may be reached.

For the conditions normally prevailing in a sugar factory, one lb. of live steam will compress about one lb. of vapour, which ratio will be useful for overall calculations.

Fig. 475 gives the existing arrangement, as installed in a cane sugar factory and tested by the author. The installation comprises a single evaporator

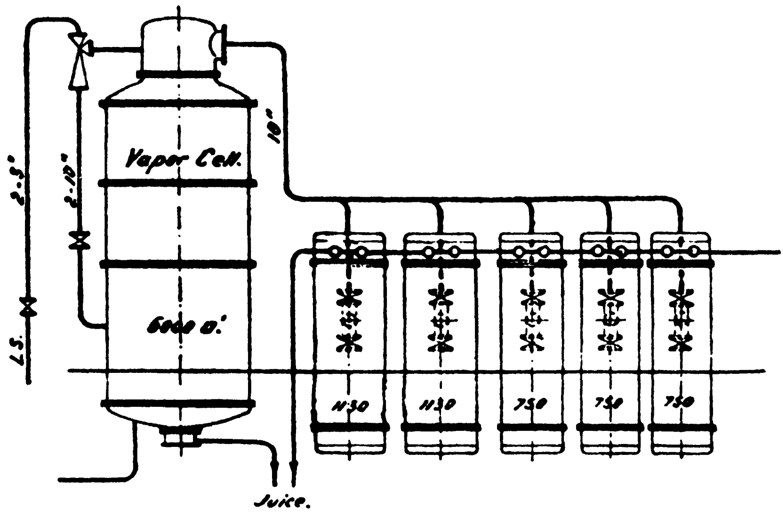


Fig. 475.—Arrangement of Thermo-Compressors on Vapour Cell.

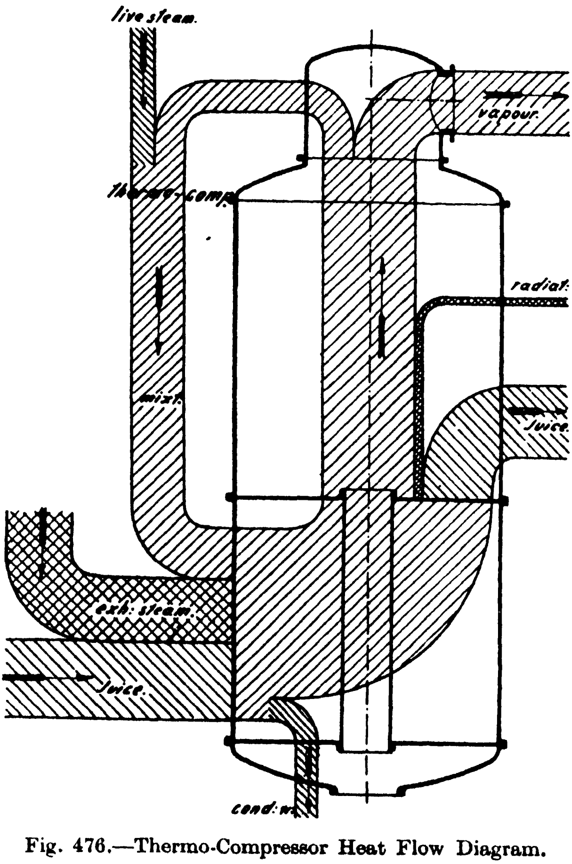


Fig. 476.—Thermo-Compressor Heat Flow Diagram.

body or vapour cell of 6000 sq. ft. heating surface, which only supplies vapours to five vertical juice heaters, having a total of 4510 sq. ft. heating surface. All the juice passes through this vapour cell before entering the quadruple effects. The juice temperature in the heaters is raised from 90 to 212°F., but the quantity of vapour required for this service amounts to a specific evaporation of about 5·83 lbs. of water evaporated per sq. ft. H.S. ; and to increase this evaporating performance, two thermo-compressors were installed, as indicated in the figure, having 3 in. steam and 10 in. vapour connexions.

In *Fig.* 476 is shown the heat flow diagram of this vapour cell, and the amounts of B.Th.U.'s are drawn to scale in the heat flow. The different values have been calculated as follows :—

Vapour to the juice heaters : 252,000 lbs. of juice have to be heated per hour from 90 to 212°F., the heat rise being 122°F. (from 32 to 90°C.). At a specific heat of 0·89 of the juice, the total amount of heat necessary amounts to : 252,000 × 0·89 × 122 =	27,362,000 B.Th.U.
Radiation losses in heaters and vapour pipes 10 per cent.	2,736,000 B.Th.U.
Heat loss in condensate from the heaters, assumed at 200°F. mean temperature, 168 ÷ (1154 — 168) = 17 per cent. 0·17 × 30,098,000	5,116,000 B.Th.U.
Total heat required for heaters..................	35,214,000 B.Th.U.

which is equal to 30,517 lbs. vapour at 3 lbs./sq. in. gauge pressure.

Live steam for thermo's. The two thermo's require 10,000 lbs. of dry steam of 90 lbs. gauge pressure, having a heat content of 10,000 × 1187·1 =	11,871,000 B.Th.U.
Aspirated vapours by thermo's. The manufacturers state that each lb. of live steam will compress 1·3 lbs. of vapour of 3 lbs. gauge pressure up to 7 lbs. back pressure. The aspirated vapours amount to 13,000 lbs. per hour, having a heat content of 13,000 × 1153·9 =	15,000,000 B.Th.U.
Heat present in juice. The juice enters the vapour cell at 200°F. (93°C.), containing : 252,000 × 0·89 × 168 =	37,679,000 B.Th.U.
The juice leaving the vapour cell amounts to 252,000 — (30,517 + 13,000) = 208,483 lbs. having a temperature of 222°F. ; and assuming the same specific heat, it contains 208,483 × 0·89 × 190 =	35,254,000 B.Th.U.

From these figures the heat balance of the vapour cell can be composed as follows :—

		In Thousand B.Th.U.'s
Outgoing :—		
Heat in vapours for heaters		.. 35,214
Heat in vapours to thermo's		.. 15,000
Heat in outgoing juice		.. 35,254
Heat lost in radiation from the vapour cell (3 per cent.) ..		.. 2,564
Heat lost in condensate of the vapour cell 200·6 ÷ (1157·8 — 200·6) = 22 per cent.		.. 19,367
Ingoing :—		
Heat in entering juice	37,679	
Heat in compressed vapour	15,000	
Heat in live steam used in thermo's	11,871	
Heat in exhaust steam (37,009 lbs. at 7 lbs. gauge pressure and dry steam)	42,849	
	107,399	.. 107,399

The juice has entered at 15·04° Brix and leaves at 18·16° Brix, whereas the specific evaporation has been increased from 5·83 up to 7·27 lbs. per sq. ft. heating surface per hour.

In *Fig.* 477 is shown an installation of thermo-compressors as designed by the author. The evaporating station of this cane sugar factory is made up of one first body, composed of three cells, having a total heating surface of 5700 sq. ft. followed by three bodies of 3100 sq. ft. heating surface each. The first body delivers vapour to the second body of 3100 sq. ft. and to three vertical juice heaters, having a total heating surface of 2250 sq. ft.

This factory had a shortage of exhaust steam, and live steam had been used on the evaporators to make up the shortage ; and as the total evaporator heating surface was unequally loaded, it was decided to install two thermo-compressors, each having a live steam consumption of 3500 lbs. per hour at a normal pressure of 90 lbs. gauge.

In the pipe lines for the vapour and steam mixture, there are provided gate valves as in the other installations mentioned. This is a necessity, as these gate valves have to be closed when the thermo-compressors are not working. These valves in some instances have been left open and the calandria pressure being higher than the vapour pressure, exhaust steam has gained admission to the upper belts of the evaporators and the evaporating performance in these bodies has nearly stopped, as the heat transmission between the calandria and the juice is only possible when there exists a difference in temperature which is equivalent to a difference in pressure between both.

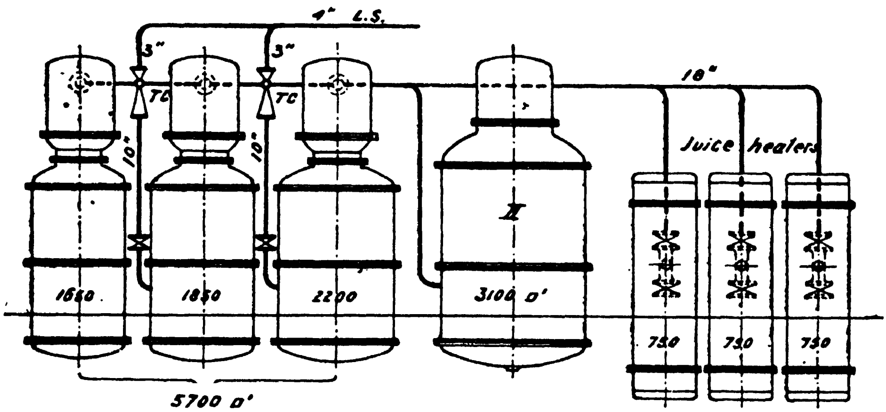


Fig. 477.—Thermo-Compressors on First Body.

With the installation of *Fig.* 477 the evaporation of the three cells was increased about 13 per cent. without any additional steam consumption. As the condensate of each body is not measured, the evaporation has to be calculated from the Brix of the entering and leaving juices. The juice samples have to be collected from small drip valves over several periods of time, so as to establish a fair average.

In *Fig.* 478 is shown another arrangement of the author's design. The evaporating plant is composed of a pre-evaporator or first body, having 6000 sq. ft. H.S., followed by three bodies of 3450 sq. ft. H.S. each. The pre-evaporator supplies vapours to three vertical juice heaters, having 2250 sq. ft. total heating surface, and to the second body of 3450 sq. ft.

The factory formerly required a considerable amount of live steam for the pan station, which was at that time small in size. By installing a large calandria vacuum pan, a larger total heating surface of 3400 sq. ft. in pan calandrias was obtained and exhaust steam as well as vapour from the pre-evaporator could be used to advantage. In case of low exhaust steam pressure, a 12 in. thermo-compressor will compress vapours into the exhaust steam main and a great flexibility in steam supply for the pans is achieved.

The three installations mentioned have worked to satisfaction under operating conditions and prove that when excess heating surface is available in the first stage of the evaporating station, thermo-compressors can be used to advantage.

Thermo-compressors can also be installed *in series*, as has been done in a large sugar refinery in U.S.A. A 10 in. and a 12 in. thermo-compressor compress respectively 10,000 and 16,000 lbs. exhaust steam of 10 lbs. gauge pressure, consuming respectively 9,000 and 15,000 lbs. live steam of 175 lbs. gauge pressure per hour, both delivering 50,000 lbs./hr. heating steam at 25 lbs./sq. in. combined.

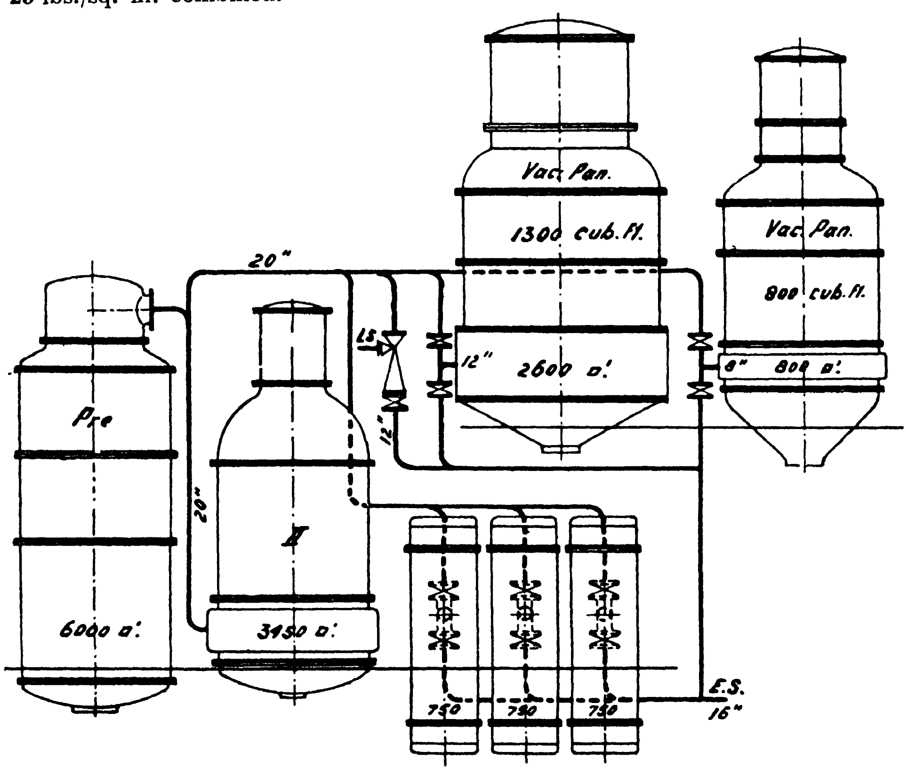


Fig. 478.—T.C. Arrangement between Vapour and Exhaust Lines.

This steam is re-compressed by three thermo-compressors up to 50 lbs./sq. in. As there had been an excess of exhaust steam, the consumption of 26,000 lbs. live steam has been replaced by the same amount of exhaust steam per hour, resulting in a decided saving in fuel.

The thermo-compressors are made in different sizes, ranging between 2 in. and 14 in. diameter of the discharge connexion. When the discharge connexion of the steam-vapour mixture is a long one, it is recommended to increase the diameter of the pipe line a few inches.

CHAPTER XXIV.

ENTRAINMENT APPARATUS.

CATCH-ALLS — STEAM TRAPS — OIL SEPARATORS — STEAM PURIFIERS.

The vapours of evaporators and vacuum pans carry with them a certain quantity of particles of the juice or massecuite from which they have emerged, which are dragged along with the vapour currents through intense boiling. These particles should be recovered, as they not only signify a direct loss (entered up under "unknown" sugar losses in the laboratory report of manufacture, which may run to 0·5 per cent. of the sugar produced); but they also may give rise to corrosive action on boilers, condensers, pipe lines and pumps—especially so when the waste water is cooled in a spray pond and thus, on re-entering the system, a vicious circle is established as sugar decomposes readily in weak solutions, forming acetic acid, and the acidity of course increases with increased additions of sugar.

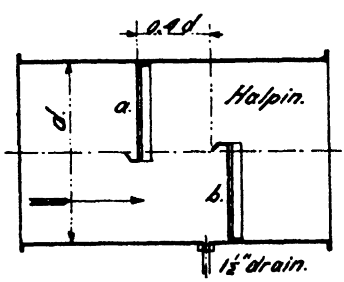


Fig. 479.—Baffle Arrangement.

The author was once called to a sugar mill, where this acidity of the cooled injection water had reached such a degree that pumps, pipe lines and condenser had corroded sufficiently to need considerable patching. Milling, of course, could not be resumed before the cause of the entrainment was eliminated.

The condensate from steam lines may do damage to the steam engines or to the turbine blading; from coils and calandrias it has to be drained regularly for better heat transmission.

Oil in the exhaust steam from reciprocating steam engines should also be removed, as it will encrust heating surfaces and may contaminate the condensate which has to be used for boiler feed. Lubricating oil is readily decomposed, forming acid components in the boilers with their corresponding corrosive action on the shell plates and boiler tubes.

The entrainment apparatus, therefore, forms a necessary part of the equipment of a sugar factory, as it prevents sugar losses and increases the efficiency of the process work, although a 100 per cent. efficiency is not always obtained.

1.—Catch-Alls.

Catch-alls, save-alls or juice-catchers, as they are variously named, are installed in the vapour lines from the evaporators or vacuum pans and can form an integral part of the latter, or may be designed as separate equipment. They have also been made as an integral part of central condensers, arranged in such a way that the vapours pass the catch-all before entering the condenser proper.

As it is important to ascertain whether there is any entrainment in a vapour pipe line, a *Baffle Arrangement* as shown in *Fig.* 479 should be installed inside the pipe. The design is for horizontally arranged vapour pipes, but can be fitted for vertical positions as well, when care is taken that all entrainment is effectively collected. The baffle a is fitted at the top part of the vapour pipe, the flow being as indicated by the arrow and it is about 2 in. over a semicircle. A ledge with the rim set at 45° is attached to this baffle for recovering any particles of liquid and this ledge is about $\frac{3}{4}$ in. shorter than the inside pipe diameter, to allow for drainage on the inside pipe walls. A second baffle b is placed at the lower half of the pipe section, also provided with a ledge as indicated in the figure. In front of this ledge is a drain connexion for collecting the entrained matter into a closed vessel (with sight glasses) of about 2 gals. capacity, provided with a drain, or a barometric leg pipe (as the author has installed in several instances) which is sealed in a container with water. After a certain lapse of time, this water will contain sugar or invert sugar, and the more rapidly it is sweetened, the greater the entrainment occurring.[1]

The distance between the baffles a and b should be such that the vapour velocity doubles ; or the free area of passage is reduced to half the full pipe section. Apparently the following equation can be deduced :—

$$\frac{\pi d^2}{4} = 2 \times d \times l \text{ or } l = \frac{\pi d}{8} \simeq 0{\cdot}4\, d.$$

The baffles, therefore, should be placed at about 0·4 the pipe diameter apart.

The most widely used type is the *Helmet Catch-all* as shown in *Fig.* 480, it being applied equally to evaporators and vacuum pans. The dimensions are from an actual installation measured by the author.

The whole design is composed of an inner tube a, which is covered by the helmet b, generally concentrically, but sometimes eccentrically arranged, and finally the outer shell c, which carries the vapour discharge outlet. Sometimes the helmet is cut as indicated by the chain dotted line at x for easier flow of the vapours.

These catch-alls are made of cast iron or steel, the former material being preferred as more resistant to corrosion.

A large drain of about 4 in. is provided at d, sometimes having a sealing cup or an inverted bend, to prevent vapours blowing through and partially short-circuiting the catch-all.

The author has seen a perforated copper tube fixed at the top of the helmet, inside the catch-all, for applying a water spray to dissolve the sugar that had actually accumulated in the catch-all and choked the passages. This clogging of catch-alls is not illusory with vacuum pans in cases of boiling-over, and it should be recollected that when water, syrup or molasses of a lower Brix is charged into a vacuum pan and mixed with already concentrated massecuite, a *gusher* might be produced, which may easily reach the catch-all. This is especially to be feared when the syrup charging device does not ensure even distribution.

The stream lines of the vapour flow are drawn in *Fig.* 480, and these may be taken as approximating closely to the actual ones, as it will be obvious that dead corners and sharp edges are rounded off. The latter in particular will cause contraction and increased vapour friction, whereas the former cause whirling effects, which are only useless hindrances so far as the separation process is concerned.

Moreover, owing to the reversal of flow at the top, the entrained matter will impinge against the inside of the helmet b on which it flows down, dripping

[1] See article from J. L. HALPIN, Proc. Queensland Soc. Sugar Cane Techn., *I.S.J.*, 1935, p. 489.

off at the bottom, whence some of it may be forced along with the vapour current towards the discharge outlet without being separated, as intended.

The *Vapour Velocity Diagram* of this helmet-type catch-all is shown in *Fig.* 481. The velocity of the vapours in the inner tube and the discharge pipe of equal diameter is to be considered as the normal velocity, marked "1·00." It will be seen that there is a decrease down to a little over 0·6 at one point only.

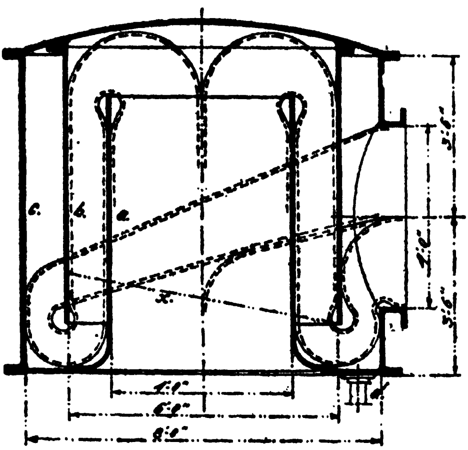


Fig. 480.—Helmet Catch-all.

The effect of separation in this type of catch-all cannot be attributed to abrupt changes in vapour velocity, where the liquor particles are thrown out of the passage of the vapours by higher inertia, due to higher specific gravity, but only to *reversal of the direction of flow*, and moreover to *abrasion* along the walls of the catch-all. It will be obvious that bubbles of liquid, which have not solidified into drops, are easily carried along in this type of catch-all. High efficiencies, therefore, should not be expected, but in many cases it may prove sufficient for the rate of flow of the vapours passing through the apparatus.

On another principle is based the *Umbrella Type Catch-all*, a German invention of several decades ago, which is of more scientific design than the one previously mentioned. It is shown in *Fig.* 482. Its principal parts are an inner tube *a*, covered at a pre-arranged distance by an umbrella *b*. The whole is arranged in an outer shell, and the given dimensions proportionate to the outside shell diameter *D* are the most convenient ones for efficient operation, as has been observed in Queensland. The vapours, emerging from the inner tube *a*, are rejected by the curved umbrella *b* and will be thrown

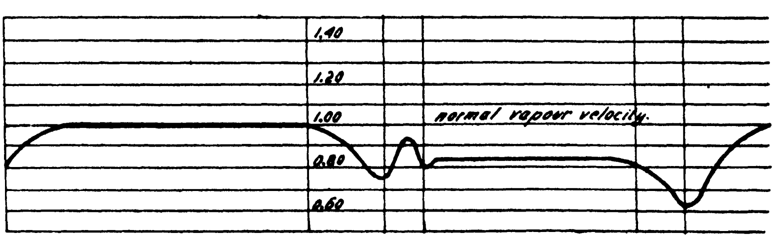
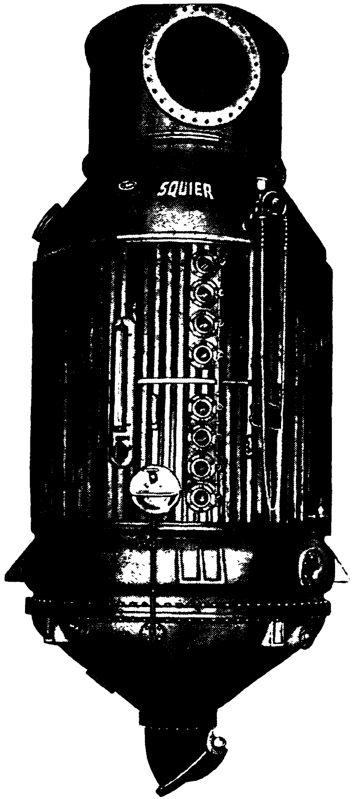
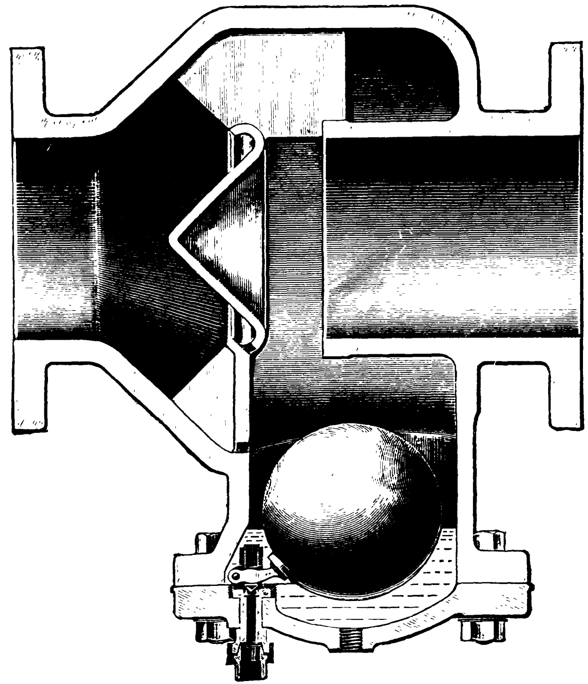


Fig. 481.—Vapour Velocity Diagram of Helmet Catch-all.

radially to the outside shell, at a downward inclination of about 20° with the horizontal. As the rejecting angle is equal to the ejecting angle with the vertical wall when the latter is assumed to be resilient, it may give rise to splashing, and the rejected particles may be forced along by the upgoing vapour current.

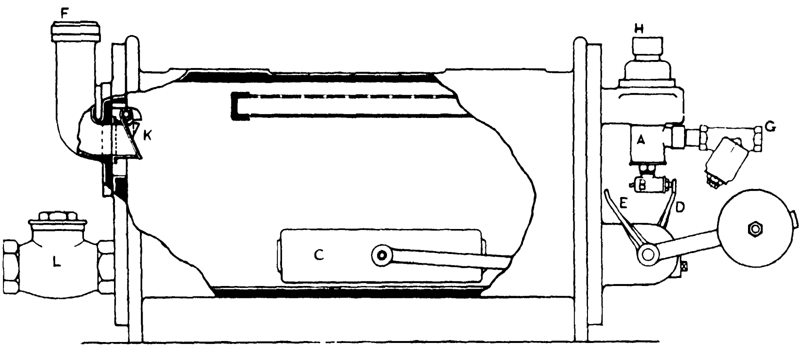


Cast Iron Calandria Vacuum Pan with Whirling Type Catch-all.
(*Squier Manufacturing Co., Inc.*)



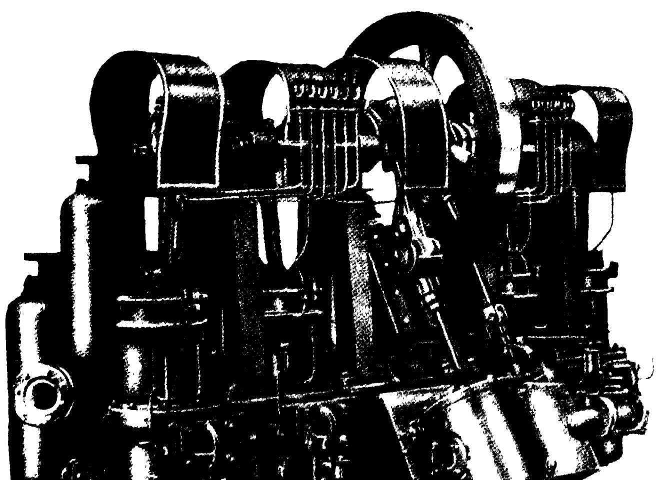
Conical Baffle Steam Separator.
(*Lancaster & Tonge, Ltd.*)

PLATES 101 & 102.



STEAM-OPERATED TRAP FOR DRAINING CALANDRIAS UNDER VACUUM.

(Royles, Limited)



FOUR-FOLD VERTICAL EDWARDS PUMP FOR CONDENSATE AND SYRUP.

(Reineveld N.V.)

On the outer shell, the author has drawn a perforated plate *e* as fitted by him in an actual case, its purpose being to avoid or reduce the splashing and to offer friction for better retention of liquid particles. This perforated plate is attached to convenient distance pieces, arranged in such a way that the free flow of the liquid down on the outer shell is not handicapped.

In *Fig.* 483 is drawn the *Vapour Velocity Diagram* of the umbrella type catch-all. In this full areas are considered, not the stream line areas indicated by *y* in *Fig.* 482, which might correspond more with what is really happening inside the catch-all. It will be seen that there is an abrupt change in the vapour-velocity, when the stream emerges from under the umbrella. This change is theoretically in the proportion of approximately 10 to 1 and takes place at the moment when the direction of flow is turned, so as to achieve the separation of the liquid particles.

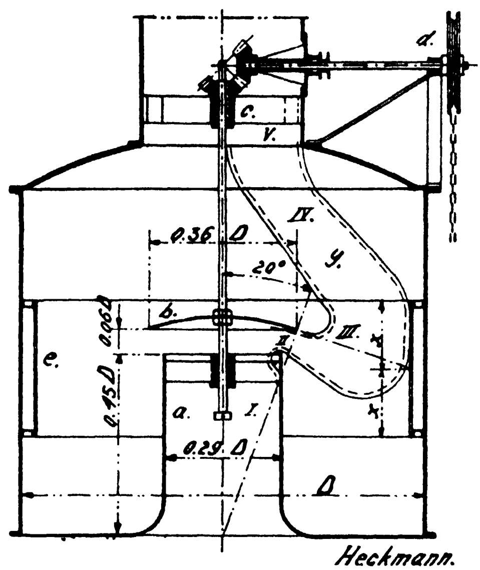


Fig. 482.—Umbrella Type Catch-all.

A vapour speed of from 150 to 200 feet per second at the umbrella discharge has been found in many instances by the author, and a fairly good separation is obtained under normal conditions.

If the rate of evaporation alters, which may happen with evaporators through incrustations or under varying grinding rates, the efficiency of a fixed catch-all of this type cannot always be maintained. With vacuum pans the evaporating performance must vary with the concentration, and although the catch-all will be most needed at the high evaporating rate when a strike

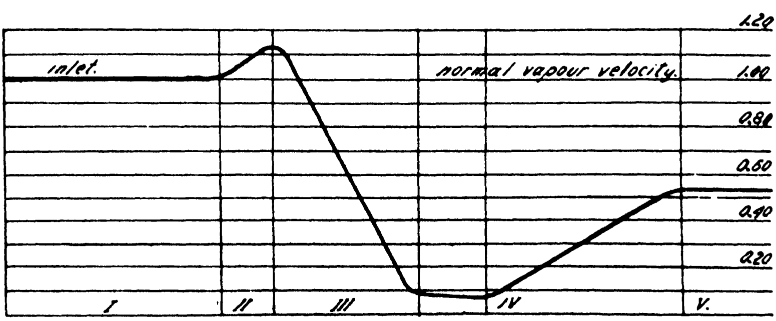


Fig. 483.—Vapour Velocity Diagram of Fig. 482.

is started, the formation of bubbles with an increase in the surface exposed to the vapour currents will be more pronounced at the end of the strike.

A provision therefore is sometimes made, to have the umbrella *b* mounted on a brass screw spindle, rotated by a brass bevel gear *c* which is manipulated from the outside by a chain wheel *d*. This arrangement is recommended by HAUSBRAND, the well-known technologist and author.

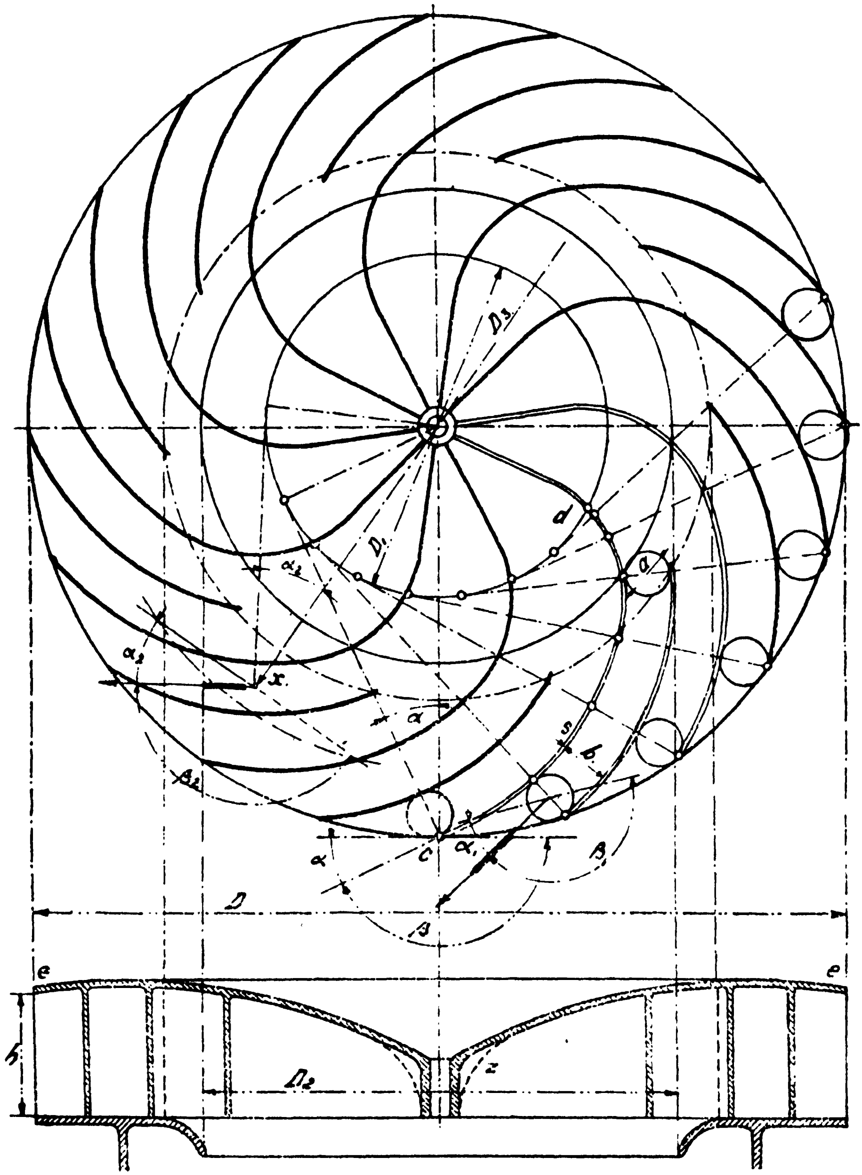


Fig. 484.—Centrifugal Catch-all Impeller.

When such an arrangement is provided, the drain of the catch-all should be brought outside the evaporator or vacuum pan on which it is installed and be equipped with sight glasses or an indicator, from which the rate of flow can be ascertained. With a little practice, the operators will soon know the most favourable position of the umbrella under varying conditions.

From the author's experience it can be said that fixed catch-alls, which had shown good operating results under normal evaporating conditions and vacuum, showed deficiencies when the evaporation rate was increased or the vacuum improved. A certain flexibility in respect to the catch-all vapour passage, as obtained by the adjustable umbrella, should therefore be advantageous under fluctuating conditions.

The design of *Fig.* 482 is rather clumsy, and unnecessary material has apparently been employed. It has too many dead corners, which may cause whirls and the change in vapour velocity will probably be other than the assumed one, when making the diagram shown in *Fig.* 483. Stream-lined catch-alls, made of cast iron, which give more freedom to the design than can be obtained with steel or copper, are receiving increased attention from manufacturers and operators.

Good operating results are also obtainable with *Cyclone Catch-alls*, which are built on similar lines to the cyclones used in combination with sugar dryers (dealt with in Chapter XXX). The construction of these cyclones is very simple and a considerable drop in vapour velocity can be achieved. Abrasive action on the catch-all walls also occurs.

An improvement upon the umbrella type catch-all is the *Centrifugal Impeller* shown in *Fig.* 484. This impeller has a two-fold purpose; first, the tangential ejection of the vapours, which causes a spiral current with efficient abrasion against the outer walls of the catch-all; and, secondly, the centrifugal action, which assists materially in the separation of the liquid particles.

In addition, the impeller is of stream-lined construction, offering less resistance to the vapour current, and splashing against the outside walls is greatly reduced, thus increasing the efficiency of the apparatus.

The design of this impeller is similar to that of the impeller of a centrifugal pump. Let D be the outside diameter of the impeller, and let the channels for unrestricted passage be constructed according to the involute form, in which the entrance width a is equal to the discharge width b. The discharge angle β is generally taken as about 155° and as the involute forms a tangent with the producing circle D_1, it follows that :—

$$D_1 = D \times \sin \alpha \quad \dots\dots\dots\dots\dots\dots \quad (122)$$

in which α is the component angle of β $(= 180° - \beta)$.

Thus the involute is produced in the customary manner from c to d, the last point being radially on the producing circle. In practice, the involute form is replaced by circular arcs, which effect only minor differences that can be neglected.

An efficient abrasion of the vapours is also obtained within the impeller. The outside diameter of the impeller can be made to suit nearly any prevailing or designing condition, it being only necessary to produce the involute shape. Of course, when circumstances permit, a larger diameter will achieve a better abrasion performance, as the blades or vanes are longer.

The inlet diameter D_2 is calculated for a vapour velocity of about 150 ft. per second. The velocity of the vapours within the impeller should not be allowed to exceed 200 ft./sec. as otherwise wiredrawing might result and the vacuum of the condenser thus not be fully felt in the connected pan or evaporator.

It will also be apparent that with a vapour velocity of $v = 200$ ft./sec. and n blades, the following equation will hold :—

$$V = n \times b \times h \times 200 \text{ or } 0{\cdot}005\ V = n \times b \times h \quad (123)$$

in which : b = width of channel in feet } see *Fig.* 484.
h = height of channel in feet }
V = volume of vapours in cub. ft./sec.

The dimension b can be easily found from :—

$$n \times (b + s) = \pi \times D_1 \quad (124)$$

s being the thickness of the blade or vane.

It will moreover be seen that at a point x the component discharge angle α_2 can be found from :—

$$D_1 = D \times \sin\ \alpha = D_2 \times \sin\ \alpha_2 \text{ and thus :—}$$

$$\sin\ \alpha_2 = \sin\ \alpha \times D \div D_2 \quad (125)$$

The design is generally so made that $D_1 = D_2$.

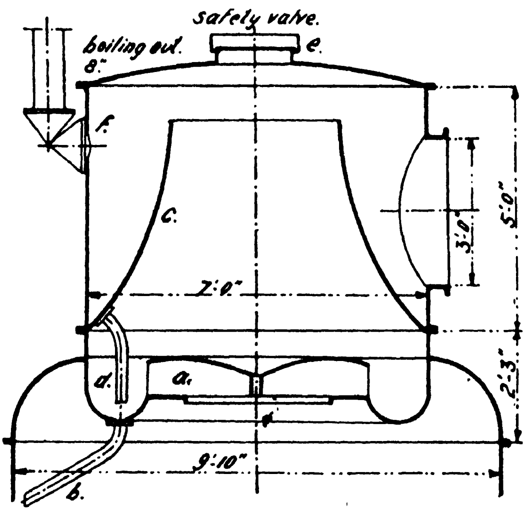


Fig. 485.—Centrifugal Catch-all.

In *Fig.* 485 is shown a *Centrifugal Catch-all* of the author's design for the coil pan of *Fig.* 452. The impeller *a* discharges into the hood *c* before the vapours go to the discharge outlet.

At *b* and *d* are the drains for the separated liquid, whereas at *f* is an 8 in. connexion leading out to the roof, provided with a stop valve for releasing the vapours of caustic soda or hydrochloric acid solutions when boiling out. These vapours are a nuisance within the boiling-house and corrode the material of the roof trusses or roof sheets, when these are made of iron. The vapour valve to the condenser is of course closed during this boiling-out operation.

At *e* is a large safety valve to neutralize pressure inside the pan when it is steamed out.

It will be seen that abrasion is fully allowed for. A large fall in vapour velocity could not be achieved, this not being required as the pan had a separate catch-all in the vapour line, just outside the pan discharge. Moreover, the vapour velocity was kept low.

2.—Steam Traps.

From the very many designs of steam traps now on the market, only a few designs of the principal types can be mentioned. These types comprise :—

(*a*) Steam traps operated by floats.
(*b*) Steam traps operated by open buckets.
(*c*) Steam traps operated by thermo-static elements.

For all steam traps the rule applies that the working parts should be as few as possible and of robust construction, causing the minimum obstruction to the passing condensate.

A *Float Steam Trap* of very plain design is shown in *Fig.* 486, having a copper float *c*, which is held in its lowest position by the screw stop *a* and thus will close the outlet nozzle *b*, which is made of non-corrosive or stainless material. The float has to be a true sphere and must resist the prevailing outside pressure. There are traps which close the inlet nozzle, so as to release the trap from inside pressure, but this is not general practice. The floats are sometimes filled with a little water, which will evaporate and develop a pressure inside the float, equal to the outside one. It is hardly necessary to mention that a leaky float impairs the working of a float trap. Exceptions are the so-called *bell floats*, which are emptied by the steam, so as to drive out the water in the trap.

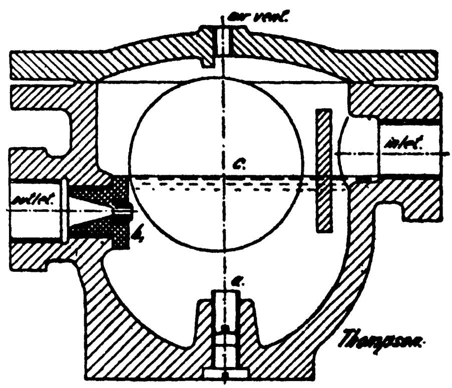


Fig. 486.—Float Steam Trap.

It will be obvious that the discharge opening is uncovered, as the float rises on the water, and in case only steam enters the trap, the float will be forced against the nozzle *b* and will close the exit.

There are also floats guided by parallel levers, which operate a *double-seated balanced valve* by means of a screw spindle with a coarse thread ; here a small lift is required for full opening.

Another type in use is the float-operated *flap-valve trap*, where the float lever opens a flap valve, which gives a large free opening for discharge. The plain *pin valve* is also used in several designs.

All steam traps should be provided with *air vents*, as air may accumulate in the trap and prevent the flow of water. Fortunately, the discharge valve is not always air-tight, but too much should not be based on this fact. For reduced pressures, like the drainage of heating bodies with low pressure vapours, this de-aeration is of paramount importance and recent designs are equipped with *automatic de-aeration* for low pressures.

It is of course understood that there must be a difference in pressure between the inlet and the discharge, so as to dislodge the condensate. Discharge from apparatus under vacuum into the atmosphere is therefore not feasible with common steam traps, and a receiving tank under the same vacuum has to be placed in the discharge circuit, which can be periodically emptied. The suction of a pump will of course render the same service for these vacuum apparatus, as already explained in Chapter XXII.

On the other hand, the condensate may have to be lifted and most of the steam traps are of the *lifting type*, allowing 2 ft. hydrostatic head for each lb./sq. in. difference in pressure between inlet and discharge.

In case the condensate needs to be raised beyond this limit, *high lift traps* have been designed. The *modus operandi* of such a trap is as follows. It is generally of the float type and as soon as the trap is filled with condensate, the connexion from the vapour space of the calandria, e.g. from where the condensate has to be drained, is broken or shut and live steam admitted. The apparatus therefore operates on similar lines to a *montéjus*, and the hydrostatic lift will amount to about 2 ft. per lb. live steam pressure prevailing in the trap. As soon as the condensate has been discharged, the float descends, the live steam connexion is shut off and the connexion with the vapour space re-established. The entry and discharge of the condensate are achieved by non-return or check valves. For discharge of condensate from heating bodies under a partial vacuum, this kind of trap can be used (vacuum-pumping trap).

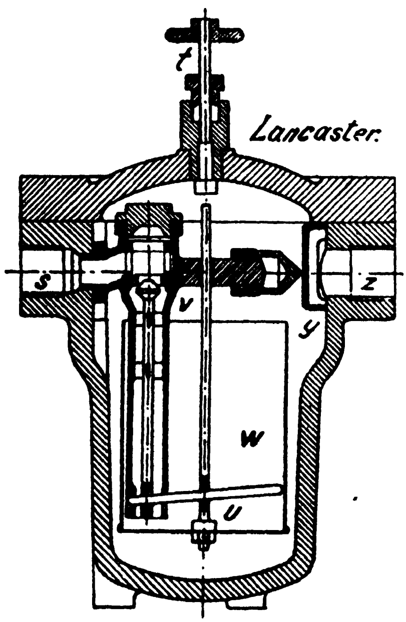


Fig. 487.—Open Bucket Type Trap.

The *Open Bucket Type Trap*, shown in *Fig.* 487, has also found a wide application. The condensate enters at z and by means of a baffle plate y it descends along the walls of the trap container. The open bucket w will float in the condensate until it overflows and it then descends, opening the valve v by means of the bar u. The steam pressing upon the liquid level discharges the condensate by the outlet s. The threaded spindle t serves the purpose of operating the bucket gear from the outside, as every trap should be equipped with such a forced discharge gear. This spindle has a centre bore, so it can be used as a blow-through or vent valve.

A *Thermostatic Steam Trap* is shown in *Fig.* 488, where its principle of operation is clearly seen. The condensate enters at d and has to be of lower temperature than the steam from which it originated. The use of these steam traps, therefore, is limited and they are not properly suited for draining heating bodies or coils, but are especially for steam lines, where there is a relatively small quantity of condensate to be drained which can be discharged at a lower temperature than the steam. The trap and the pipe connexion have to give sufficient cooling effect to make the operation feasible.

The trap is composed of an outside tube of cast iron and an inside tube w of brass or a special alloy, having a larger coefficient of expansion than cast iron. The valve head v is adjustable, the discharge being at y and an air vent is provided at u. The larger the specific expansion coefficient of the material of the tube w, compared with cast iron, the better the trap will be. The cool condensate obviously will cause the thermostatic tube to shorten, and the valve seat attached to it will clear from the valve head, thus making the trap discharge. As soon as the condensate has been released and the steam enters the trap, the thermostatic tube expands under the higher temperature and the discharge valve closes. It will be obvious that :—

$$h = L \times (\alpha - \alpha_1) \times (t - t_1) \quad \dots\dots\dots\dots\dots\dots \quad (126)$$

the symbols being : h = Valve lift in inches.
L = Length of thermostatic tube in inches.
α = Linear expansion coefficient of thermostatic tube per °F.
α_1 = Do. of outside tube per °F.
t = Temperature of steam in trap in °F.
t_1 = Temperature of condensate in trap in °F.

When there is no difference between the temperature of the steam and the condensate, then $t - t_1 = 0$ and thus $h = 0$ and the valve does not function. For full opening :—

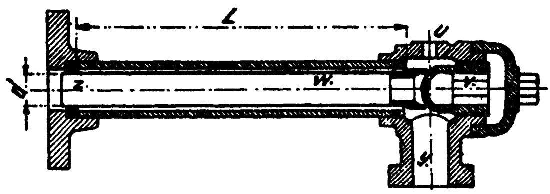


Fig. 488.—Thermostatic Steam Trap.

$$\frac{\pi d^2}{4} = \pi d \times h \times c_c \text{ or } h = \frac{d}{4} \times c_c \quad \dots\dots\dots\dots \quad (127)$$

in which : d = Inside valve diameter.
c_c = Contraction coefficient (average about 0·6).

As it will require a considerable length L of the thermostatic tube to open the valve fully, it is advisable to have the valve of a larger diameter than the thermostatic tube w in the figure.

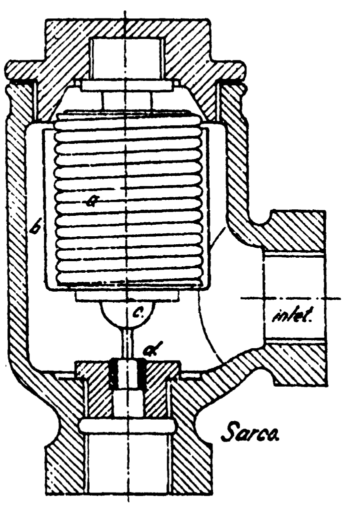


Fig. 489.—Thermostatic Steam Trap.

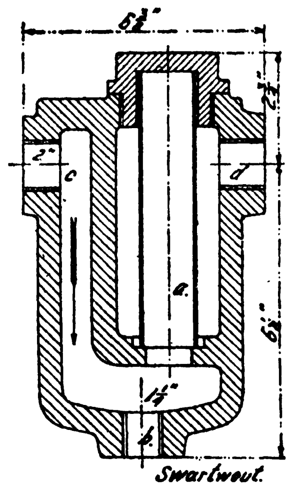


Fig. 490.—Condensate or Steam Filter.

In *Fig.* 489 is shown a *Thermostatic Steam Trap* having a tubular expansion member a, protected by a shield b against the steam and condensate flow or any dirt or scale that might enter the trap. The semi-spherical valve *c* closes upon a seat *d* of stainless material. These traps can also be used to advantage for de-aeration.

As the operating engineer is sometimes in doubt as to what might enter a steam trap, be it grit or scale, carbonized lubricating oil, fragments of sugar, caramel, or other foreign matter, all of which will harm the valves and operating gear or stop the trap functioning at all, a *Condensate or Steam Filter* should be applied in such cases. The general construction of such a filter is shown in *Fig.* 490 ; the steam enters at *c*, passes through the perforated cylinder *a* and is discharged at *d*. The grit is deposited at the bottom of the apparatus and can be periodically removed by a plug or a sluice valve, connected to this end of the filter at *b*.

The fitting of such a condensate filter in front of the steam trap will be obligatory when the so-called *labyrinth steam traps* are used. These traps, of recent design, have a large number of very narrow passages, so that the capillary action of the condensate closes the steam exit. As the passage area is in excess of the area of the connecting pipe line, the resistance or friction against the free flow of the condensate is only small, but will become great when steam has to be discharged. Labyrinth traps have not yet found general application, but are mentioned here for the sake of completeness ; there are no moving parts in them.

A stop valve has to be provided in front of each steam filter or trap, to allow for the overhaul of the equipment while the heating body to which they are connected is at work. The trap proper should have a free discharge and have a free-flowing funnel connexion to the condensate main, so as to be able to check the condensate in respect to flow and sucrose content.

The capacity of a steam trap varies greatly, as the pressure difference and the flow-friction have a bearing upon it, since it will be obvious that not the pipe connexion, but rather the inside discharge, sometimes of pin-hole proportions, is the leading factor. A continuously working trap will in general have the largest draining capacity.

In case an existing steam trap has become too small for the service required, a *flushing valve* should be applied in front of it. The action of an automatic flushing valve is obtained by means of an unbalanced discharge valve, which remains open until the pressure has risen to within a few lbs. of the working pressure, whereupon it closes and the condensate is then drained by the connected steam trap. The heavy initial condensation will thus be released by the flushing valve and it will be obvious that such an arrangement will allow a smaller sized trap to be used.

A *Steam Trap Capacity Chart* for pressures from 3 to 25 lbs. per sq. in. is shown in *Fig.* 491 ; here it will be noticed that the flap type trap, operated by a float, as indicated by full lines, will have the largest drainage capacity, as compared with the float-operated traps with double-seated valves, whose capacities are drawn by the chain dotted lines.

Float traps are built up to a 6 in. size of the pipe connexions, the bucket type from ½ in. to 3 in., and the thermostatic type up to 1 in. diameter of inlet and discharge. The drainage capacities for 5 lbs. gauge pressure are given by the manufacturers as follows :—

Float type :

½ in.	....	from 1,000 to 2,000 lbs. per hour.
1 in.	....	from 1,000 to 5,400 lbs. ,,
2 in.	....	from 4,000 to 32,000 lbs. ,,
3 in.	....	up to 15,000 lbs. per hour.
4 in.	....	up to 28,000 lbs. ,,
5 in.	....	up to 35,000 lbs. ,,
6 in.	....	up to 55,000 lbs. ,,

Bucket type :

½ in.	....	from 600 to 2,000 lbs. per hour.
1 in.	....	from 2,300 to 6,000 lbs. ,,
2 in.	....	from 9,000 to 23,000 lbs. ,,

Thermostatic type :

½ in.	....	from 300 to 350 lbs./hr.
1 in.	....	from 450 to 800 lbs./hr.

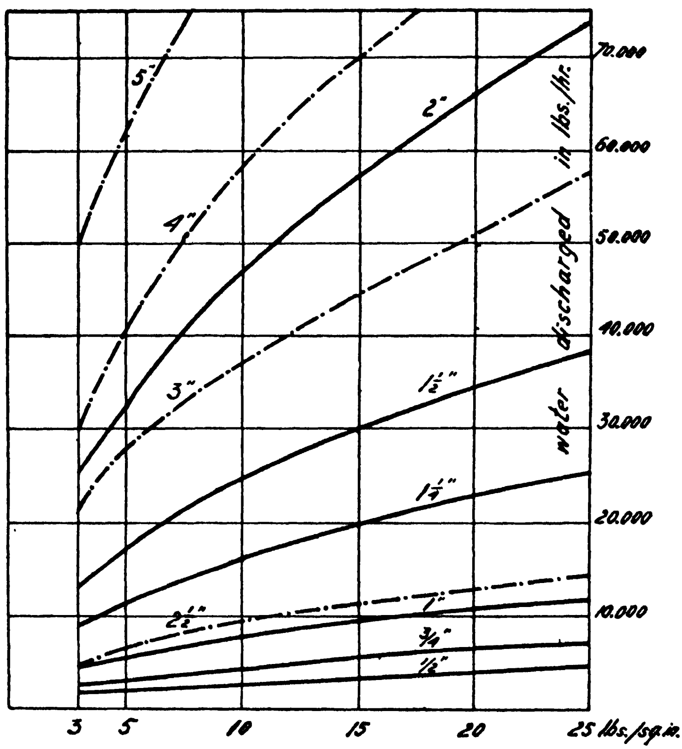


Fig. 491.—Steam Trap Capacity Chart.

For average service, the rated trap capacity should be three times as large as the normal quantity of condensate per hour. For overall calculations. the following table may be of use :—

For heaters	15 lbs.	rated capacity per sq. ft./hr.
For calandrias	20 lbs.	,, ,, ,, ,,
For coils	25 lbs.	,, ,, ,, ,,

Average values for exhaust steam lines per square foot exposed tube area are (rated capacity thus three times average drainage) :—

Uncovered exhaust steam lines .... 12 lbs./sq. ft./hr.
Exhaust steam lines with 1 in. insulation 4 lbs./sq. ft./hr.
Exhaust steam lines with 2 in. insulation 2 lbs./sq. ft./hr.

For live steam lines of about 100 lbs. gauge pressure twice the abovementioned rated capacity should be applied, the surrounding temperature being taken as about 85°F.

3.—Oil Separators.

There are two general types of oil separators for exhaust steam, viz. :—

(*a*) The chicane or baffle type.
(*b*) The inertia type.

The first comprises a container or recipient, having a plain or compound baffle, against which the flow of the steam is directed and separation of the oil particles is achieved by abrasion and shock. The compound baffles are of the grate or screen grid type, so as to form a large abrasive surface and a repeated change of the direction of flow.

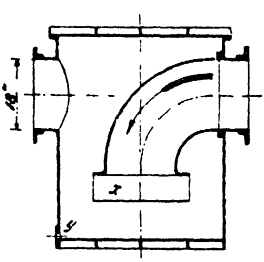


Fig. 492.
Inertia Oil Separator.

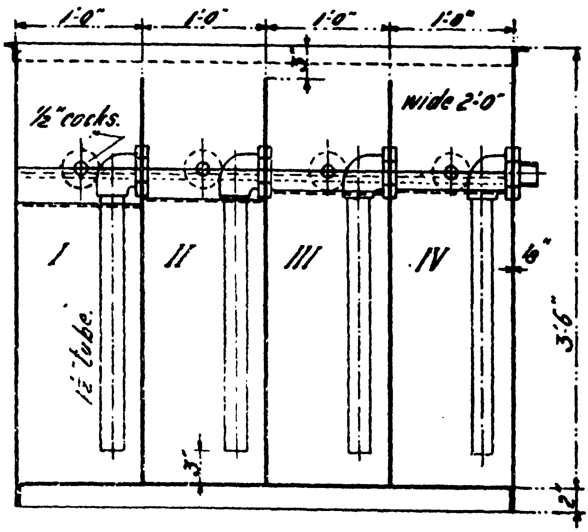


Fig. 493.—Oil Separating Tank.

The inertia type is similar to the catch-all principle as explained in *Fig.* 484. Of the author's design, and built for an 18 in. exhaust line in a mill house, is the construction of *Fig.* 492, the receptacle being of welded steel construction. The exhaust steam is spread by a welded impeller x and obviously the oil and condensate particles are thrown downwards to the bottom and thus do not have to traverse the upward steam current. The oil-condensate mixture is drained at y by means of a steam trap. The bottom as well as the removable top cover are reinforced by welded flat irons or bars.

The steam trap used has to be of an easily accessible type, so as to make periodical cleaning with kerosene possible.

The oil-condensate mixture is generally discharged into the drains or factory sewer, but it is worth while installing an *Oil Separating Tank* as shown in *Fig.* 493, which has four compartments.

The trap discharges into compartment *I*, the oil floating on the surface of the water, whereas the latter is discharged by a syphon into the next compartment and so on. As emulsification may have taken place through the turbulence caused by the trap discharge, four compartments have been incorporated, it being the common construction used on board ships. Half-inch cocks are provided at or a little below the water level for oil drainage. The operation of these tanks is quite satisfactory, the design of *Fig.* 493 being for the discharge of a 1 in. float trap. The larger the tank, the better the results of course.

The recovered oil can be used for secondary purposes, like gear lubrication, thread-cutting, rail car journal lubrication and the like. The oil is sometimes strained over a piece of woven material, so as to remove any dirt that may have mixed with it. To prevent entry of dust, the tank should have a cover.

There is also a mechanically-operated oil separator of the inertia type on the market, having a kind of turbine blading with a mechanically-operated impeller, so as to increase the inertia or centrifugal force. Although this type is costly, the separation obtained is good.

4.—Steam Purifiers.

The design of steam purifiers is based on the same principle as that of oil separators and it is a useful piece of equipment, as it will prevent water hammer in steam lines or engines, or erosion of turbine blades. In case the boilers are equipped with superheaters, these purifiers or *tracifiers*, as they are sometimes called, have to be arranged in the boiler drums or between the boilers and the superheaters. Superheating can only be applied to moisture-free steam, and with wet steam the superheating performance may be rendered invalid, as the superheater would become merely a saturated steam dryer.

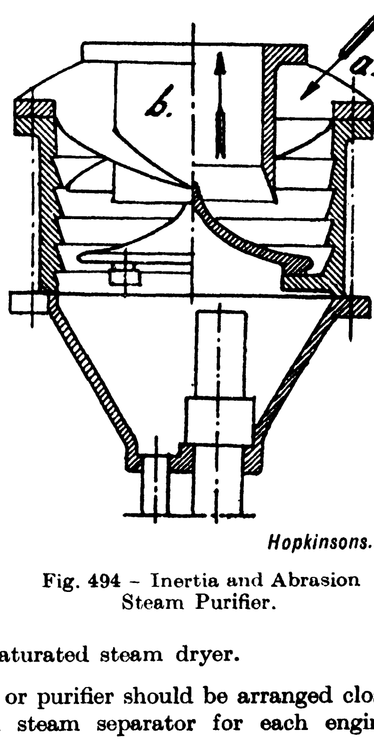


Fig. 494 – Inertia and Abrasion Steam Purifier.

For saturated steam, the steam dryer or purifier should be arranged close to the prime movers, and an individual steam separator for each engine can also be counted good practice. Baffle as well as inertia steam separators are on the market.

An *Inertia and Abrasion Steam Purifier* is shown in *Fig.* 494, which can be built into a live steam receptacle. The steam enters at *a* and a whirling action is imparted by the scroll blading. Abrasion against the saw-toothed walls takes place and the condensate is discharged at the bottom. The steam leaves on the top at *b*.

CHAPTER XXV.

CONDENSING EQUIPMENT.

PRINCIPLES OF CONDENSING — CONSTRUCTION AND DETAILS — AUXILIARY CONDENSING EQUIPMENT — SPRAY AND COOLING PONDS.

Vacuum condensing in the process of making sugar has become compulsory to obtain a large heat drop in the multiple evaporating plant and to allow for boiling at a relatively low temperature in the vacuum pan.

Two kinds of vacuum condensing are applied for process work in general, the first being based on the direct mixture of the vapours with the injection water, and the other is where these two mediums are separated by a metallic wall, the vapours generally being passed over the exterior of the tubes, whereas the cooling water is circulated inside them.

The term describing the first type is *jet condensation*, whereas the second type is called *surface condensation.*

In cane sugar factories, the final condensing is achieved by jet condensation, thus by the direct mixture of vapours and injection water in a vessel under vacuum. The condensate is mixed with the cooling water and leaves with the so-called waste water.

Surface condensing is also found in the cane sugar factory, as all heating bodies, like calandrias and coils belong *de facto* to the second type ; from these the pure condensate is recovered for boiler feed and other purposes.

Power plant condensing, not to be obtained in the already mentioned heating bodies (which are condensers under atmospheric or gauge pressure) but in vacuum condensers, is carried out only in a few cases, but it might be of interest in future developments and will be treated in the corresponding Chapter XXXIV on Power Plants.

1.—Principles of Condensing.

The principle of jet condensing is obviously the transmission of the total heat contained in the vapours to the cooling or injection water. The increase in heat content of the latter thus necessarily has to be equal to the total heat content of the vapours ; the heat carried away by the withdrawn air or incondensible gases being neglected. The equation can thus be established :—

$$I - (t_i - 32) = W \times (t_i - t_{ii}) \quad \text{or :}$$

$$W = \frac{I - (t_i - 32)}{t_i - t_{ii}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (128)$$

in which :

W = the weight of injection water in lbs. per lb. of vapour to be condensed (i.e. the *cooling water ratio*).

I = the total heat in B.Th.U./lb. of vapour.

t_i = the temperature of the waste water in °F.

t_{ii} = the temperature of the injection water in °F.

For metric units, the formula reads :

$$W = \frac{I - t_i}{t_i - t_{ii}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (128\text{A}).$$

where I is in cal./kg. and t_i and t_{ii} in °C.

For tropical cane sugar factories where average conditions are t_i = 108°F. (42°C.), t_{ii} = 86°F. (30°C.) and 26½ in. vacuum, based on 30 in. barometric pressure, the cooling water ratio will amount to :—

$$W = (1113 - 76) \div 22 \simeq 47.$$

In those factories where a cooling pond is at hand for back cooling of the waste water, the prevailing temperatures t_i and t_{ii} are higher and are generally about 96°F. (36°C.) and 118°F. (48°C.) respectively. As the vapour temperature has to be above the waste water temperature, so as to make heat transmission possible, the vacuum obtainable will be influenced by the waste water temperature. The thoroughness of the mixture between the injection water and the vapour has a bearing upon it and the better the condensing performance, the lower the difference will be. From practical observations, the author has found with barometric condensers, that the difference between vapour and waste water temperature ranges between 5° and 20°F., and in some cases, where an unnecessarily large cooling water ratio had been applied, even larger. A difference of 10°F. may be considered as a very good performance, and with back cooling a higher water ratio has to be applied, when the same vacuum is to be maintained. An overall figure of 50 will fully cover the requirements.

Now the capacity of the injection pump is normally given in Imp. or U.S. gals./min., whereas the quantity of vapours is in lbs./hr. and the conversion formulae read, with a cooling water ratio of 50 :—

$$V_i = \frac{W_v \times 50}{60 \times 10} = \frac{W_v}{12} \text{ Imp. gals./min.} \quad \ldots\ldots\ldots \quad (129)$$

and similarly :—

$$V_i = \frac{W_v \times 50}{60 \times 8{\cdot}32} = \frac{W_v}{10} \text{ U.S. gals./min.} \quad \ldots\ldots \quad (129a)$$

in which :—

V_i = the volume of the injection water in gals./min.

W_v = the weight of the vapours in lbs./hr.

For 36,000 lbs./vapour per hour to be condensed, there are thus required 3000 Imp. or 3600 U.S. gallons of injection water per minute.

Although the sugar liquors in the evaporators and vacuum pans may be deemed air-free through repeated boiling, in practice there are incondensible gases or air entering the condenser during its operation, which have to be removed and an air or vacuum pump becomes a necessary part of the condensing equipment.

This air (all incondensible gases entering a condenser are called "air") has come from three sources :—

(a) From the injection water, especially when from a cooling pond where the water has been aerated.

(b) From air leakages.

(c) From incondensible gases like ammonia present in the juice (much more frequent in beet than in cane juices).

A standard method is not feasible for calculating the quantity of air to be removed, as it varies beyond control and a happy average has been found by HAUSBRAND from practical observations, it being 20 per cent. of the volume of the injection water and this volume is taken at atmospheric pressure. The specific gravity of air being 0·00125 at average temperature, the weight of the air to be removed from the condenser amounts to about 0·25 per thousand of the weight of the injection water. For sugar-house work, this figure is too high and 0·18 per thousand on weight (15 per cent. on volume) is a good practical figure, according to the author's experience.

The volume of the air, according to MARIOTTE's law, depends upon the air pressure, and the pressure in a condenser is made up of the sum of vapour and air pressures. DALTON's law is thus written :—

$$P_{cond.} = P_{air} + P_{vapour} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (130)$$

The vapour pressure now depends upon the temperature at the spot where the air is extracted from the condenser and it will be obvious that a lower temperature will produce a lower vapour pressure. With a lower vapour pressure $P_{vap.}$ it will also be clear that with the same condenser pressure $P_{cond.}$, the air pressure P_{air} *increases* and thus the volume will decrease, after the above-mentioned law of MARIOTTE. The air extraction should therefore be carried out at the coolest spot in the condenser, as this will materially reduce the required volumetric displacement of the vacuum pump.

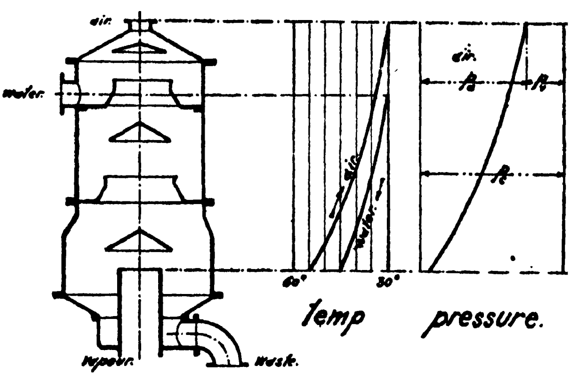


Fig. 495.—Counterflow Type Condenser.

Two systems now have come into vogue for sugar-house jet condensing, one being called the *dry vacuum* and the other the *wet vacuum* system.

The dry vacuum principle is graphically explained in *Fig.* 495. The condenser is of the *counterflow type*, the air and waste water being released at opposite points. The top part is the coolest and the vapour pressure P_v gradually decreases from the vapour entrance towards the air extraction outlet, as diagrammatically shown on the right side of the Fig. The temperature of the air decreases too and that of the injection water increases, as indicated by the arrows in the temperature diagram in the middle of the Figure.

These counterflow condensers are generally arranged at barometric height, this being the reason why they are called *Barometric Condensers.* The waste water pipe (also called a barometric leg, seal or tail pipe) has to form a water column, which will exert a pressure at its base higher than the prevailing absolute pressure within the condenser and the waste water thus will be discharged by gravity. For each in. of vacuum 1·11 ft. tail pipe length is required and for absolute vacuum, which is assumed here as 30 in., a tail pipe length of 33·4 ft. will be the lowest safety limit. In practice, the tail pipe is generally made a few feet longer, so as not to flood the condenser by obstructed flow or similar hindrances to the waste water discharge.

The *absolute vacuum*, assumed beforehand at 30 in. mercury column in the Toricellian tube, varies, but at sea lèvel this figure can be taken without incurring any great error. For higher elevations, the absolute vacuum will decrease about 1 in. on the mercury column for each 900 ft. height, and therefore factories located at places well above sea level never reach as high a vacuum on the mercury column as those on the lowlands. This, however, has no influence on the condensing performance, as the absolute pressure is the deciding factor.

The injection water (see *Fig.* 495) is assumed to enter at 30°C. (86°F.) and the vapours at 55°C. (131°F.), which corresponds with a condenser pressure of about 25·35 in. vacuum. The air is deemed to be released at the temperature of the injection water, and the prevailing vapour temperature at this point thus amounts to 0·613 lbs./sq. in. abs. (86°F.). The condenser pressure at 25·35 in. vacuum is 2·279 lbs./sq. in. abs. and the air pressure accordingly :—

$$2{\cdot}279 - 0{\cdot}613 = 1{\cdot}666 \text{ lbs./sq. in. abs.}$$

As the atmospheric pressure is 14·7 lbs./sq. in. abs., the volume according to MARIOTTE's Law is increased :—

$$14{\cdot}7 \div 1{\cdot}666 = 8{\cdot}82 \text{ times.}$$

Taking a condenser pressure of 1·465 lbs./sq. in. abs. (27 in. vacuum) and the other condition of air releasing temperature being equal, this increase amounts to :—

$$14{\cdot}7 \div (1{\cdot}465 - 0{\cdot}613) = 17{\cdot}25 \text{ times.}$$

Hence, it will be easily apparent that the air pump displacement required will greatly increase with increased vacuum. The increase in injection water temperature has the same effect, as with 36°C. (96°F. vapour pressure 0·838 lbs./sq. in.) the proportion amounts to : 14·7 ÷ (1·465 — 0·838) = 22·44.

A low injection water temperature is therefore of the greatest importance for maintaining a high vacuum.

With a 20-fold increase in volume through the MARIOTTE expansion, the air pump displacement V_a for the dry vacuum obviously amounts to :

$$V_a = \frac{V_i \times 0{\cdot}15 \times 20}{6{\cdot}23} = 0{\cdot}48\, V_i \text{ cub. ft./min.}$$

or generally :—

$$V_a = \frac{V_i \times 0{\cdot}15 \times 14{\cdot}7}{6{\cdot}23 \times P_a} = 0{\cdot}353\,\frac{V_i}{P_a} \quad \ldots\ldots\ldots\ldots (131)$$

in which : V_a = dry vacuum pump displacement in cub. ft./min.

V_i = volume of injection water in Imp. gals./min.

P_a = the absolute air pressure at the condenser air outlet in lbs./sq. in.

The air volume has been taken as 15 per cent. on the injection water volume and the atmospheric pressure 14·7 lbs./sq. in. abs.

For U.S. gallons the divisor 6·23 has to be changed to 7·48.

The wet vacuum principle is shown in *Fig.* 496 ; the condenser being of the *parallel flow type*. The waste water and air are withdrawn together by the wet vacuum pump connected to the condenser proper. This type of condenser is therefore sometimes called a *low level condenser*, although other arrangements are feasible and counterflow condensers can also be placed at a low level.

The injection water is sprayed into the condenser and, the condenser pressure being assumed as 2·279 lbs./sq. in. absolute as in the previous case, it will be understood that the vapour pressure corresponding with 113°F. (the temperature of the waste water) has to be considerably higher (1·386 lbs./sq. in. abs.), thus maintaining an air pressure of 0·893 lbs./sq. in. abs. and the expansion of the air thus becomes : 14·7 ÷ 0·893 = 16·4 times as compared with 8·82 times with a dry vacuum.

The wet vacuum pump displacement is derived from the fact that air as well as the waste water is removed by the same pump :

$$V_a = \frac{V_i}{6{\cdot}23} + 0{\cdot}48\ V_i = 0{\cdot}64\ V_i \text{ cub. ft./min.}$$

or in general :—

$$V_a = 0{\cdot}16\ V_i + 0{\cdot}353\ \frac{V_i}{P_a} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (132)$$

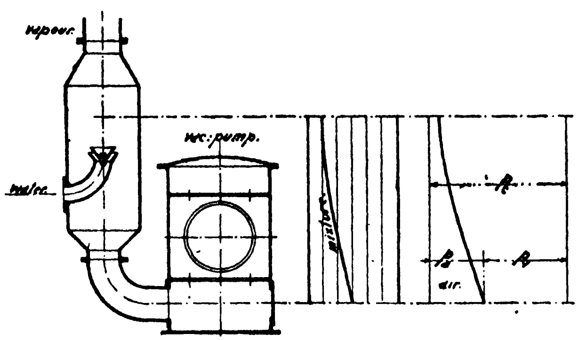


Fig. 496.—Parallel Flow Type Condenser.

For quantities of injection water given in U.S. gals., the divisor 6·23 changes to 7·48 as with formula (131).

The wet vacuum pump thus requires in these cases well over 1·5 times the displacement of a dry vacuum pump, as V_i for wet vacuum is larger than for dry vacuum ; but it should not be overlooked that only one pump is required and not two. For smaller installations, the wet vacuum system is still in use, but for larger ones and for *central condensers*, i.e., the arrangement with a single condenser for all the evaporators and vacuum pans of a factory, the dry vacuum system has inherent advantages.

The question has been mooted whether a central condenser or a number of individual condensers is to be preferred. Mechanically, a central condenser is the better arrangement and less costly with its elimination of a lot of piping, but the manufacturing staff, especially when different varieties of consumption sugars have to be boiled, often prefers individual condensing, which gives greater flexibility in respect to variation in vacuum and pan boiling temperatures.

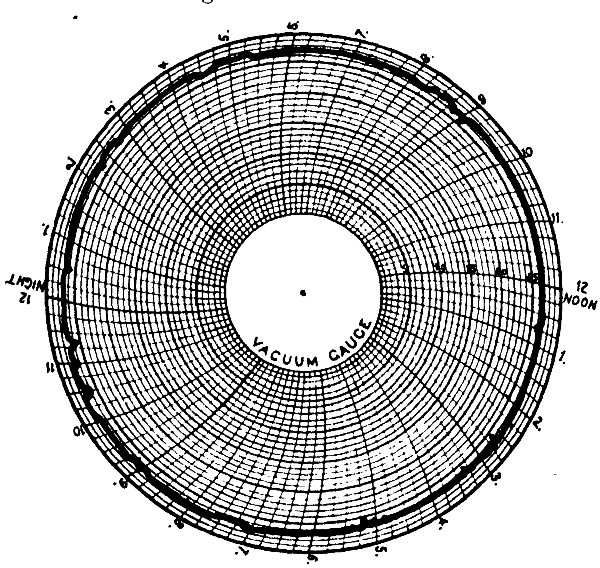
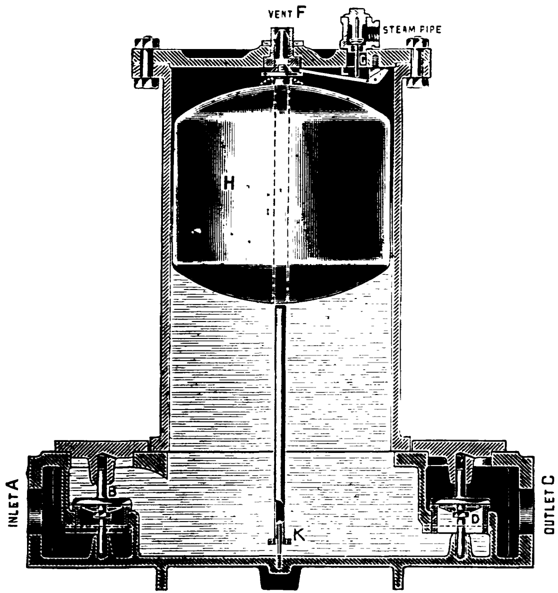
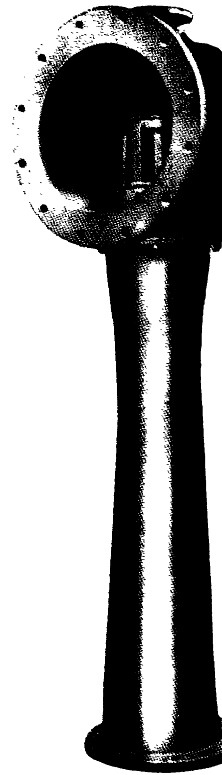


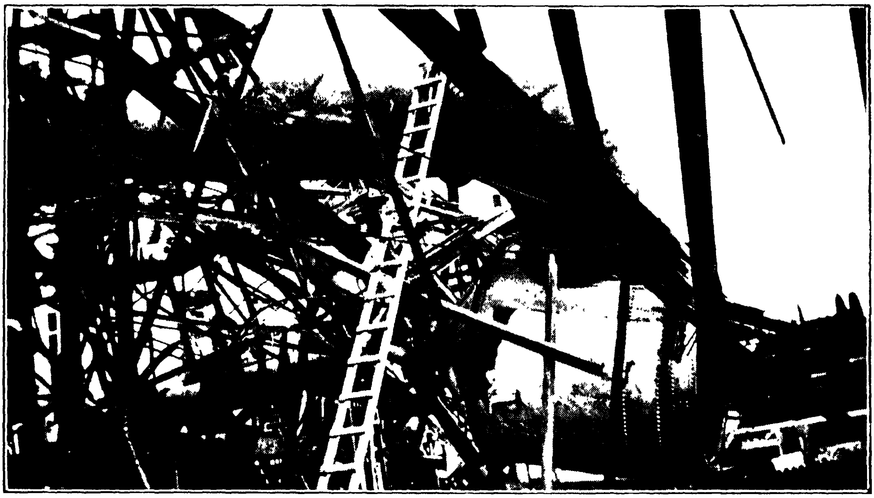
Fig. 497.—Central Condenser Vacuum Recording Chart.



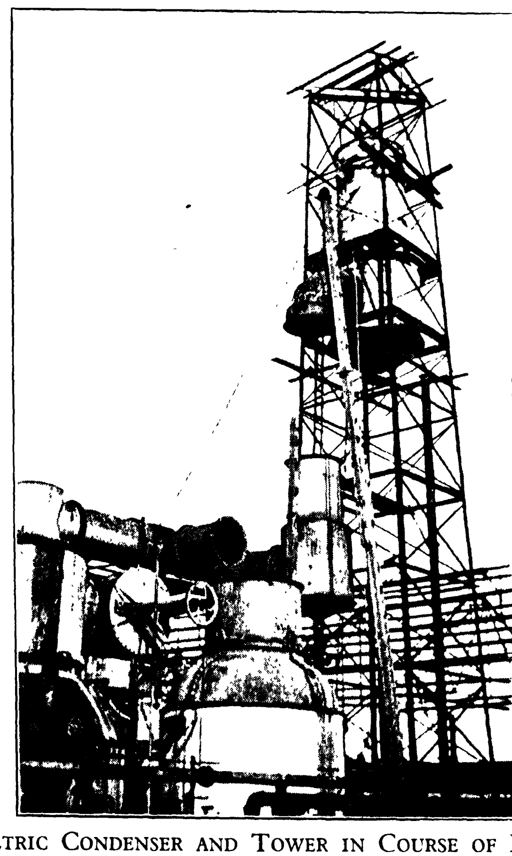
PUMPING STEAM TRAP.
(*Lancaster & Tonge, Ltd.*)



THERMO-COMPRESSOR.
(*Korting Bros.* (1917) *Ltd.*)



Barometric Condenser Tumbled Over during a Cyclone.



Barometric Condenser and Tower in Course of Erection.

Many installations are not built to a true individual system, although each vacuum pan has an individual condenser. The injection water supply has been centralized in nearly all installations and this does not alter the performance in each condenser, but complications may be expected when a central vacuum pump is connected to a main vacuum line, as differences in absolute pressure in individual condensers then tend to equalize. In some installations each pan is accordingly provided with an individual vacuum pump, which is the best arrangement for individual condensers, but costly.

A further point of argument is the fear that a constant vacuum cannot be maintained with a central condenser, when an empty vacuum pan is cut in and the vacuum will thus show fluctuations. This inconvenience can be easily overcome with an *auxiliary or booster vacuum equipment*, and each central condensing plant should be equipped with one for good operation. This booster pump will produce a vacuum in the empty pan, before it is connected to the main condenser. From the author's file is copied *Fig.* 497, a *Vacuum Recorder Chart* of the average operating performance at a Cuban cane sugar factory, having a central condenser, where the vacuum *de facto* is kept within very close limits.

Sometimes two condensers are provided, one for the evaporators, having a vacuum of about 24 to 25 in., and one for the pan station with a vacuum from 26 to 27 in.

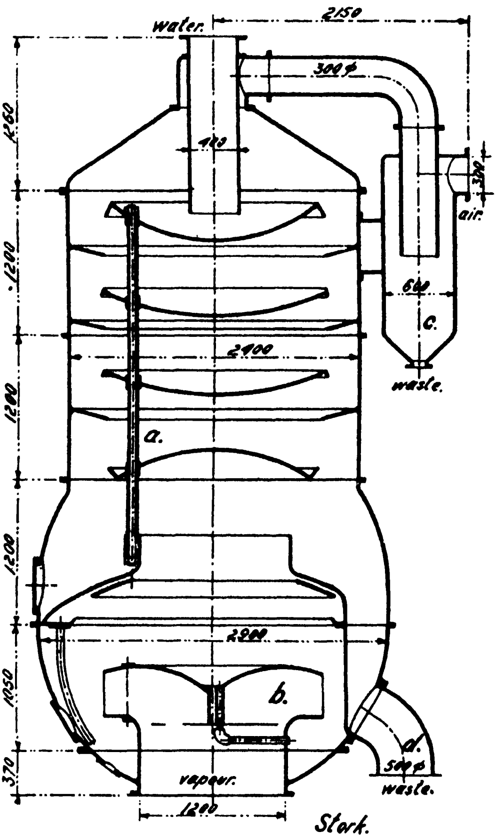


Fig. 498.—Multi-Curtain Condenser.

2.—Construction and Details.

A *Cast Iron Central Condenser* of a type furnished by the author is shown in *Fig.* 498. This *multi-curtain or baffle type* is of the counterflow, barometric arrangement, the air being efficiently cooled before it is withdrawn from the condenser. Any entrainment will be separated in the vessel c.

The injection water is charged through the top into a tray and 7 baffles and trays are formed before it reaches the bottom. The trays have to be fixed quite horizontal, and after the condenser has been erected, water should be pumped in with manholes open, so that the performance inside can be observed. Sometimes the manometric head of the pump is not sufficient to pump the water into the condenser without vacuum, in which case this test cannot always be made.

The trays are supported by three rods *a*, having tubes as distance pieces between the trays.

The vapour entrance is at the bottom and a centrifugal type catch-all *b* is embodied in the design. The condenser is self-supporting on the large vapour pipe, which has a special T-piece for the vapour pipe connexions. The waste water is discharged at *d*. The diameter of this condenser is twice the vapour pipe diameter, but there is no rule as to this proportion. An ample passage has to be provided for the air and injection water, as the author has known cases where this has not been done, especially when the condenser through enlarged capacities of the factory has had to cope with a much larger amount of injection water than that for which it had originally been designed and the condenser actually got choked.

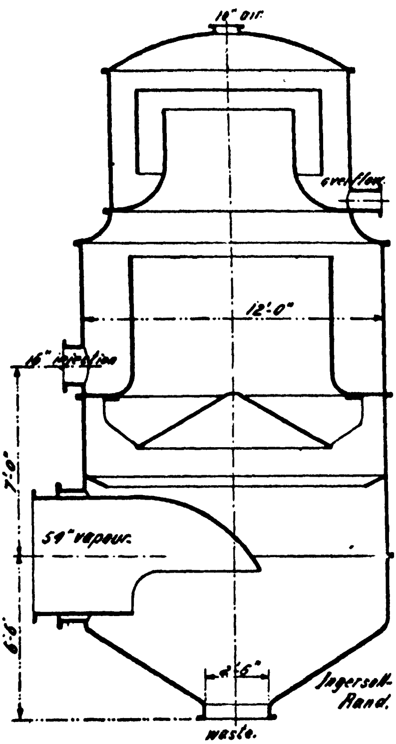


Fig. 499.—Barometric Counterflow Condenser.

Another type of *Barometric Counterflow Condenser*, shown in *Fig.* 499, is one that the author has seen in actual operation. A thin water-curtain is produced and little air resistance is encountered. The difference in vacuum before and after the condensing is a sure indication whether choking occurs. This condenser is connected to the last body of a quadruple effect, having 7000 sq. ft. heating surface. Some 42,000 lbs. of vapour are condensed by 4350 U.S. gals. water/min. of 90°F. inlet and 110°F. discharge temperature. The empty condenser weighs about 15 short tons.

Observant engineers have adopted the view that condensing is an instantaneous performance and a good intermixture is the only thing required. This has led to the design of the *Rain* and *Spray Type Condensers*, which have given very good results. The perforated tray has holes of from $\frac{3}{8}$ to $\frac{1}{2}$ in., but the inconvenience is that adhering silt or mud or foreign matter easily clogs these holes and necessitates straining the injection water through $\frac{1}{4}$ in. perforated plates (not wire gauze). A few baffles should be provided to prevent the air current from short-circuiting.

The condenser of *Fig.* 499 is supported on the tail pipe and a few cable guys are not superfluous. The catch-all is of the helmet type to prevent entrainment of the air current.

Many condensers are supported in towers and such a *Condenser Tower* (designed by the author) is shown in *Fig.* 500, to replace a previous one which had collapsed in a cyclone, as shown in Plate 105 after the débris had been cleared for taking the photograph. In Plate 106 the tower of *Fig.* 500 is shown *in situ* and it is noticeable that the erection of the condenser is greatly simplified. For this purpose two diagonally placed 12 in. I-beams are arranged at the top square with a $1\frac{1}{2}$ in. U-bolt for attaching the hoisting gear. It is hardly necessary to mention that the condenser should be hoisted in sections for safety's sake. The author once saw a condenser coming down, where this precaution had not been taken, and the beams and hoisting gear failed under a load for which they were not intended.

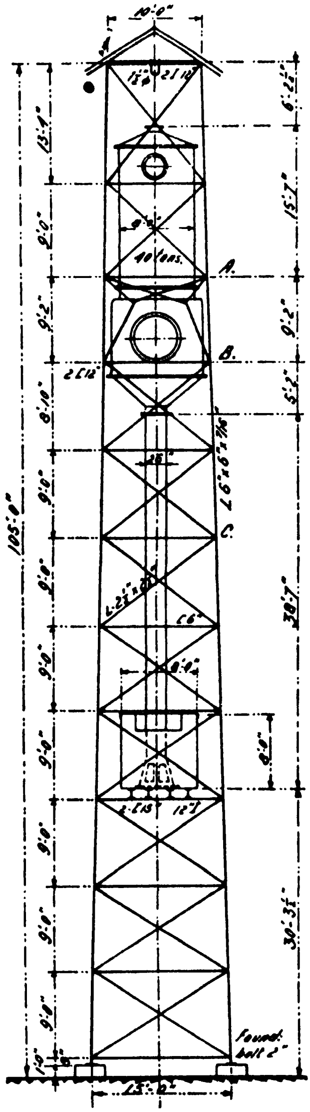


Fig. 500.—Condenser Tower.

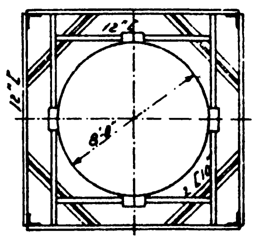


Fig. 501.—Upper Supporting Frame in Fig. 500.

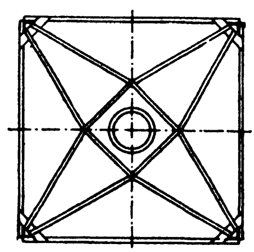


Fig. 502.—Horizontal Bracing in Fig. 500.

The condenser's weight of 40 tons is under operating conditions with attached piping. The frame of the tower is supposed to withstand a wind pressure of 42 lbs./sq. ft., assuming for calculating purposes that the sides are covered with sheeting, which actually is not the case. Stairs will make access easy, but ladders will do as a last resort.

The tower has a height of 105 ft. and the condenser is placed at such a level, that the discharge of the seal tank of the tail pipe will remain well above the height of the cooling tower. This seal tank must have a capacity above the seal openings, sufficient to fill the tail pipe 34 feet. A liberal excess is convenient, as a break in vacuum through the tail pipe will cause the whole condenser to move. Horizontal sections of the tail pipe should not be allowed.

The waste water conduit is sloped about one foot per hundred, which gives the water a velocity of about 15-20 ft./sec. The relation of bottom width to height is generally made 4 to 1 and at curves the convex sides have to be raised to 1·5 times the normal height. These curves must not be abrupt as the water would then be thrown over the sides.

The discharge of the seal tank has been used by the author as a weir for calculating the quantity of waste water discharged. This is of special importance, where use is made of centrifugal pumps, which sometimes deliver considerably less than that indicated on the manufacturer's label, due to differences in manometric head, seal ring leakage or other causes.

The condenser tower has to be reinforced by diagonal bracing and in this case a special construction had to be made at the locus of the big vapour pipe of the condenser. The condenser proper bears upon the tower by means of footings at *A* and *B*.

In *Fig.* 501 is shown the upper supporting frame at *A*. Sometimes it is forgotten to provide horizontal bracings in these tower structures, although these give additional resistance against collapsing and torsional stresses. In *Fig.* 502 is shown such a horizontal bracing at *C* in *Fig.* 500, in which clearance had to be provided for the tail pipe. The latter is not supposed to be a support for the condenser although it will give additional rigidity.

3.—Auxiliary Condensing Equipment.

Injection and vacuum pumps will be treated in the corresponding pump Chapters, but air extractors, operated by steam or water, now form efficient auxiliaries for the condensing plant, so will be referred to here. A *Steam-operated Air Ejector* or ejector air pump is shown in *Fig.* 503 ; it is built on similar lines to the thermo-compressors previously dealt with. Steam jet air pumps are nowadays applied to an increasing extent for the condensing plant of up-to-date power plants. The arrangement there is in series of two or three ejectors with inter-cooling and for high vacua the efficiency is above those of the mechanically-driven reciprocating or rotary air pumps.

For sugar factory work these high vacua are not required for process work ; but through its plain construction without moving parts, this type of pump has value as an auxiliary or booster equipment for central condensers, to produce a vacuum in a discharged pan before it is connected to the central condenser.

The minimum available steam pressure has to be given, when ordering these ejectors, as they might stop pumping below the pressure for which they were designed.

These ejectors are built in sizes having from 50 to 2000 lbs./hr. steam consumption and with 100 lbs. gauge pressure will evacuate 1000 cub. ft., at a maximum of 25 in. vacuum, in from 4·75 to 1·90 minutes respectively.

Steam ejectors are preferably equipped with a pre-cooler, this being a vessel with a water spray, to reduce the air volume to be aspirated by the ejector. It will be apparent that the efficiency of the booster equipment will be increased thereby.

The same principle is to be found in the *Water-operated Air Ejector*, and the general arrangement of an individual pan condensing plant is shown in *Fig.* 504. The efficiency is high and the field of application has already reached beyond the booster or priming stage.

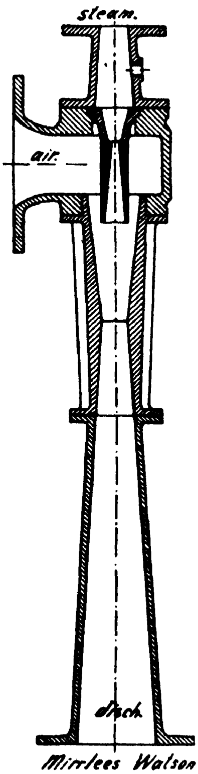


Fig. 503.—Steam Operated Air Ejector.

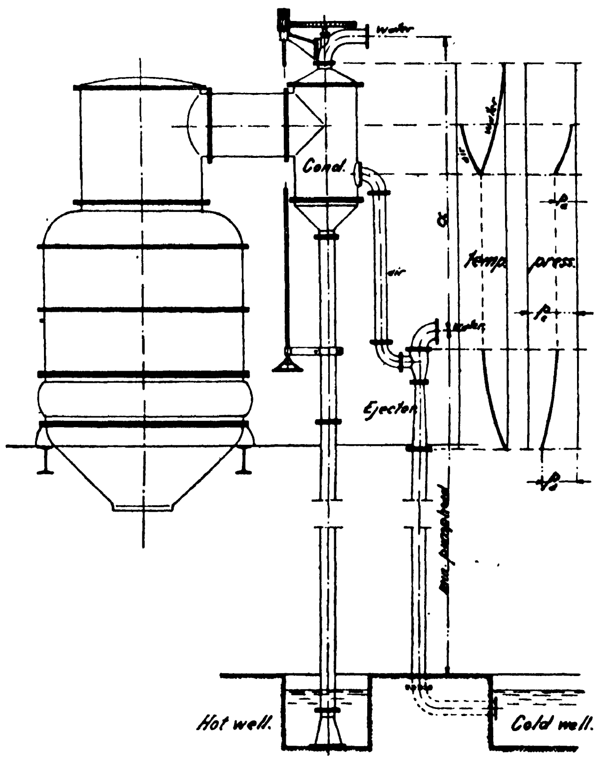


Fig. 504.—Parallel Flow Condenser with Ejector.

The condenser in *Fig.* 504 is of the previously described parallel flow type, having an adjustable umbrella spray at the top of the apparatus ; the waste water is discharged by a barometric tail pipe and a baffle inside the condenser is provided to prevent the waste water entering the orifice for air extraction.

The total quantity of injection water required is split into two parts, one for the condenser and the other for the air ejector ; and from the temperature and air pressure diagrams, drawn at the right hand side of the Fig., it will easily be seen that the dry vacuum performance is accomplished in two stages : the condenser delivers the air-vapour mixture at the waste water temperature, and the latter is instantly cooled when entering the air-ejector and the waste water from the latter is practically not heated and is returned to the cold well.

The orifice of the ejector is provided with cross ribs or webs, so that the water jet will be divided into four quadrant-like sections, having a greatly increased adhesive surface as explained in the Chapter on thermo-compressors, and a multi-jet performance is thus approximated.

Water-driven ejectors are built for capacities from 5000 to 30,000 lbs. vapour per hr. or more, and the cooling water ratio for condensing and air extraction combined is about 70 for, say, 26 in. vacuum and 86°F. injection water temperature. The water ratio for the condenser alone is about 30 and the waste water from the condenser leaves with a temperature close to that of the vapours.

The jet requires about 28 ft. manometric drop in head, thus the combined head above and below the orifice has to surpass this limit; generally 34 ft. is available. The distance *a* in the figure has to be less than the barometric head (normally not over 24 ft.) as otherwise the condenser will not aspirate the injection water. The head of the pump has to be in any case of such magnitude, that the ejector height is fully reached when there is no vacuum.

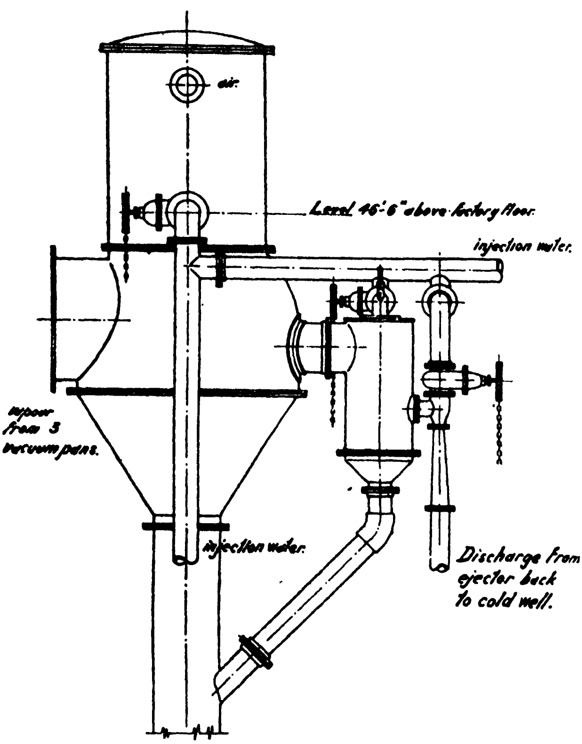


Fig. 505.—Arrangement for increasing Condenser Capacity.

A few typical installations, designed by the author, are given below. *Fig.* 505 shows an arrangement for increasing the capacity of an existing condenser, which actually had been choked, as the factory where it had been installed gradually increased its grinding rate from 800 to 1200 tons cane/24 hrs. The *auxiliary condenser* of 16 in. vapour inlet is bolted to the bowl of the existing main condenser for the vacuum pans. The injection water now is split up according to the approximate proportions :—

Main condenser	900 Imp. gals./min.
Auxiliary condenser	400 ,, ,,
Ejector	400 ,, ,,

The same injection and vacuum pumps remain in use, and an increase of 1½ in. vacuum has been obtained.

For priming the pans of a 2000-ton cane sugar factory, a booster vacuum pump had been used of approx. 330 cub. ft./min. displacement, but the large

1200 cub. ft. calandria pans took about 20 min. to attain 18 in. vacuum. In view of the fact that each lb. of moisture left in the pan after the steaming-out procedure would produce 130 cub. ft. of vapours at 24 in. vacuum, a cooler or condenser with air injector was installed between the priming main line alongside the pans and the auxiliary vacuum pump. The new arrangement has shortened the time to about 4–5 min. and raised the vacuum in that time to about 24 in. In *Fig.* 506 this arrangement is shown.

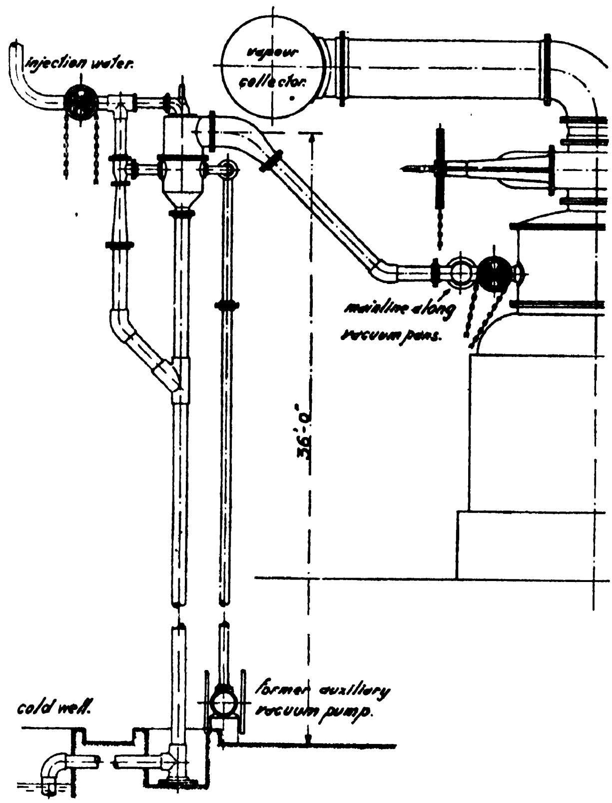


Fig. 506.—Booster Vacuum Equipment for priming Vacuum Pans.

To group the condensing and the air extraction in one apparatus of plain design, without moving parts, the *Multi-jet Condenser* has been designed and the newest construction is shown in *Fig.* 507. Several of these multi-jet condensers have been installed in cane sugar factories. In Chapter XXIII the advantages of the multi-jet process have been explained, and greater efficiency is obtained than is possible with a single jet.

In *Fig.* 507 the injection water is split into two branches, one to the water jacket *a* through the bronze nozzles *c*, whereas the rest goes through the spray inlet *b* to the multi-jet nozzle plate *d*, which discharges into the diffuser throat, finished on the inside for less friction and proper discharge of the water-air mixture.

The top spray undertakes the condensing of the vapours, whereas the lower jets accomplish the entrainment of the air, and thus the dry vacuum principle is approached in this recent design.

The water pressure at the nozzles needs to be only about 5 lbs./sq. in., the tail pipe suction doing the rest as in the previous cases mentioned. The temperature difference between injection and waste water lies normally between 5 and 10°F. and a good operating performance is assured. The Java Experimental Station recommends the use of multi-jet condensers in those instances where an ample water supply is available.[1]

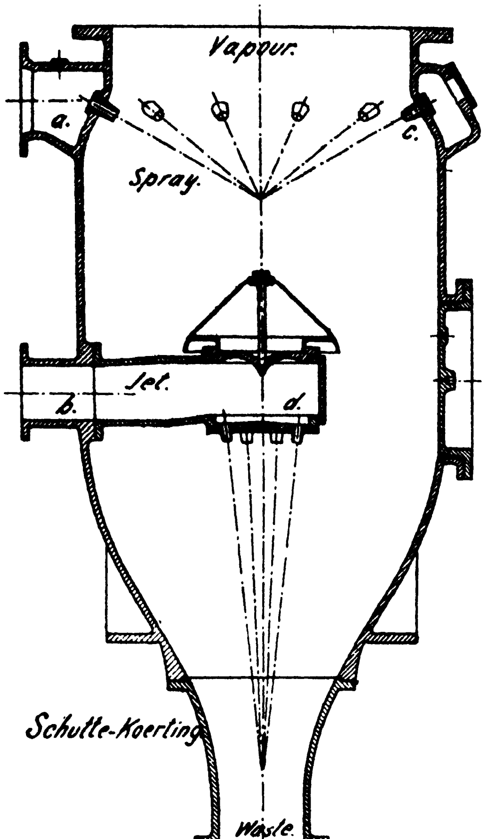


Fig. 507.—Multi-jet Condenser.

Another condensing equipment, in which condensing and air extraction is achieved in one apparatus, is the *Rotary Jet Condenser and Air Pump* of patent construction shown in *Fig.* 508, as installed in Australian cane sugar factories. This rotary condenser can be used equally for central or individual condensing and it is composed of the jet condenser proper at the right hand side of the Fig. and a dry air pump at the left hand. The waste water of the air pump is discharged into the suction well of the condenser.

The *modus operandi* is as follows: the injection water enters through the vacuum caused by the pump suction into the inside condenser impeller of bilateral centrifugal design *a* and is thrown out through an outlet or distribution ring into the diffuser space *b*, in which the vapours coming from the top are condensed. The waste water plus the entrapped air between the water jets is discharged through a guide wheel *d* into the inner housing *e* for discharge. The wet vacuum principle is applied in this right hand side of the rotary condenser.

Dry air extraction is achieved at the coolest spot close to the distribution ring before-mentioned and is guided by the channels *f* to the diffuser space *g* of the left hand bilateral impeller, it being discharged with the water jets through the guide wheel *h*.

The top outlet of the dry air pump (left hand part) is connected to the main priming line along the pans, so as to produce a vacuum when a pan is cut in. This applies, of course, when the rotary condenser is used as a central condenser.

The author has no data at hand as to power consumption, vacuum obtained or injection water ratio, but the design is of sufficient interest to be mentioned here.

[1] See the articles of G. J. SCHOTT, *Het Archief*, 1929, p.p. 772-784, and L. H. DE LANGEN, *Het Archief*, 1933, p.p. 555-567.

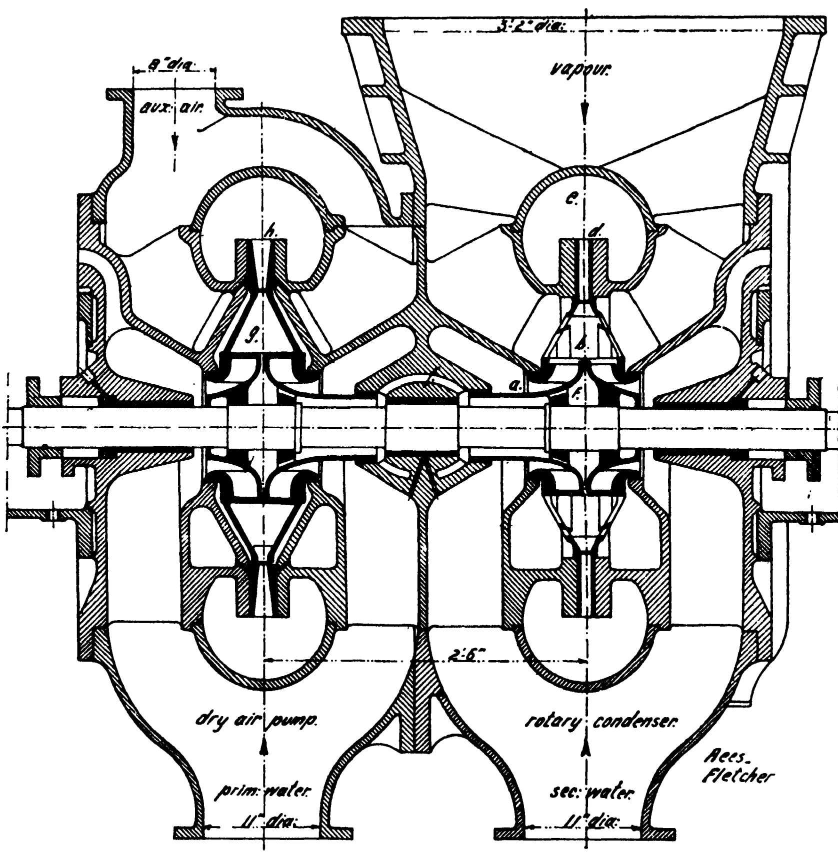


Fig. 508.—Rotary Jet Condenser and Air Pump.

4.—Spray and Cooling Ponds.

Not all cane sugar factories have the good fortune to be situated close to the sea shore or to a river having sufficient fresh water supply for the condensing performance of say 2·5 to 3 Imp. gals./min. for each ton of cane ground per 24 hours, and re-cooling of the waste water from the condensers has to be provided.

Pumping such large quantities of water from wells will be quite beyond economic limits in nearly all instances.

From the foregoing sections it has been learnt that the cooler the injection water, the smaller the amount required, and the author knows of reported data of 73°F. (23°C.), but general practice in Cuba and Mexico gives from 85° to 96°F. (30° to 36°C.) in the day time, the night temperatures being about 10°F. lower.

The back cooling of the waste water depends on heat absorption by the surrounding air, thus actually on the heating of this air above the prevailing temperature and for the larger part upon evaporation of a part of the waste water, both of which will be discussed below.

The *heat absorption* is derived simply from the following formula :—

$$W_{waste} \times (t_{wi} - t_{wii}) = W_{air} \times c \times (t_{ai} - t_{aii}) \ldots\ldots (133)$$

in which : W_{waste} = weight of waste water in lbs./min.
t_{wi} = temperature of waste water in °F.
t_{wii} = „ „ cooled waste water in °F.
W_{air} = weight of air required for cooling in lbs./min.
c = specific heat of air (0·2375 as an average).
t_{ai} = temperature of heated air in °F.
t_{aii} = „ „ surrounding air before heating in °F.

Let it now be assumed that $t_{wi} - t_{wii} = 22$°F. and $t_{ai} - t_{aii} = 16{\cdot}5$°F., then it follows that :—

$$W_{waste} \times 22 = W_{air} \times 0{\cdot}2375 \times 16{\cdot}5$$

or

$$W_{air} = 5{\cdot}61\ W_{waste} \ldots\ldots\ldots\ldots\ldots\ldots (133a)$$

and according to formula (84) this will mean that for each lb. of waste water about 79 cub. ft. of air of 100°F. average temperature will be required. These large quantities of air cannot be supplied by natural draft through a cooling tower or over a spray pond.

Evaporation, therefore, plays an outstanding rôle in cooling tower work, but the air will only absorb moisture when it is not yet saturated. At a higher temperature the percentage saturation will decrease, whereas at a lower temperature it will increase, until the saturation has reached its full value and condensation of the vapour in the air (which is called moisture) starts. This point is called the *dew point* and when the latter, which is measured by the wet bulb thermometer, coincides with the surrounding dry bulb temperature this indicates that 100 per cent. saturation of the air prevails. Generally, the dew point is below the dry bulb temperature, and for Cuba, of which country the author has extensive data, the saturation is 75 per cent. as an average, but even in those countries in the tropics where 100 per cent. saturation exists, the heating of the air through the waste water will provide the necessary drop in saturation.

From DALTON'S Law (formula 130) it will be seen that vapour plus air pressure forms the atmospheric pressure and it has been found that, e.g., at 86°F., 1000 cub. ft. atmospheric air (thus the mixture of air and vapour) at 100 per cent. saturation contains 1·899 lbs. moisture. From the psychrometric tables the following data are copied :—[1]

Moisture in Air at 100 *per cent. Saturation.*

Temp. °F.	Lbs. of moisture per 1000 cub. ft.
75	1·104
80	1·382
85	1·832
90	2·131
95	2·469
100	2·851
105	3·282
110	3·766
115	4·312
120	4·924

The author has taken, as a fair average, that at 15°F. air temperature rise, thus by heating through the waste water, one lb. of moisture can be taken up by 1000 cub. ft. of atmospheric air in saturated condition, and by taking 80 per

[1] See the psychrometric tables of the Wheeler Condenser and Eng. Co., of Carteret, N.J., U.S.A.

cent. efficiency of the absorption performance only 0·80 lbs. moisture will be taken up in the form of vapour. The latent heat of vapour at 100°F. average temperature is now 1036 B.Th.U./lb., and for 0·80 lbs. thus 828 B.Th.U.

Let it further be assumed that the waste water is cooled down 22°F., which is an overall practical figure, then it will be obvious that

$$W_{waste} \times 22 = V_{air} \times 828 \div 1000 \text{ in cub. ft.}$$

or

$$V_{air} = W_{waste} \times 26{\cdot}5 \text{ cub. ft.} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (134)$$

Each lb. of waste water to be cooled thus requires 26·5 cub. ft. of circulating air or one-third of the quantity required for air cooling, taking the most unfavourable case of 100 per cent. saturation of the non-heated air, which under practical conditions mostly met with will be excessive.

The absorption of the vapours by the air depends on the circulating air velocity alongside or through the cooling tower or over the spray-pond. In tropical countries there are generally trade winds or a breeze, and an average wind velocity of 2·65 ft./sec. can be adhered to, it being assumed that the air passes the cooling tower crosswise. The exposed area is thus the product of length and height; the formula can then be established as follows :—

$$A = \frac{V_{air}}{60 \times 2{\cdot}65} = \frac{W_{waste} \times 26{\cdot}5}{60 \times 2{\cdot}65} = \frac{W_{waste}}{6} \quad \ldots \quad (135)$$

in which : A = cooling tower area, i.e. length × height in sq. ft.
V_{air} = volume of air in cub. ft./min.
W_{waste} = weight of waste water in lbs./min.

Per Imp. gal./min. of waste water 1·6 sq. ft. cooling tower area is thus required. The author has on record cooling towers where only 0·6 sq. ft. had been allowed per Imp. gal. waste water/min., but it all depends on how effectively the air can accomplish its vapour-absorbing work.

Cooling towers for cane sugar factories are generally of the open type; chimney-coolers are not required for a factory in the open field. The height lies between 20 and 35 ft. and the width from 6 to 8 feet, with intermediate floors each 2 ft. apart. The frame of the cooling tower is made either of wood, constructional steel or concrete, whereas the floors are of lattice work. Structural steel towers have to be painted every crop and the cost of upkeep of the lattice floors is heavy.

From the above-mentioned calculation it will be seen that 22 ÷ 828 = 2·6 per cent. of the waste water will be evaporated. Moreover, the wind drifts about 3 per cent. away, and since the condensate from the vapours from evaporators and vacuum pans at a cooling water ratio of 50 is obviously 2 per cent., a make-up has to be provided of about 3 per cent. The cooling pond should contain 20 to 25 times the required volume per minute, and 30 feet on each side of the cooling tower will give protection for wind spraying. Enclosures or louvres should be avoided if possible as they may prevent the free circulation of the air. If more than one lane has to be built, an ample space of over 30 ft. should be provided between each. A cooling tower width of over 8 ft. is generally useless, as the air will be saturated before it has penetrated through the whole width.

The maintenance costs of a cooling tower are considerable, although its efficiency is higher than what spray cooling can achieve (about 15 per cent.); but the latter has so many advantages in respect to cleanliness and easy supervision, that it has replaced the cooling tower in many instances.

The effectiveness of the spray pond depends upon the sprayers used and a bronze *involute spray nozzle* (supplied through the author) is shown in ***Fig.* 509.** It has the advantage that it is practically unchokeable. These nozzles are generally grouped in clusters of five, having 2 in. pipe connexions, each 4 ft. long, to the "sweep tees" on the main supply line. These supply lines are constricted towards the end and a water flow of 9 ft./sec. is a fair maximum.

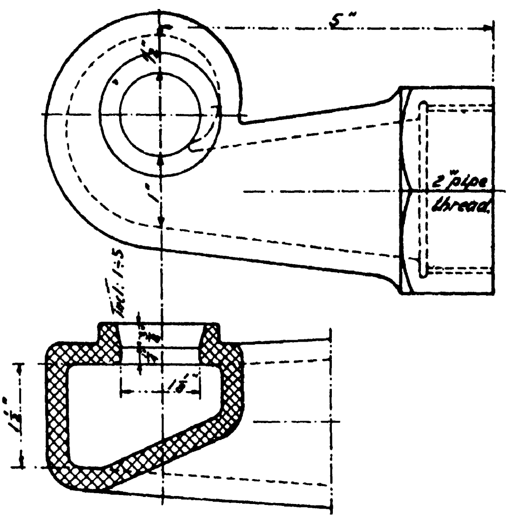


Fig. 509.—Bronze Involute Spray Nozzle.

A pressure of 7 lbs./sq. in. or 16 ft. hydrostatic head at the nozzle has been found to give the best operating result, and the quantity each nozzle will discharge at this pressure can be ascertained from the hydro-dynamic formula :—

$$V_n = \frac{A \times \sqrt{2gH} \times \eta \times 60 \times 12}{277} \text{ Imp. gals./min.} \quad \ldots (136)$$

in which : A = nozzle outlet area in sq. in.

g = 32·2 feet per sec². $\sqrt{2g}$ = 8·021.

H = hydrostatic head at nozzle (16 ft. average).

η = friction and contraction coefficient (0·5 average).

An Imp. gal. = 277 cub. inches, so these 2 in. nozzles will discharge about 40 Imp. gals. per minute. In U.S. 40 U.S. gals. is taken as standard and each cluster thus will discharge 200 gals. per minute. The cluster centres should be 13 ft. apart and 30 ft. from the cooling pond border or adjacent main line. Five square feet pond area per Imp. gal. waste water/min. gives a good average.

The main lines should be well supported on concrete piers above the water level in the pond.

CHAPTER XXVI.

RECIPROCATING PUMPS.

PRINCIPLES — PUMP DESIGNS — VACUUM PUMPS.

Several types of reciprocating pumps are to be found in a cane sugar factory and some have already been mentioned in the earlier chapters. The materials to be pumped are quite different: from the pure water used, say, for boiler feed ; they include sugar solutions up to 60° Brix (some of which contain a large amount of impurities, such as filter-press mud), air and gases like CO_2 to very viscous bodies like massecuite and molasses. The underlying principle of displacement applies for all these different pumps ; the liquids or semi-liquids have a fixed volume, whereas gases have a variable volume dependent on the prevailing pressure.

1.—Principles of Reciprocating Pumps.

The pumping instrument in reciprocating pumps consists of cylinders equipped with pistons, trunks or plungers, and the pump displacement per stroke will obviously be the product of piston area and the stroke. The pump can be single or double acting, and there will always be some leakage in the piston or the pump valves ; the *volumetric efficiency* therefore generally lies between 90 and 95 per cent. The pump output can thus be written down in the following formula :—

$$V_{min} = \frac{A \times L \times n \times i \times \eta_v}{277} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (137)$$

in which V_{min} = pump output in Imp. gals./min. per pump cylinder. (For U.S. gals. the divisor 277 becomes 231).

A = piston area in sq. in.

L = length of stroke in inches (single stroke).

n = number of double strokes per minute.

i = 1 for single acting, and 2 for double acting pumps.

η_v = volumetric efficiency.

The power consumption of a pump of a given output depends upon the manometric head, the specific gravity of the liquid to be pumped, and the *mechanical efficiency* of the pump plus the driving mechanism, this being generally between 0·85 and 0·95. The manometric head is the sum of the suction head, the discharge head and the pipe friction. Small velocities of flow, long bends and Y branches will reduce the latter. The maximum suction head for cold water will be about 26 ft., whereas for hot water the maximum has to be reduced by the equivalent head of the vapour pressure in lbs./sq. in. abs. corresponding to the prevailing temperature. Water at 150°F. has a corresponding vapour pressure of 3·714 lbs./sq. in. abs. and as each lb./sq. in. is equivalent to 2·42 ft. hydrostatic head, 8·98 ft. has to be deducted from the maximum of 26 ft. suction head. When the liquid temperature approaches the boiling point, the liquid has to flow by gravity to the pump, as the pump suction will be nil.

The power input of a pump amounts to :—

$$N = \frac{V_{min} \times 10 \times H \times S}{60 \times 550 \times \eta_m} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (138)$$

in which N = Power consumption in h.p. (b.h.p.).

H = Manometric head in feet.

S = Specific gravity of the liquid (for water 1, for syrup of 60° Brix, 1·29).

η_m = Mechanical efficiency.

The pump can be steam driven, by means of a gear, a belt, or, as recently, by trapezoidal or V-ropes.

The air pump volume is derived similarly as in (137) but the volumetric efficiency will be discussed in the corresponding sub-heading. The power input can be best derived from the mean effective pressure, taken from the indicator card from the air cylinder, in case the vacuum or air pump is not steam driven. If steam driven, only the steam cylinder has to be indicated. Formula (71)[1] applies to this case, but has to be divided by the mechanical efficiency. Wet air pumps have to remove a larger volume of air than is the case with dry air pumps, and moreover the waste water has to be pumped as well. When the waste water has to be delivered at a certain head above the pump, it should be remembered that the air has to be compressed at a pressure corresponding with this discharge head.

2.—Pump Designs.

The type of pump which has found an extensive field of application is the *Duplex Steam Pump*, shown in *Fig.* 510 ; it has an unequalled record for continuous operation under nearly all imaginable conditions. The author has seen these pumps at work in flooded pump pits and although heavy condensation losses occurred, the pumps succeeded in emptying the pits.

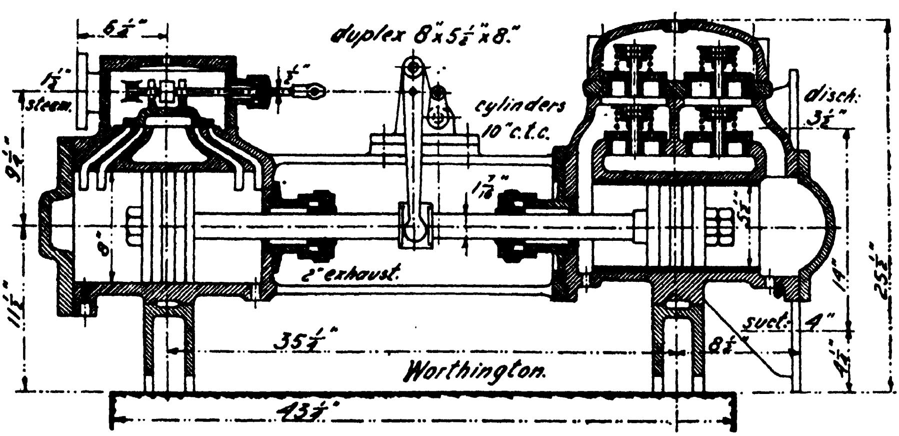


Fig. 510.—Duplex Steam Pump.

The general construction may be deemed well known ; in *Fig.* 510 is shown a longitudinal section of a duplex steam pump having two steam cylinders of 8 in. dia., two pump cylinders of 5½ in. dia., whereas both pumps have a stroke of 8 in. The size of this pump is customarily indicated by 8 in. × 5½ in. × 8 in.

[1] Page 241.

The steam cylinders have *double steam ports*, the outer ones for admission of live steam and the inner ones for exhaust. The steam valve is of the flat or D-type and there are no steam or exhaust laps. The operation of the valve is secured by means of a lever mechanism from the piston rod travel of the adjacent pump half, and a clearance of about 25 per cent. of the valve rod travel is provided between the lugs on top of the valve and the valve rod nut. The valve will thus move when the piston of the adjacent pump is approaching the end of its stroke. In case the valves of both pump halves are in mid-position, there will in practice be unequal leakage through the four cutting edges and the equilibrium of the steam pistons is upset; the pump thus will start under this condition.

The stroke of a duplex pump is not of a fixed value, as is the case with a flywheel pump. The compression or steam cushion is obtained by the closing of the exhaust ports by the piston, and knocking will thus be avoided. Cushion valves are sometimes furnished in the larger duplex pumps, providing a connexion between the steam and the exhaust ports so that the compression can be regulated at will.

The duplex pump belongs to the full-admission type of steam engine and a lower thermo-dynamic efficiency is obtained than with a flywheel pump, where expansion performance takes place. But it should be borne in mind that the speed of a duplex pump is regulated by a hand or pressure operated throttle valve and the throttled steam possesses a larger volume than when the full boiler pressure is acting, thereby reducing the steam consumption. A thermal efficiency of 0·25 to 0·35 (see page 242) is the general average and for small size pumps the lowest figure applies.

The relation between the diameters of the steam and the pump cylinders obviously depends upon the available minimum steam pressure and the maximum liquid pressure. A duplex pump having 6 in. pump dia. will require a steam cylinder of 7½ in. for 75 lbs./sq. in. water pressure, but 10 in. in case this water pressure amounts to 150 lbs./sq. in. Too large a steam cylinder will result in greater steam consumption and careful selection should be made when ordering a duplex pump.

In general the formula can be written :—

$$\frac{\pi D^2}{4} \times p_s = \frac{\pi d^2}{4} \times p_l \times y$$

or :

$$D^2 \times p_s = d^2 \times p_l \times y \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (139)$$

in which D = steam cylinder diameter in inches.
p_s = minimum available steam pressure at the pump in lbs./sq. in.
d = pump cylinder diameter in inches.
p_l = maximum liquid pressure at the pump in lbs./sq. in.
y = convenient safety factor.

From pumps in practical operation it has been found that a safety factor of 1·25 for large ones and 1·5 for small sized is the normal one, although a figure of from 2·5 to 3 has been found by the author in several instances. The volumetric efficiency of a duplex pump lies between 0·8 and 0·9 and formula (137) has to be applied with $i = 2$ and the result multiplied by 2 for the two pumps of each unit.

The valves of the pump are on removable seats and hand holes are provided for easy access. The valves proper are of brass or rubber, the former having a longer life, but they should be seated or ground in after each crop. Rubber valves should not be used for very hot liquids, as the rubber becomes soft and rapid wear will occur. Kinghorn valves, composed of three brass plates, about

$\frac{1}{16}$ in. thick, have also given good results, as they are very light and only small inertia forces prevail during the lift of the valves.

For juice, syrup and condensate of vapours, the pumps preferably should be made completely of bronze, as corrosive action is not illusory. The normal construction is bronze-lined with piston liners, valves, valve seats and the piston rod all made of bronze.

The piston speed of a duplex pump for thin liquids varies between 0·4 and 1·2 ft./sec. according to the size of the pump. For thick liquids about 0·6 of this speed is allowed and *ball valves*, which are suitable for this kind of liquid, are used ; 35 to 50 double strokes per minute are the normal limit.

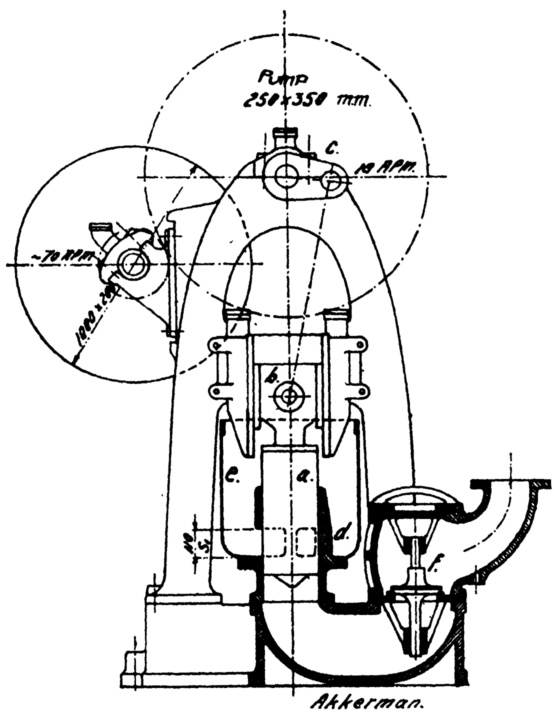


Fig. 511.—Vertical Massecuite Pump.

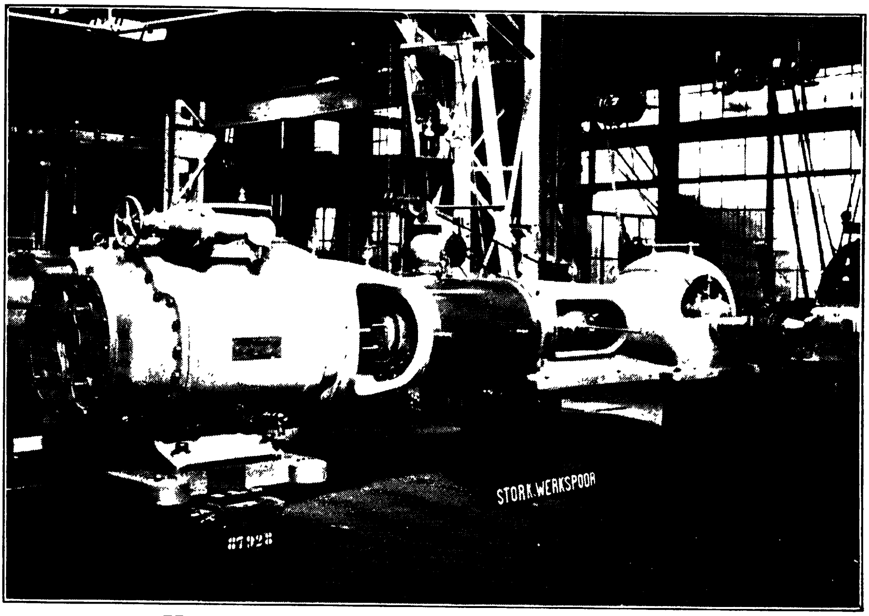
The liquid velocity in the pipe lines for reciprocating pumps should be kept low, as heavy inertia forces will prevail and the accelerating forces required for long columns are heavy and may have a pounding effect. Five to six ft. per second for small size pumps will give good operating results, whereas large capacities should not be handled over 3 ft./second, when large air vessels are not provided.

Air vessels on the suction as well as on the discharge side of a reciprocating pump will facilitate smooth running. The size of these vessels depends upon the length of the liquid columns to be accelerated. The suction air vessel has to be placed very close to the pump, having a volume of from 5 to 16 times the displacement of a single pump stroke. Pressure air vessels can be dimensioned according to the discharge line length and will be for duplex pumps :—

for	60 ft. length	4	times single stroke displacement.
for	300 ft. length	6	,, ,, ,,
for	3000 ft. length	12	,, ,, ,,

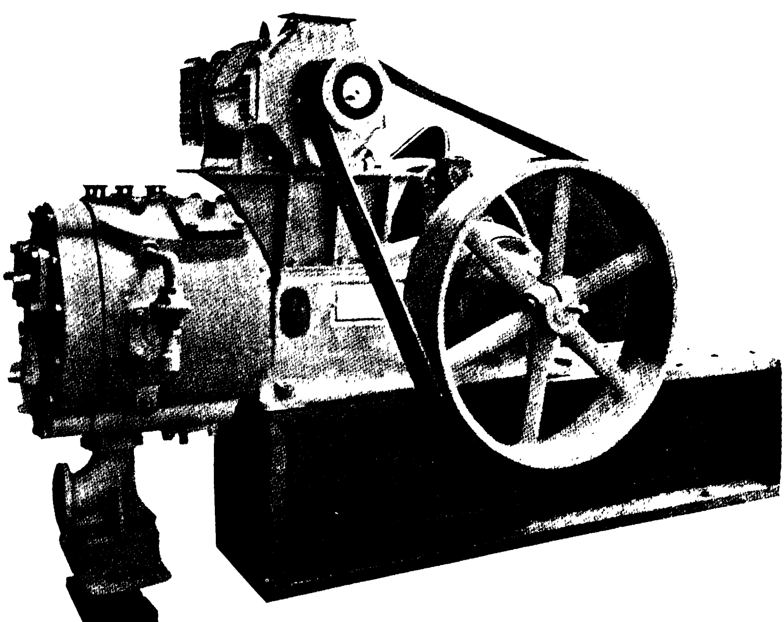
The capacity of the duplex pumps as used in sugar factories ranges between 6 and 3000 gals. per minute, and they are used for boiler feed, general service water, condensate, injection water, hot and cold juice, filter-press service, syrup and molasses ; the last three types are equipped with ball valves.

For very viscous materials like massecuite, the suction intake of a common pump will develop too much friction and the pump will act with only low volumetric efficiency. Specially designed pumps are therefore made and a *Vertical Massecuite Pump* of suitable design is shown in *Fig.* 511. The



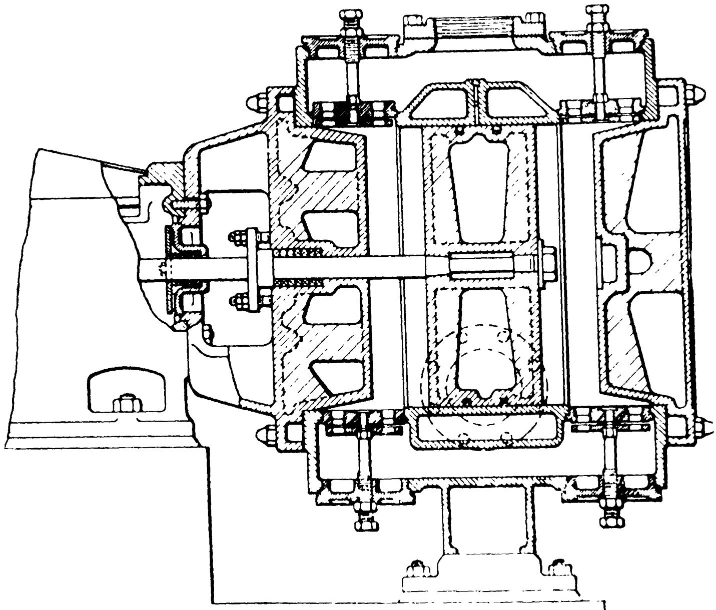
HORIZONTAL STEAM-DRIVEN DRY VACUUM PUMP, 550 × 1000 × 800 mm.

(*Gebr. Stork & Co.*)



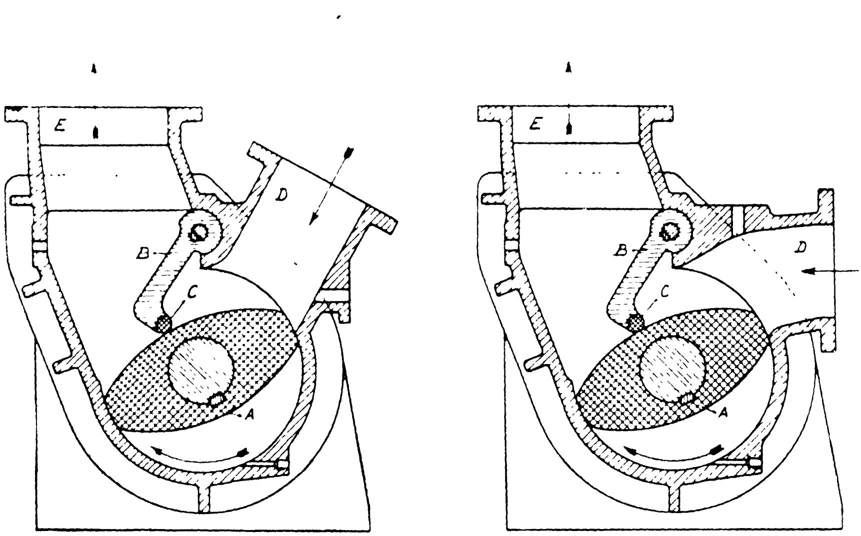
HORIZONTAL SIMPLEX POWER-DRIVEN DRY VACUUM PUMP WITH MOTOR AND "V" BELT DRIVE.

(*Worthington-Simpson, Ltd.*)



CROSS SECTION OF VACUUM PUMP CYLINDER.

(Ingersoll-Rand Co., Ltd.)



P. 82

SECTIONAL VIEWS OF ROTARY MASSECUITE PUMPS.

(Reineveld N.V.)

massecuite is charged into a trough or gutter e and with the upward stroke of the plunger a, which is guided by the crosshead b and driven by the crank c, the slotted openings in the plunger lining d will give free admission of the massecuite into the cylinder. With the downward stroke, as soon as the slotted openings are closed, the massecuite will be forced towards the discharge valve f. This valve is guided on both sides, so it cannot be diverted off its seat by the massecuite stream. Ball valves are also used for this kind of pump.

The output of this pump amounts to :—

$$V_{min} = \frac{A \times (L - S_1) \times n \times \eta_v}{277} \text{ Imp. gals. } \dots . \text{ (137a)}$$

The proportion $(L - S_1) \div L$ is generally taken as between 0·65 and 0·70, whereas the volumetric efficiency η_v will not be over 75 per cent. in practice.

The pumps are belt-driven ; the pump shaft, driven by a gear having a ratio 1 : 4, makes 18 r.p.m. About 14,000 lbs. massecuite will be discharged per hour with a single pump. Duplex arrangement is also quite practicable, as is also the horizontal arrangement.

Direct steam-driven massecuite pumps are also made, but the flywheel type is preferable, to avoid any hammering effect in the pump performance proper.

3.—Vacuum Pumps.

The *Wet Vacuum Pump* can be built according to the arrangement of *Fig.* 510 with a packed piston. The EDWARDS design shown in *Fig.* 444 and the flywheel type are both frequently used for smooth pumping performance.

The *Dry Vacuum Pump* will allow a higher piston speed than will pumps for liquids, and 8 ft. per second is now considered standard practice. The flywheel type, either steam or belt driven, is exclusively used ; but a high speed type is coming into favour for drives from high speed electric motors up to 1150 r.p.m. by means of belts or V-ropes. For the plain belt drive a belt tightener has to be provided.

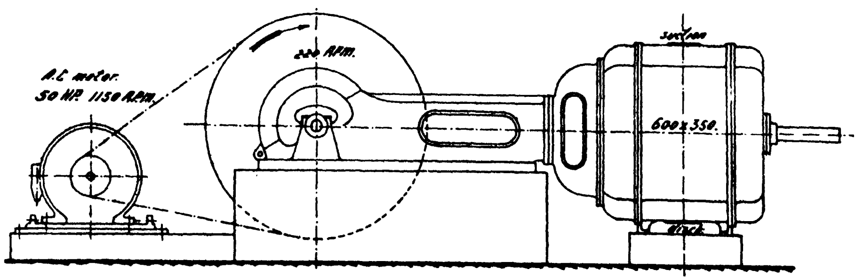


Fig. 512.—General Arrangement of a High Speed Dry Vacuum Pump.

In *Fig.* 512 is shown the general arrangement of a *High Speed Dry Vacuum Pump* of medium size, driven by a 50 h.p. A.C. motor running at 1150 r.p.m. The valves are located in the covers, but as this sometimes proves inconvenient in the front cover, where accessibility is handicapped by the frame connexions, it has led several designers to put the front valves laterally at the sides of the pump cylinder.

The pump is driven by *V-ropes* as shown in *Fig.* 513, this being a very reliable form of power transmission, from which the author has had favourable results. The belts will stretch a little at the outset and the driving motor should be arranged so that it can be adjusted backwards. After this initial stretching the belts do not require further tightening.

A first class rubber fabric is used in their manufacture and the makers deliver these belts in endless shape, so a fixed centre distance for given wheel and pulley diameters must be applied. Different sizes are made and the author has on record: size $\frac{2}{3}\frac{1}{2}$ in. × $\frac{7}{16}$ in. for power outputs from 2 to 25 h.p., and $\frac{7}{8}$ in. × $\frac{5}{8}$ in. for power outputs from 10 to 100 h.p The pitch length of the loops varies between 36·37 and 360 in. The average pull on each band for the smaller size is about 35 lbs. and for the larger one about 60 lbs. Speeds up to 64 ft. per second are allowable. The pitch diameter of the sheaves is one-fourth of an inch less than the outside diameter.

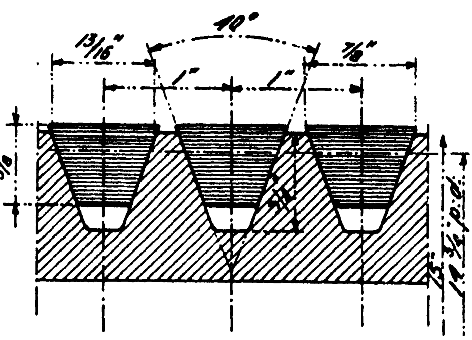


Fig. 513.—V-Belts.

The valves of the vacuum pump in *Fig.* 512 are *Feather Valves* as shown in *Fig.* 514. These are light and will last for a long time, re-grinding not being necessary. A buffer plate and four small brass springs make the valve operation silent. Air velocities from 60 to 80 ft. per second at the valve lips are allowable. The valve lift is between $\frac{1}{8}$ in. and $\frac{3}{8}$ in. and the design of *Fig.* 514 can be used for suction or discharge valve, two ledges being provided for this purpose on the outer valve seat ring. A joint of $\frac{1}{16}$ in. asbestos packing is laid under these ledges. The valve seat is held in place by means of a pressure bolt in the cover of the valve chamber. These bolts should not be tightened too heavily, as the author has known cases where valve seats have been broken through excessive tightening.

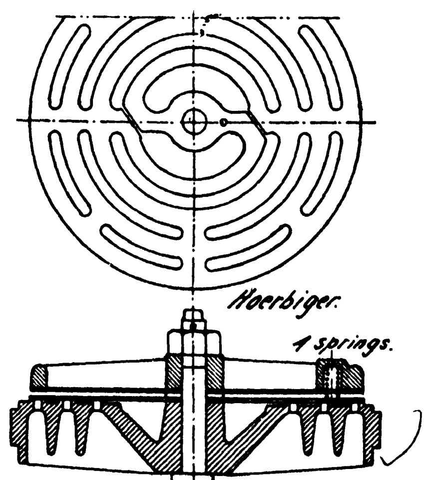


Fig. 514.—Feather Valve.

Feather valves are very reliable and of economic operation. A good lubricating oil of pure mineral composition should be used for vacuum cylinders. A poor quality oil will carbonize and may clog the ring-shaped valve openings.

In *Fig.* 515 is seen the diagram of the original dry vacuum pump. In this diagram *isothermic* compression is assumed, where all the produced heat is taken up by the cooling water. In practice, nevertheless, this will not be the case and the compression curve will lie between the isothermic and the adiabatic. The compressed air will not be released exactly at the atmospheric pressure, but a little above, so as to overcome the resistance caused by the expelling mechanism, in this case the discharge valve.

As soon as the piston is on the return stroke, there will be no suction before the air present in the dead space, *DS*, composed of the piston clearance, and the dead spaces of the valve seats or the ports leading thereto will have expanded until the condenser air pressure $p_{cond.}$ has been reached at point *a*.

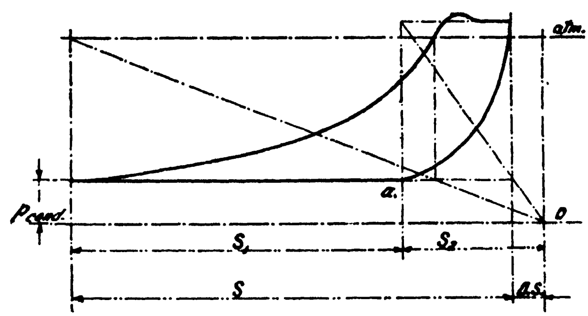


Fig. 515.—Diagram without Pressure Equalizing.

A part of the piston stroke, therefore, has no effect in respect to air suction, and the *volumetric efficiency* amounts to :—

$$Vol.\ eff. = \frac{S_1}{S}$$

With a dead space of 6 per cent. of the cylinder volume, a discharge pressure of 1·08 atm. abs. (16 lbs./sq. in. abs.) and a suction pressure of 0·1 atm. abs. (1·47 lbs./sq. in. abs.), the expansion of the air in the dead space will take place, when :—

$$0{\cdot}06 \times S \times 1{\cdot}08 = S_2 \times 0{\cdot}1$$

$$S_2 = 0{\cdot}65S,\ \text{and}\ S_1 = 1 - (0{\cdot}65 - 0{\cdot}06) = 0{\cdot}41S.$$

In these formulae the stroke *S* is taken as equivalent to the cylinder volume, as the cylinder diameter is a constant factor in all the cases of volume calculation.

It will also readily be understood that the volumetric efficiency with this kind of dry vacuum pump depends exclusively on *the dead space of the pump (air) cylinder.*

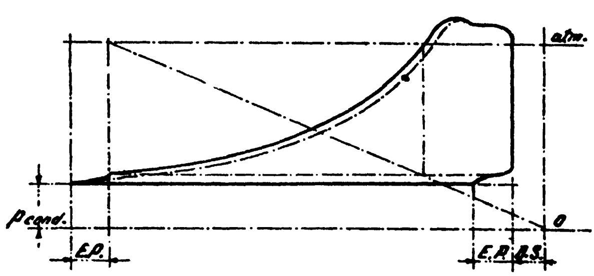


Fig. 516.—Diagram with Pressure Equalizing.

WEISZ added to the original construction by introducing an *equalizing valve* (automatically operated by the moving parts of the pump, and at the moment when the piston is in dead centre) to connect the compression end with the suction end; thus the condenser air pressure $p_{cond.}$ would be instantly reached and the volumetric efficiency brought up to about 100 per cent. In *Fig.* 516 a theoretical diagram of this performance is drawn and the equalizing of the pressure takes place at the distances EP of the stroke. It is obvious

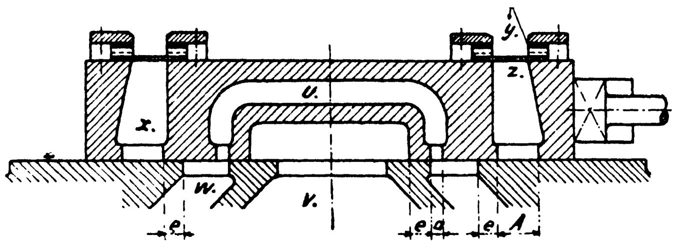


Fig. 517.—Equalizing Slide Valve.

that the *power consumption* of a dry vacuum pump with an equalizing valve will be higher, as the diagram area is increased.

In *Fig.* 517 such an equalizing slide valve is drawn. The ports w are connected to both cylinder ends and the suction is at v, so the atmospheric pressure will keep the slide valve on its seat. In mid-centre the port u connects both cylinder ends and as soon as the valve starts to move to the right, the left port a will close. When a distance e has been travelled, the port w is connected at the left side with the discharge port x and on the right side with the suction port v. The discharge ports x of the valve are covered by brass plates z, held on their seats by flat springs y. These plate valves are necessary, as the compressed air will only be expelled during the latter part of the stroke, and entrance of atmospheric air to the cylinder has to be prevented, as otherwise the power input will greatly increase.

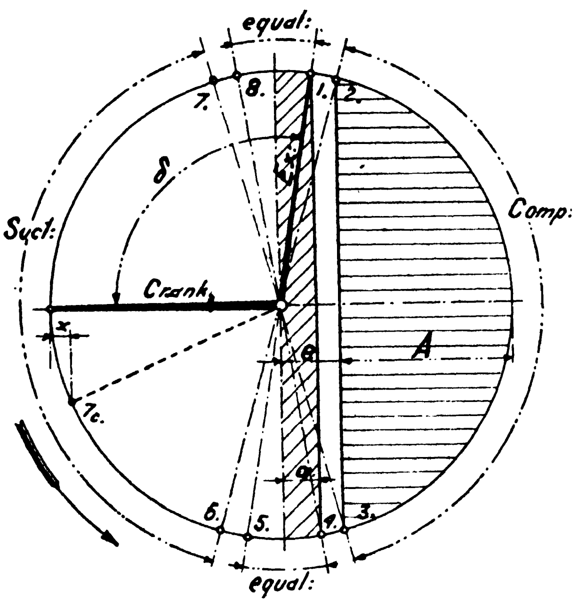


Fig. 518.—Equalizing Valve Diagram.

In *Fig.* 518 is shown the *equalizing valve diagram* and the eccentric Ex runs at the angle δ *behind* the crank. Assuming infinite rod length, at the left dead centre of the piston, the valve at 1 will start to open the equalizing port a. By further rotation this port is closed at point 8 and at point 7 the suction will start. The corresponding position of the crank at 7c gives only a small piston stroke x, which is equal to the distance EP in the diagram, *Fig.* 516, when small wire-drawing effects are not considered. At point 6 the valve has closed the suction port and opens at 5 the equalizing port, which is closed again at 4, whereas at 3 the discharge port x of *Fig.* 516 is opened and expulsion of compressed air takes place as soon as the compression has reached the limit above the atmospheric pressure. At point 2 the discharge port is cut off again and at 1 the cycle is repeated.

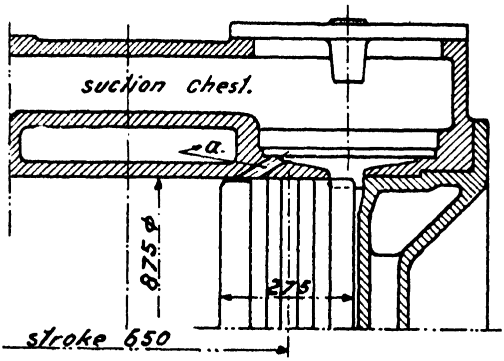


Fig. 519.—Equalizing through Piston.

To simplify the equalizing device, designers have tried to create a construction without the slide valve, as this valve needs lubrication and has to be absolutely air-tight, as otherwise a poor vacuum will be the result. Moreover, there is needed a power input to move or operate the equalizing slide valve.

In *Fig.* 519 is shown part of a dry vacuum pump cylinder, having a set of *feather valves* (HOERBIGER or ROGERS) which give a very reduced air resistance and have a very small lift from the valve seats. The suction chest is shown, although the feather valve and seat proper are left out. The original design of the pump, which had 875 mm. piston diameter and 650 mm. stroke, called for four holes a on each side, having a diameter of 30 mm. The equalizing is effected by the piston, as the holes are cleared when the piston is in dead centre and the pressure side is connected with the suction side. It follows from the drawing, that the equalizing had to take place through very limited passages, partly obstructed by the piston body protruding beyond the piston rings; so the number of holes was increased to eight on each side. This has given a somewhat better performance, but what was feared did happen—the holes were easily clogged by carbonized lubricating oil, the result of using an unsuitable lubricant; this deposit also caused scratching of the cylinder walls.

These drawbacks led to the entire omission of any equalizing device and the reduction of the dead space, as the latter is of paramount importance. The feather valves are located in the covers and a very small piston clearance of about 0·3 per cent. of the stroke is applied. An equalizing port a, as shown in *Fig.* 520, can be arranged, but generally this is not present in modern dry vacuum pumps. The pump cylinder has a water-jacket around the piston course and the covers should also be cooled, but the available space is small, as the feather valves have to be of the biggest size possible.

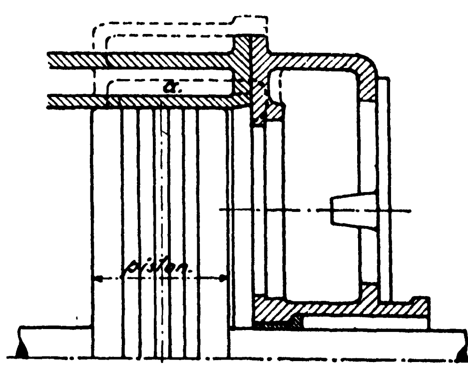


Fig. 520.—Non-equalizing Air Cylinder.

There are two kinds of dry vacuum pump of different types, which are widely used in sugar factories, one having a long stroke for a reduced number of revolutions and the other a short stroke for higher speed. In *Fig.* 521 both principles are shown and in both cases the clearance of the piston is reduced to the smallest practical limits of about $\frac{3}{32}$ in. It will be readily seen that the long stroke vacuum pump has the advantage, as the total dead space is about 0·5 per cent., whereas the short stroke pump with the same piston clearance will have about 1·2 per cent. The short stroke pump at the lower part of *Fig.* 521 has revolving or swinging intake valves, similar to Corliss construction. These valves have been replaced also by feather valves as the latter do not need lubrication and keep air-tight better. In respect to first costs, the high-speed vacuum pump is cheaper than the long stroke vacuum pump for the same volumetric displacement.

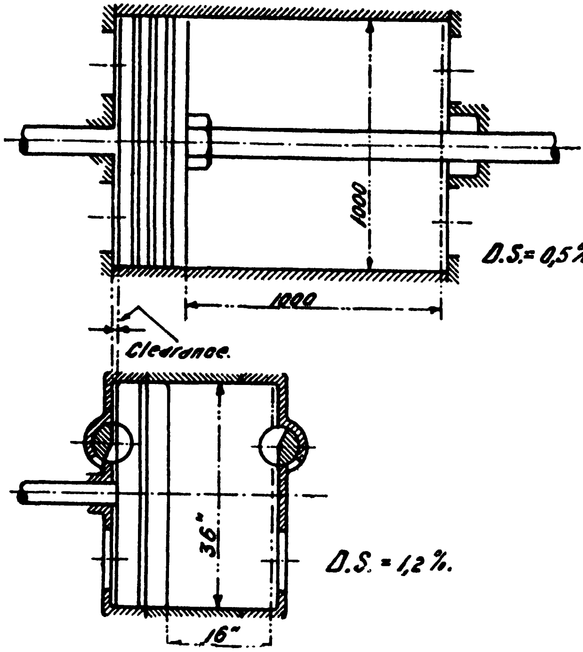


Fig. 521.—Long and Short Stroke Vacuum Pump.

The writer had occasion at the same sugar factory to take indicator diagrams of an equalizing and a non-equalizing vacuum pump, both having tandem arrangement of the cylinders and both equipped with piston valves on the steam end. The equalizing pump had a flat slide valve as shown in *Fig.* 517 and the non-equalizing pump was of modern design with feather valves in the cylinder covers.

In *Fig.* 522 the steam diagrams are shown on the right side, while on the left side are diagrams of the vacuum pump with an equalizing valve. The

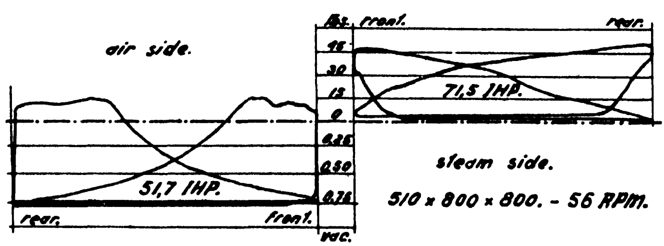


Fig. 522.—Diagrams of Equalizing Vacuum Pump.

indicator cards were taken when there was no juice in the house, and this explains the low vacuum. The back pressure on the steam side was also lower than under factory operating conditions, as the exhaust was blown off through the roof. It is seen that the *resistance of the equalizing valve* is considerably higher than when feather valves are used. The equalizing effect is nearly perfect.

The power input, as taken from the average of several diagrams, was 71·5 i.h.p., whereas the output of power was 51·7 i.h.p. which gave a *total mechanical efficiency of about* 72 *per cent.*

In *Fig.* 523 are shown identical diagrams, taken from the non-equalizing pump. As both vacuum pumps are connected to the same condenser and one is of larger size than the other, some fluctuations occurred when one of the pumps was slowed down for attaching the indicator cord and tests were made to let both pumps run at the same speed. Therefore maximum and minimum diagrams are shown on both steam and air ends. It is seen that the discharge resistance of the feather valves is very small and the volumetric efficiency very high. As the cooling of the air cylinder was not sufficient through lack of water, the compression lines of front and rear air diagrams

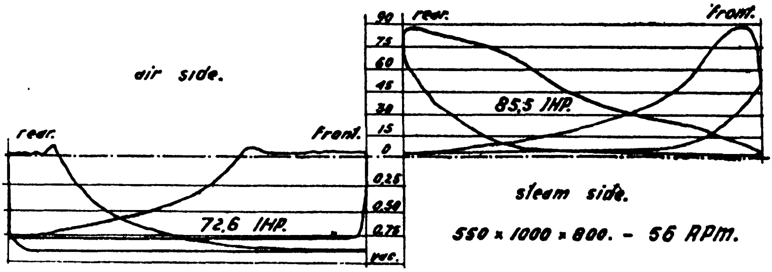


Fig. 523.—Diagrams of Non-equalizing Vacuum Pump.

were different, the front one being more isothermic and the rear one more adiabatic. From several diagrams taken a fair mean could be established and the power input, derived therefrom, amounted to 85·5 i.h.p. The power output was 72·6 i.h.p., which gives an overall *mechanical efficiency of* 85 *per cent.*, considerably in excess of that of the equalizing vacuum pump.

The diagrams were taken with one indicator, but the vacuum was watched closely, so as to obtain results as closely true as possible; and the difference between the two types of dry vacuum pump may be considered as well established.

Air Compressors are also used in cane sugar factories for supplying compressed air to sulphur furnaces, agitating milk-of-lime by means of perforated tubes, emptying closed crystallizers, for filter-presses and for shop work. The construction is similar to that of the dry vacuum pump and the underlying theoretical principles are to be found in any engineering handbook.

Water cooling of the cylinders and when possible the covers is essential for vacuum pumps and compressors.

CHAPTER XXVII.

CENTRIFUGAL AND ROTARY PUMPS.

PRINCIPLES OF CENTRIFUGAL PUMPS — DESIGNS — ROTARY DISPLACEMENT PUMPS.

A similar development as from reciprocating to rotary prime movers has also taken place with pumping equipment, and a further advantage is to be found in the higher speeds at which these rotary pumps can be run, making them especially fit for direct-coupling to electric motors or steam turbines.

Those pumps in which kinetic energy or centrifugal force is used as the propelling agent are called centrifugal pumps, whereas the other types are based on the displacement principle and are called rotary displacement pumps.

Centrifugal as well as other rotary pumps are of very ingenious construction, requiring small floor space for large capacities. The absence of valves also makes these pumps cheap as compared with piston pumps and for medium and large capacities, the centrifugal types have now replaced to a big extent the reciprocating piston ones.

1.—Principles of Centrifugal Pumps.

In the centrifugal pump, the liquid to be pumped enters an impeller, which revolves at a high speed so as to accelerate the speed of flow, and the kinetic energy thus created is later on transformed in the outer pump casing into potential or pressure energy.

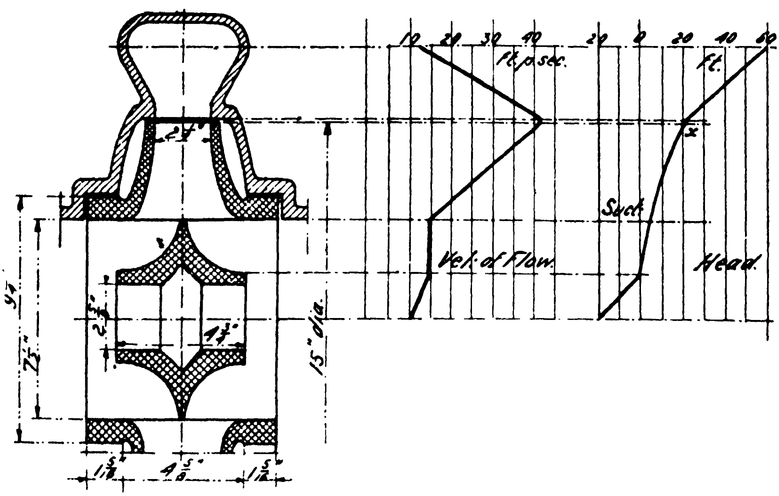


Fig 524.—Impeller of a Centrifugal Pump.

In *Fig.* 524 is shown the *Impeller of a Centrifugal Pump*, having bilateral entrance; the impeller is thus equilibrated against lateral or axial thrust, as the suction force acts equally on both sides. At the right side of the figure is shown first the *velocity diagram*, from which it will be seen that the liquid enters the impeller with a suction velocity of about 10 ft. per sec., this being raised to about 42 ft./sec. at the periphery and then drops in the pump casing until the discharge velocity is about 12 ft./sec.

Next is shown a *pressure diagram* at the extreme right; here at the pump entrance there is a vacuum of about 20 ft., necessary for the pump suction. In the impeller the pressure gradually rises until x at the periphery, and this pressure is called the split pressure, as the water will be pressed between the split joint which is the opening between the impeller and the pump casing. At the entrance of the impeller there are also split joints, which are kept as small as smooth running will allow without friction. A small loss of about 5 per cent. of the pump output is necessarily occasioned by these openings. In the involute or turbine casing of the pump, the pressure increases further until a discharge pressure of 60 ft. head is reached and the total lift of the pump will thus be 80 ft.

The centrifugal pump cannot be primed by itself, as a sufficient vacuum cannot be produced to obtain a suction lift of the liquid to be pumped. The pump thus has to be completely filled before it is started and this can be done by means of a steam-ejector, a vacuum connexion to the condenser or a foot-valve at the bottom end of the suction line; the latter should be vertical and only possess one bend or elbow. Air pockets or air entrances in the pump-suction may cause the pump to fail and will impair continuous operation.

Self-priming Centrifugal Pumps are now on the market, which are provided with a small rotary vacuum pump on the pump shaft to evacuate the pump casing, and consuming about 1 h.p. In all those instances where failure of the suction performance is possible, these self-priming centrifugal pumps are to be recommended.

Every centrifugal pump has to be provided with a discharge valve, as medium and large size centrifugal pumps, contrary to the displacement pumps, have to be started with the discharge valve *closed*. This is done to avoid water-hammer through excessive output caused by the lack of a manometric head, which not only may cause the pump to fail, but also may burst the pump casing, as has happened ere now.

The usual working of a centrifugal pump can be learnt from *Fig.* 525. The *absolute entrance velocity* V_1 of the liquid is assumed in radial direction, as is the case in most instances. The *tangential velocity* U_1 of the impeller at the diameter D_1 combined with V_1 gives the resultant *relative liquid velocity* V_r and the impeller blades or vanes have to be curved according to this velocity V_r for the proper entrance of the liquid into the impeller. The passages between the blades widen towards the outer periphery, so the relative liquid velocity decreases until V_{r2}. This latter velocity combined with the tangential velocity U_2 of the impeller at the diameter D_2 gives the resultant absolute discharge velocity V_2, which can be split into a radial velocity F and a tangential liquid velocity W.

The velocities V_r and V_{r2} obviously depend upon the impeller area at D_1 and D_2 and the volume of liquid they have to pass per minute, thus upon the pump output. Moreover, $U_2 = U_1 \times D_2 \div D_1$.

The head or manometric height to which the liquid has to be pumped depends exclusively on the tangential component W, and as the pump output will vary with the head, which is, e.g., possible with injection pumps pumping water into a barometric condenser with varying vacuum, it will be seen that V_r and V_{r2} are liable to variation. In *Fig.* 525, therefore, is shown by dotted lines an increase in the velocity V_{r2} and thus the resultant velocity V_3 will get a different angularity. For the entrance diagram a similar performance takes place and the entrance blade curve thus is only for a given pump output and any variation from this will give a reduced efficiency through *cavitation*, and corrosion of the blades is the result. When the corrosion shows itself on the top side of the blades an increased output has been the cause or if on the bottom side a reduced output.[1]

Centrifugal pumps, having guide vanes around the outer periphery of the impeller, will show similar effects with varying pump output.

If only one impeller is in use, the pump is called a *single stage*, low pressure or low lift centrifugal pump; this is the most common type in cane sugar factories. These pumps are built for heads up to 150 ft. without guide vanes and up to 350 ft. with these guide vanes.

Multi-stage, high pressure or high lift centrifugal pumps have more than one impeller, arranged in such a way that the discharge from the first impeller is directed by a special return guide wheel to the suction of the next one. The total head of the pump thus will be the algebraic sum of the individual heads of the impellers. For high pressure pumps, high speeds are essential and direct coupling to alternating current motors with respectively 3000 and 3600 r.p.m. for frequencies of 50 and 60, are now made for boiler feed, and water supply under pressure for operating hydraulic centrifugals.

The pump casing of the low lift centrifugal pump has an increasing area towards the discharge end and on account of its construction is called an *involute pump*. H.P pumps, having guide vanes with radial discharge, have a circular pump casing and therefore are called *turbine pumps*.

The fundamental equation for centrifugal pumps is taken from the theory of water turbines, as a centrifugal pump *de facto* is an inverted water turbine. This formula reads :—

$$E \times H = \frac{V_r^2 - V_{r2}^2 + U_2^2 - U_1^2 + V_2^2 - V_1^2}{2\,g} \quad .. (140)$$

in which : E = total efficiency of the pump.

H = measured head in feet.

V_r and V_{r2} = relative velocities of liquid to impeller at D_1 and D_2 in ft./sec.

U_1 and U_2 = tangential velocities of impeller at D_1 and D_2 in ft./sec.

V_1 and V_2 = absolute velocities of liquid at D_1 and D_2 in ft./sec.

g = gravity acceleration 32·2 ft./sec.^{2}

The total pump efficiencies for involute pumps can be taken as :—

From	75 to	250	Imp. gals.	per min.		55 to 65 per cent.
From	250 to	1000	,,	,,		70 per cent.
From	1000 to	3000	,,	,,		70 to 73 per cent.
From	3000 to	6000	,,	,,		73 to 75 per cent.
From	6000 to	10000	,,	,,		75 to 80 per cent.

The suction of a centrifugal pump should not be taken as over 24 ft. at sea level and with cold water. A foot valve in the suction line should be provided for high suction lifts.

The *power consumption* of a centrifugal pump can be derived from formula (138), but instead of η_m, the efficiency E from formula (140) should be applied, as well as the measured head H.

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[1] For the design of centrifugal pumps see: FRITZ NEUMANN, Die Zentrifugalpumpen, JULIUS SPRINGER, Berlin; or CARL DE LAVAL, Centrifugal Pumping Machinery, McGraw-Hill Co., New York and London.

For a given output, the centrifugal pump will have a somewhat larger power consumption than a reciprocating piston pump.

The *characteristics* of the centrifugal pump are very interesting and any order for a pump should be based on them. Manufacturers supply these characteristic curves for every pump type they make, and the range for which such a pump can be efficiently used is easily learnt. In *Fig.* 526 the characteristic curves for the same pump at different speeds are shown, although in sugar factories centrifugal pumps with generally constant speed are the most prevalent.

The *pump head,* shown at the top of the diagram, is only 50 per cent. of the normal for 100 per cent. output at 800 r.p.m., whereas at 1200 r.p.m. it is 120 per cent. of normal. This indicates that the pump head is approximately proportionate to the square of the speed.

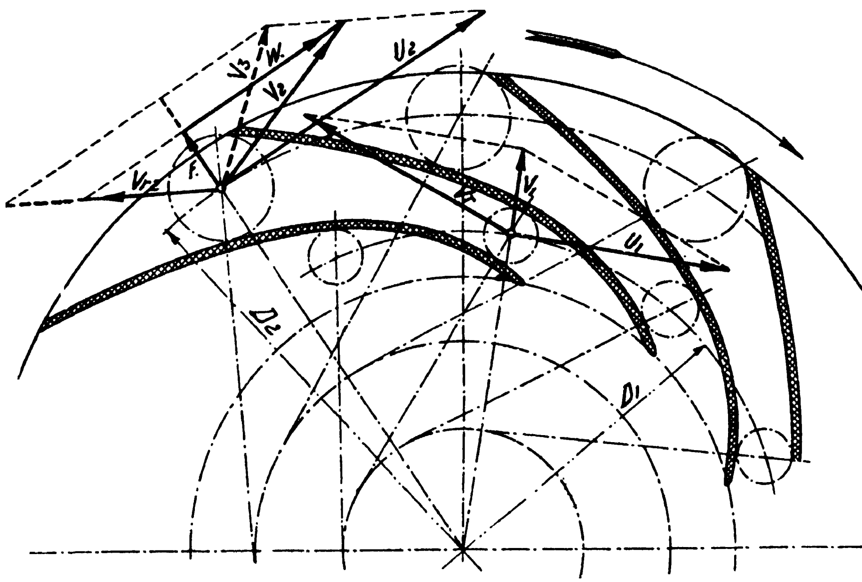


Fig. 525.—Centrifugal Pump Performance.

The *power consumption* in the middle of the diagram gives about 28 per cent. at 800 r.p.m. against 87 per cent. at 1200 r.p.m. or nearly proportionate to the third power of the speed.

There is a peculiarity to be reckoned with, as with reduced head, not corresponding with the normal speed for which the pump has been designed, the pump output and the power consumption will increase, as shown on the right hand side of the upper characteristics, and in case of electrically-driven pumps, an over-size motor may be required, which is less desirable for alternating current. For electric-driven pumps a *flat* power characteristic is of great value for obtaining a good overall power factor, when varying head prevails.

The *efficiency* of the pump, in *Fig.* 526 at the bottom, is good between 80 and 160 per cent. of the normal output of 1000 Imp. gals. per minute.

For constant speed the three characteristics, head, power consumption and efficiency are generally arranged in one diagram. But it should be remembered that the characteristic of efficiency is for the pump alone, and

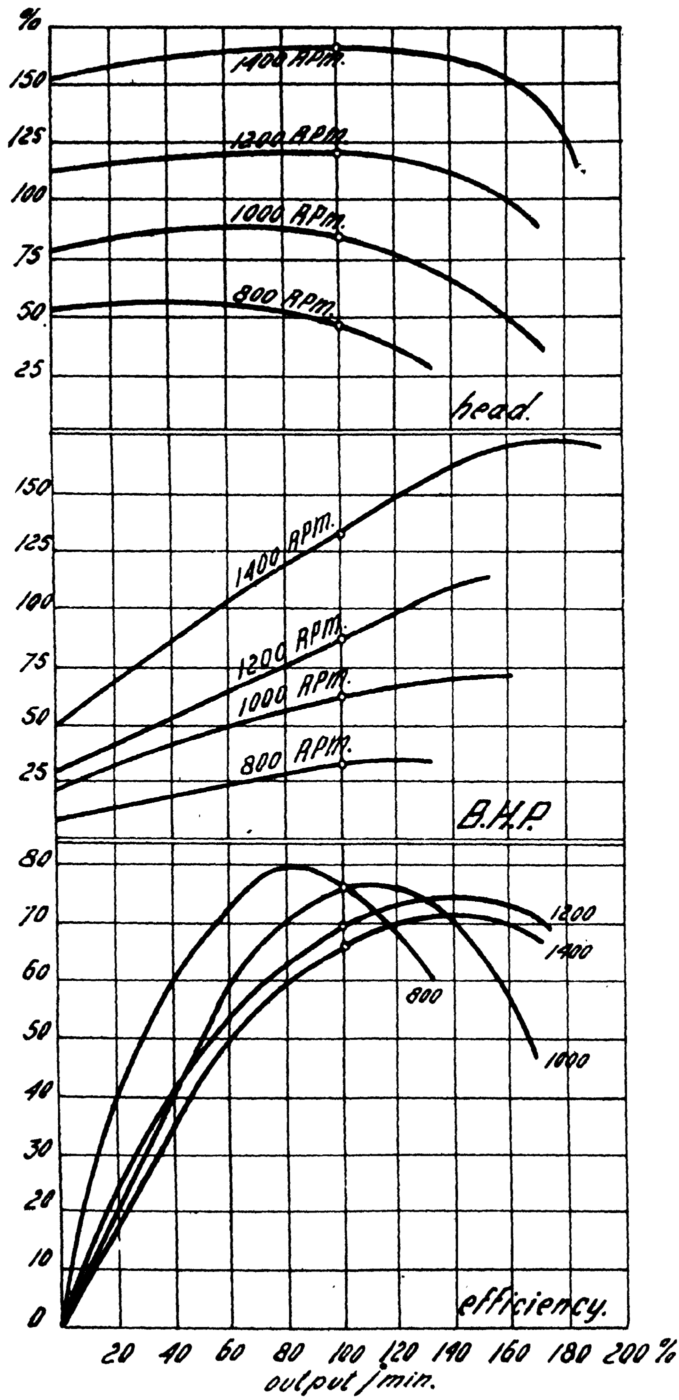


Fig. 526.—Centrifugal Pump Characteristics.

pipelines should be of ample size, generally connected by tapered reduction pieces to the pump intake and discharge, so as to have low pipe friction through a low velocity of flow.

In this Chapter should also be mentioned the *propeller* or *screw pumps*, sometimes also called helicoidal pumps. They do not belong to the centrifugal pumps proper, although kinetic energy is produced to overcome the head. The origin of the propeller pump is to be sought in the fact that for irrigation and other purposes a low head pump was required and the centrifugal principle thus necessitated small impeller diameters with short and inefficient blading, or low speeds had to be used, requiring costly electric motors. Propeller pumps are made for speeds between 1200 and 1800 r.p.m. or above, and the principle is that of a marine propeller, but with a higher fullness or discharge coefficient. For heads up to 30 ft. these pumps have a fair efficiency of about 0·66 or above and are low in cost. For irrigation purposes with lifts of 6 to 10 feet, excellent operating results have been reported.

2.—Centrifugal Pump Types.

An electrically-driven *Involute Centrifugal Pump* for raw juice in actual operation is shown in *Fig.* 527. The motor and the pump are arranged on a cast iron bedplate, which makes alignment easier, but care should be taken that the bedplate is not spanned by the foundation bolts, when not properly grouted. The pump casing is split on the horizontal centre line, the interior of the pump being thus easily accessible by lifting the pump cover.

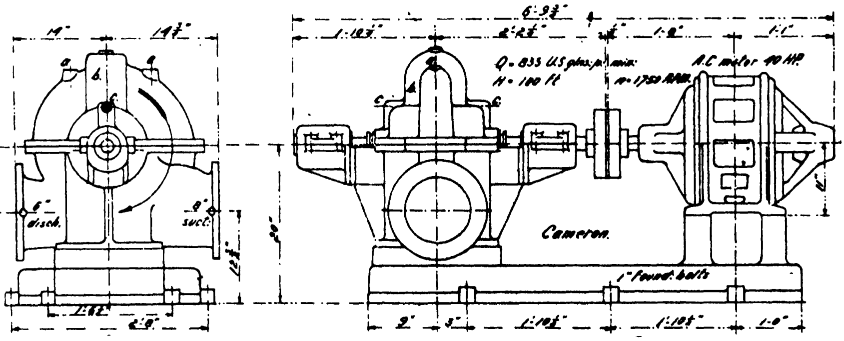


Fig. 527.—Involute Centrifugal Pump.

The coupling is of the flexible type, but it should not be overlooked that anything beyond micrometric misalignment will destroy the coupling or cause hot bearings. At *a* eye-bolts are inserted for lifting the cover ; *b* is an equalizing passage, to make sure that there does not exist any difference in pressure on both sides of the impeller, which is of bilateral type. Axial thrust is thus avoided and special thrust bearings on the pump shaft are not required. At *c* two angle valves with built-in seats are provided for the lantern bushes in the packing boxes for the so-called water seal.

In the left hand part of the Fig. a side view of the same pump is shown; the suction and discharge connexions are in the lower pump half and have not to be dismantled for inspection of the impeller.

In *Fig.* 528 is given the cross-section of an *Involute Injection Pump*, having also an impeller with bilateral entrance and a split pump casing. The valves *a* are on the pipes connecting the pressure side of the pump to the lantern bushes in the packing boxes. These packing boxes should be tightened slightly and some water should flow to the outside, and be collected in the bearing supports, which are used as drip pans. The water seal is of great importance for the uninterrupted suction performance of the pump.

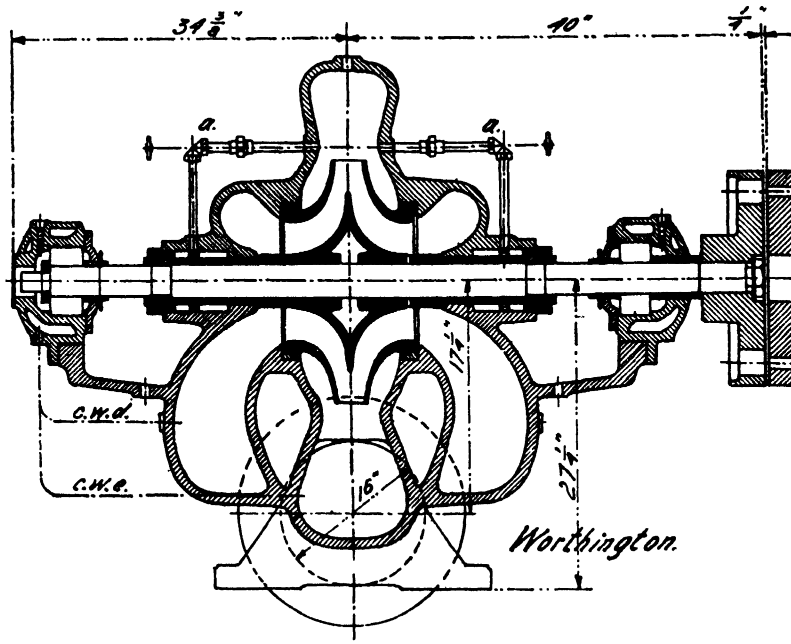


Fig. 528.—Involute Injection Pump.

Bronze sleeves are mounted over the pump shaft, which can be replaced when worn at the loci of the packing. A good soft and non-abrasive packing should be used.

The author has seen unprotected pump shafts which had corroded close to the impeller, to nearly half the original diameter within a few crops, due to

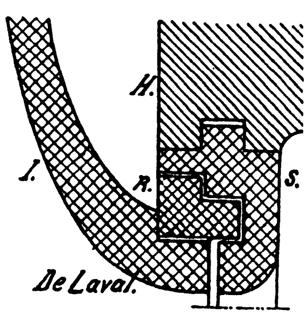


Fig. 529.—Improved Design of Seal Ring for L.P. Pump.

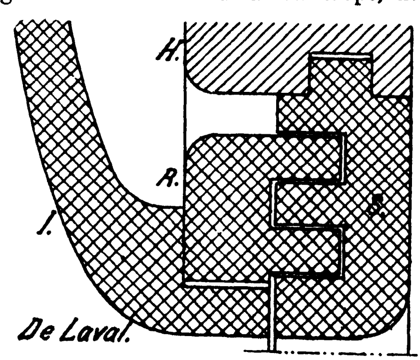


Fig. 530.—Labyrinth Construction for a High Pressure Pump.

acidity of the injection water. This acidity sometimes happens when a spray pond is used, the same water being thus circulated in the same cycle again, and acidity will increase if the catch-alls allow some sugar to be drawn over into the condensers.

The pump shaft is supported in double row ball bearings and these latter are water-cooled. The cooling water enters at *c.w.e.* from the pump discharge and is returned by *c.w.d.* to the pump suction. A regulating valve for throttling the rate of flow is provided for each bearing.

The coupling is of rubber bushing and steel pin construction, but other types of flexible couplings are used as well.

Impeller Bushings or Seal Rings have received special attention from several manufacturers, and the higher the pressure difference between suction and discharge, the more water or liquid will flow back to the suction side, reducing the pump efficiency. In *Fig.* 529 is shown an improved design for a low pressure or involute pump, whereas in *Fig.* 530 a labyrinth construction is shown for a high pressure pump. In case of split housings H, the seal rings S are in one piece, firmly held in a groove in the housing; its rotation is prevented by a pin. Interchangeable rings R are threaded on the impellers I.

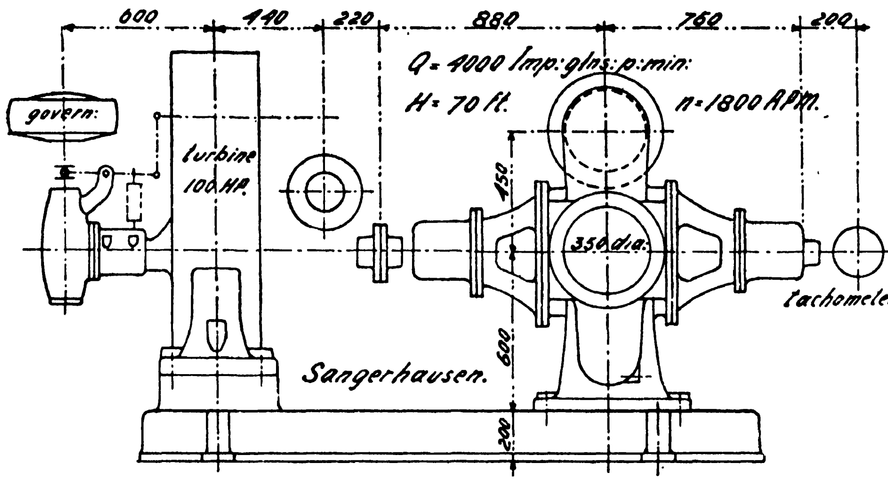


Fig. 531.—Turbine Driven Injection Pump.

In *Fig.* 531 is shown a *Turbine-driven Injection Pump* which is in actual operation. The impeller is of bilateral design, and the pump and turbine shafts are supported in ball-bearings. The pump has a Francis impeller of small diameter and the pump casing of the involute type is of reduced dimensions. Since the pump casing is not split, the impeller has to be withdrawn laterally by removing the off-side cover.

In *Fig.* 532 is shown the side elevation of the turbine, which is of the direct impulse type. The steam enters at the steam separator, *S.S.*, and then passes the main steam valve, *M.S.V.* Before entering the steam chest it has to pass the quick-closing or emergency valve, *E.S.V.* The governor acts on a throttling valve built into the *M.S.V.* housing and an emergency governor for excess speed is also provided. The turbine has four nozzles, I - II - III - IV, but with 100 lbs. steam pressure only two nozzles are required for 100 h.p. output.

This turbine has shown good operating results and a fair steam consumption. The speed is normally 1800 r.p.m., but the outfit has run at over 2000 r.p.m. for a larger pump output, which was possible by attaching additional weights to the governor lever.

Centrifugal pumps with *open impellers* are used for gritty liquids, like milk-of-lime and filter-press mud. Wear on these impellers is heavy, but they can be easily cleaned. White metal open impellers for lime-milk centrifugal pumps have given good operating performance. The pump has to be

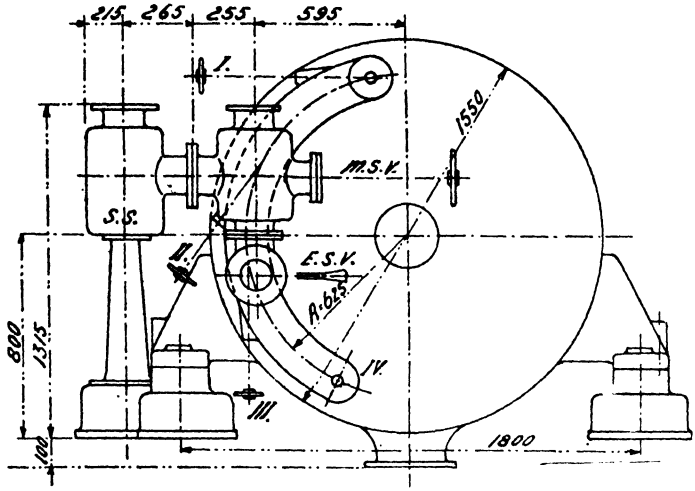


Fig. 532.—Elevation of the Turbine.

cleaned after each stoppage, so as to avoid the possibility that the lime will settle in the lower part of the pump body, which would result in the breakage of the blades when starting.

3.—Rotary Displacement Pumps.

For pumping injection water several of the well known ROOT'S drum pumps are still to be found at a few cane sugar factories. They are direct-connected to a steam engine or to a geared turbine or electric motor. But in most instances they have been replaced ere now by centrifugal pumps.

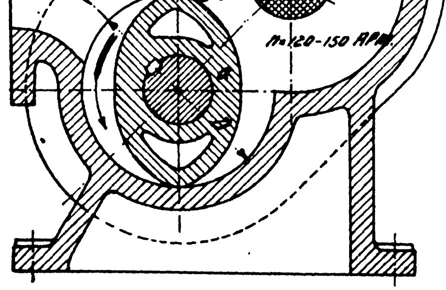
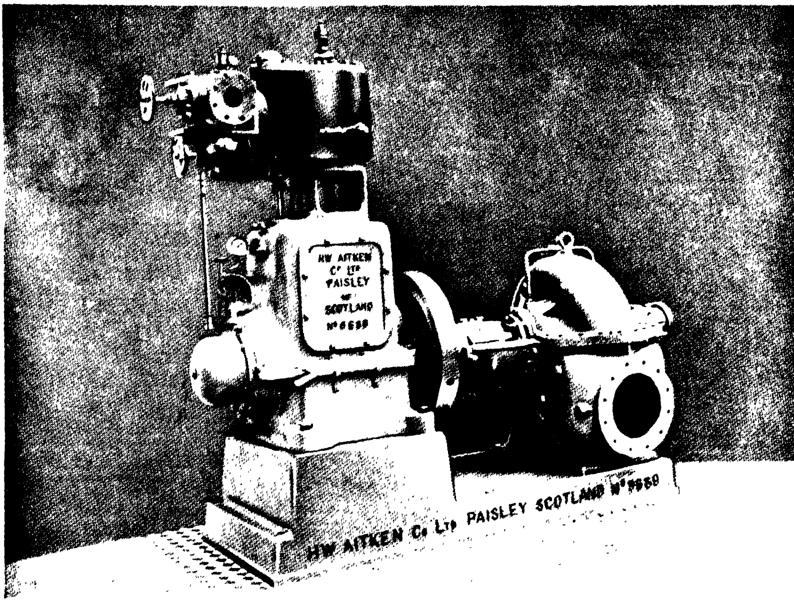


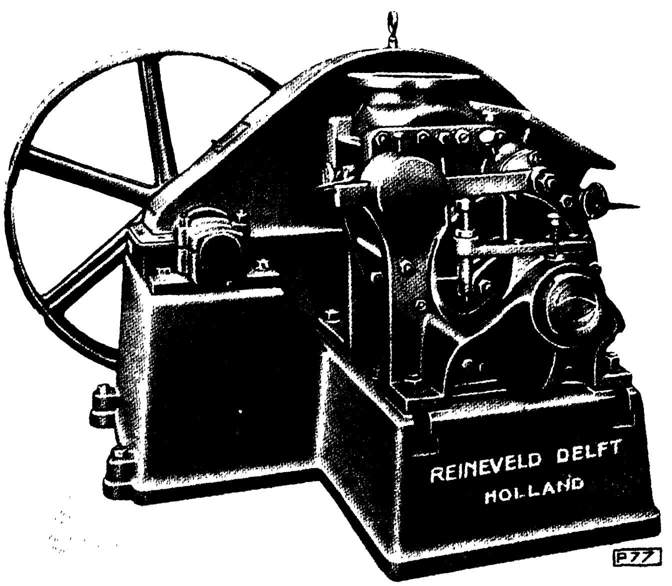
Fig. 533.—Rotary Displacement Pump.

The proper field for rotary displacement pumps is not in the first place to pump water or juice, but chiefly viscous liquids like syrup, molasses, fuel oil and massecuites. For the pumping of massecuites, the design has to be so arranged that sugar crystals will not be ground by the pump parts and moreover the crystals should have as little abrasive effect on the pump interior as possible.

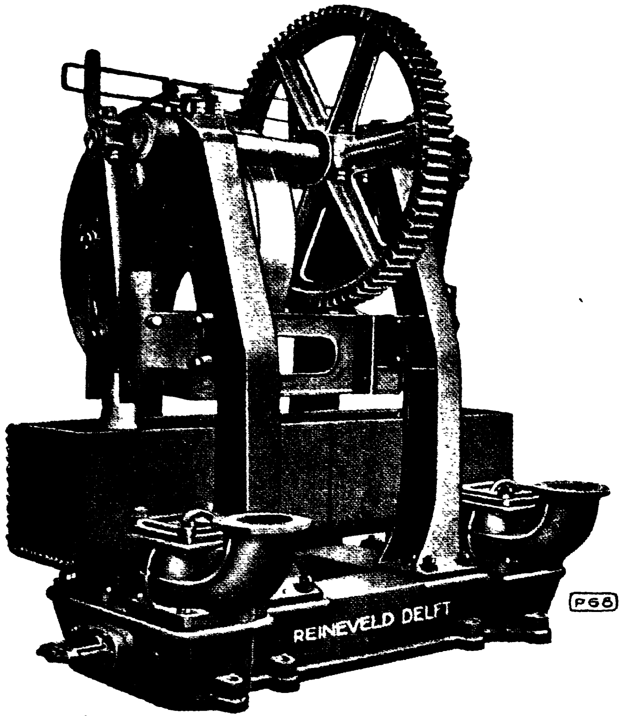
In *Fig.* 533 is shown a *Rotary Displacement Pump* for molasses. The author has installed a pump of this design and favourable operating results have been obtained. The drum *a* is rotated at between 120 and 150 r.p.m. for molasses and



High Speed Steam Engine with Centrifugal Pump for Injection.
(H. W. Aitken Co., Ltd.)

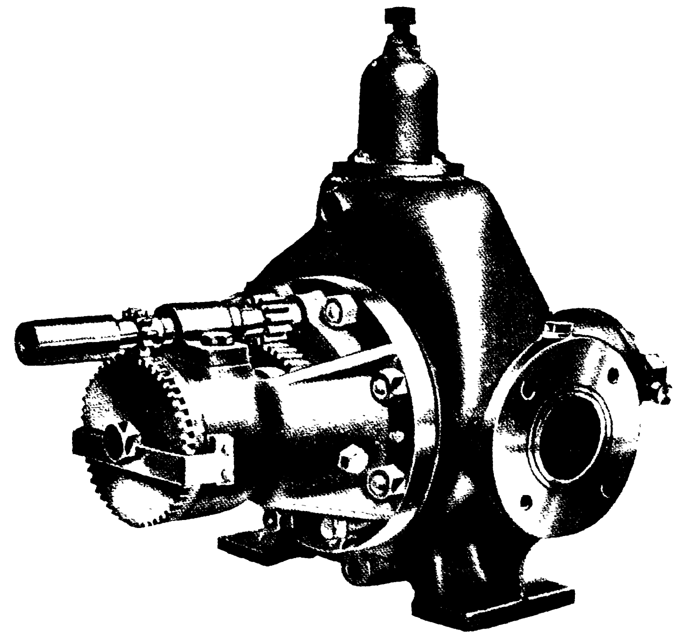


Belt-driven Rotary Massecuite Pump.
(Reineveld N.V.)



VERTICAL MASSECUITE PUMP (AKKERMAN TYPE).

(*Reineveld N.V.*)



REVERSING FLOW SYRUP PUMP FOR LOADING AND DISCHARGING TANKS WITH CONSTANT DIRECTION OF ROTATION.

(*Stothert & Pitt, Ltd.*)

at about one-fourth of these speeds for massecuite. A scraper b is held on the periphery of the drum by means of two spring-loaded levers on the extremes of the shaft on which it is mounted ; it provides a seal between the suction and discharge spaces of the pump. There is some wear on the scraper end, but this can be readily renewed by having a detachable tip. The very small clearance of the drum in the pump housing will be of little importance to the volumetric efficiency in case of thick liquids.

When D is the inside diameter in inches of the housing in which the drum revolves, L the drum length in inches, and A the area of the drum section in square inches, then the pump output in Imp. gals. per minute will amount to :

$$V_{min} = \frac{(0{\cdot}785\ D^2 - A) \times L \times n \times \eta_v}{277} \ \ldots\ldots \ (141)$$

in which n is the number of revolutions per minute of the drum shaft. In case U.S. gallons are required, the divisor 277 has to be changed to 231. The volumetric efficiency η_v is generally between 0·60 and 0·75, decreasing with the clearance, through wear, of the drum in the pump housing. The power consumption must be calculated according to formula (138) with a mechanical efficiency of about 0·75, the sticky material having a large friction and adhesion coefficient.

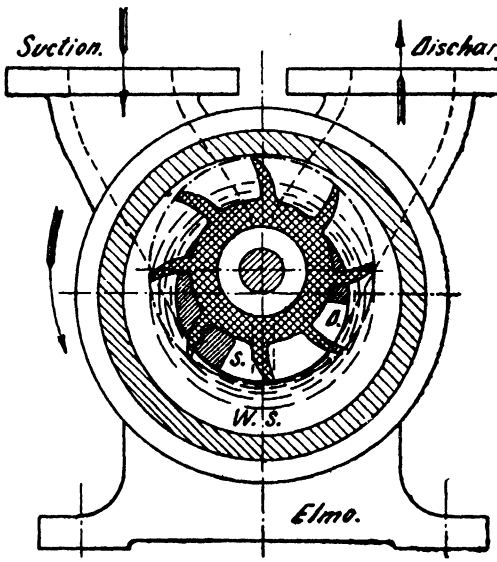


Fig. 534.—Rotary Displacement Air Pump.

For molasses the pump shaft carries a fast and loose pulley for belt drive, and two drums, arranged at 90° on the shaft, ensure a smooth discharge. A division wall between the two drum chambers is required. For massecuite a heavier design is used with only one drum, which revolves at a slow speed. The driving is by means of single or double gear, for belt and direct electric drive respectively.

The pumps will aspirate about 10 ft., but have to be primed with molasses to form a good seal for the drum. Heads up to 100 ft. (about 70 lbs./sq. in.) are obtainable. Pump sizes are from 4 to 8 in. pipe connexions and from 3,000 to 10,000 Imp. gals. per hour.

Several other types of rotary displacement pump are used for molasses and fuel oil with satisfactory operating records. The handling of massecuites, containing sugar crystals, in displacement pumps of the geared impeller type, will cause difficulties through the grinding of the crystals and the heavy abrasion on the intermeshing pump members.

In *Fig.* 508 has been shown a rotary air pump of the centrifugal type, but for vacuum or compressor performance there are now *Rotary Displacement Air Pumps*, and an ingenious design, already in use in cane sugar factories, is shown in *Fig.* 534. A bladed brass drum rotates eccentrically in the pump housing. A small amount of water is constantly entering the pump interior and through the high speed of the drum, a circular water seal *W.S.* is formed and the annular space thus obtained below the rotor or drum is the displacement space. The suction openings *S* are in both lateral covers, as well as the discharge openings *D*. The direction of rotation is given by the arrows and it will be seen that the annular space widens from the suction passage towards the bottom and is reduced towards the discharge opening. Suction and compression thus take place in consecutive order.

The displacement of the pump has to be calculated according to (141), in which *D* is the interior diameter of the water seal. A part of the sealing water is discharged with the air, but this water can re-enter the cycle and a small circulation water tank is thus all that is required. The water serves for cooling during the compression performance.

A vacuum up to 28 in. or higher can be obtained with these pumps, and they are built up to capacities of 900 cub. ft. per min. Their speed is up to 3000 r.p.m. and they can be direct-coupled to an electric motor.

Rotary Air Compressors of the displacement type are built in single or multi-stage arrangement up to 3500 cub. ft./min. air aspirated. For pressures up to 20 lbs./sq. in. they are air-cooled, but for higher pressures require a cooling water jacket. Turbo-compressors are only built for very large capacities, beyond the requirements of cane sugar factories.

CHAPTER XXVIII.

CRYSTALLIZERS.

PRINCIPLES OF MASSECUITE COOLING — TYPES OF CRYSTALLIZERS — RAPID COOLING CRYSTALLIZERS.

In Chapter XXII it has been explained that the degree of supersaturation of the mother-liquor of a massecuite will increase when the temperature drops ; at high temperature more sugar is kept in solution.

The cooling of the massecuites thus has a definite purpose and it can be done either after the vacuum pan has been discharged or, as is nowadays possible, within the pan itself. On cooling, the crystals of a given massecuite will grow and a higher sugar recovery is obtained. In *Fig.* 450 (Chapter XXII) a chart is drawn to show how far the recovery of sugar from a massecuite will go in practice.

The run-off—that is, the mother-liquor left over after the crystals have been removed—of first boilings is re-introduced into the process, and thus cooling of first massecuites is generally considered of lesser importance ; but the fundamental fact should be reckoned with that the more sugar is crystallized out of the first mother-liquors, the more efficient the process will be.

The mother-liquor remaining from final boilings is called molasses and has value only as a by-product, and the tendency is to exhaust these molasses as far as possible. This goes to prove that the cooling of the final massecuites is of paramount importance.

1.—Principles of Massecuite Cooling.

The ordinary cooling of massecuites is done in crystallizers, these being open vessels of semi-cylindrical shape, with a prismatic top section. Lyre-shaped cross-sections are also found. The purpose of all these open crystallizers is to transfer the heat of the massecuite to the surrounding air, and a more effective heat dissipation takes place when the water contained in the massecuite is allowed to evaporate, as explained in Chapter XXV (*re* cooling ponds).

The mother-liquor of a massecuite being a very viscous material, especially at low purity and decreasing temperature, the heat transfer is obviously greatly handicapped and the time involved sometimes runs to 120 hours for cooling massecuite from about 70°C. (158°F.) down to 33°C. (91°F.); so makers of this kind of equipment have directed their energies to evolving a design in which the cooling cycle would be shortened.

This has resulted in the evolution of the *rapid cooling crystallizer*, of which there are now several efficient designs on the market. The cooling agent is water or air and, in all, three types have been established : —

(1) Cooling by heat dissipation through the air and by surface evaporation of the massecuite. The cooling performance in open crystallizers belongs to this class.

(2) Cooling in open or closed crystallizers by means of water, which is led through pipes or closed conduits, the separating wall generally being of iron. Most of the rapid cooling crystallizers are based on this system.

(3) Cooling in open system by the evaporation of cooling water which is separated from the massecuite by a metallic (iron) wall.

In all crystallizers where water is evaporated during cooling, an increase in Brix results, which also adds to the supersaturation of the mother-liquor. The author has on record a density increase from 94 to 96° Brix, and the point should be emphasized that crystallizers must be located in a well-ventilated building, so as to have efficient air circulation, but this basic fact is not always adhered to.

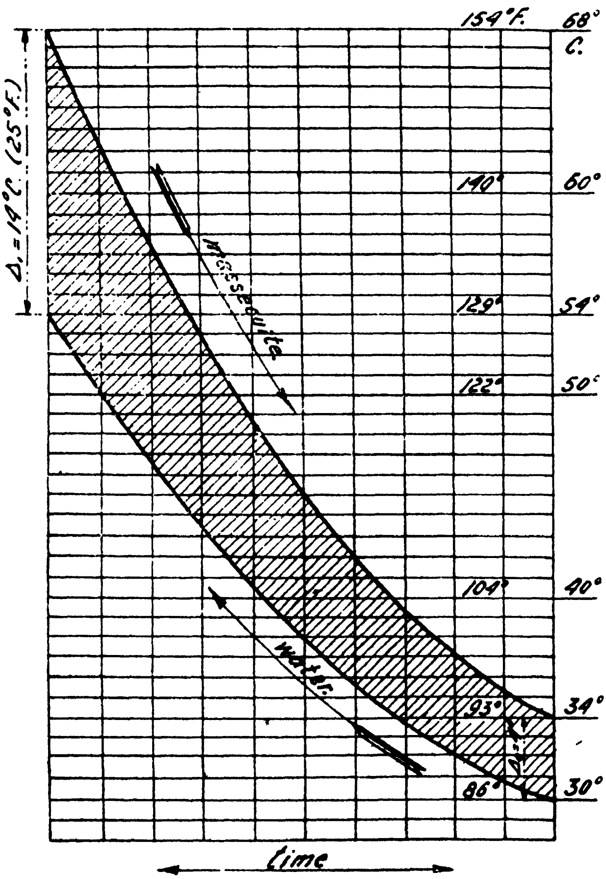


Fig. 535.—Counter-Current Cooling Graph.

Closed crystallizers under vacuum will also utilize evaporation, but only until the cooling temperature has reached the vapour temperature equivalent to the prevailing vacuum of the condenser to which they are connected; further evaporation then stops.

The viscosity of the mother-liquor increases under reduced temperature and purity and increased density; circulation by mechanical means has therefore to be provided, not only for more uniform cooling but also to prevent the sugar crystals from settling at the bottom of the crystallizer with the troublesome consequence of cake formation. With high purity massecuites, e.g., for white sugars in refineries, this caking will clog the discharge outlets, etc., when proper circulation is not maintained and the author knows instances where the trouble has had to be removed with iron bars and sledge-hammers.

Generally, the circulation in a common crystallizer is achieved by a slowly revolving spiral ribbon; but some operating engineers have expressed doubts to the author whether such a spiral was a necessity, as no transportation in any particular direction is needed, but simply a stirring motion, and hence there is no inconvenience in having the crystallizer equipped instead with paddles, as has been done in several instances. These paddles, preferably, should be provided with tips of heavy flat iron, arranged at an angle of about 30 to 45° with the crystallizer axis, and two adjacent paddle tips should have opposed angularity.

Factory superintendents also differ in their ideas regarding the speed of cooling and the lowest temperature feasible for handling the viscous low grades. Nowadays, the prevailing opinion is that the cooling speed, if raised, has no detrimental effect on the formation, e.g., of false grain ; and before separating the crystals in the centrifugals, the cooled massecuite should be heated rapidly to about 110°F. to reduce the viscosity. This final heating, when done quickly, has not led to any re-dissolving of already crystallized sucrose.

The cooling performance differs from any heating performance so far explained, as there is no constant temperature nor any constant temperature difference, but only varying ones. The cooling medium, air or water, will have a higher exit than entrance temperature, whereas with the massecuite the opposite is the case.

For air-cooled crystallizers only empirical data are known and a cooling time of 72 hours is considered normal practice for final boilings. First and second boilings in raw sugar factories are cooled for a very short period of time, the crystallizers generally being storage tanks before the massecuite goes to the mixers of the centrifugals.

In Chapter XXII the calculation for the proportion of the different boilings is given ; and according to the assumed purities the dry matter in massecuites amounts to about 150 per cent. on dry matter in the syrup. The total amount of massecuite is divided as follows :

	Per cent.
First boilings	50
Second boilings	28
Third boilings	22
Total	100

From previous assumptions, it has been concluded that one ton of cane will yield 2150 lbs. thin-juice at 15° Brix and with massecuite of 90° Brix minimum $2150 \times 15 \div 90 = 356$ lbs. of syrup will be produced. One cubic foot of massecuite at this density weighs about 92·5 lbs. and taking into consideration the before-mentioned 150 per cent. as being the total quantity of massecuite in process, there will be $356 \times 1{\cdot}5 \div 92{\cdot}5 \simeq 6$ cub. ft. of massecuite in process per ton of cane ground.

The division for first, second and third boilings thus will be respectively 3, 1·6 and 1·4 cub. ft. The cooling time is taken as :—

For first boilings	12 hours
For second boilings	12 ,,
For third boilings	72 ,,

In a 24-hour period the required crystallizer capacity will amount to :— $3 \times 12 \div 24 + 1{\cdot}6 \times 12 \div 24 + 1{\cdot}4 \times 72 \div 24 = 6{\cdot}5$ cub. ft. per ton cane ground per 24 hours or about 160 cub. ft. per ton cane ground per hour, which is an average figure in practice. From a few factory inventories taken in Cuba, the total crystallizer capacity varies between 7·5 and 10 cub. ft. per ton cane/24 hours. In the Philippines, the latter figure per short ton of cane/24 hours is considered standard for air-cooled crystallizers.

The capacity of each crystallizer should be such that it holds a full pan strike. In those factories having great variations in vacuum pan size, it should be arranged that strikes from different pans are not mixed in the crystallizers.

Water-cooling of the massecuites is done according to two different systems, i.e., *counter-current cooling* and *intermittent cooling*. The first system must be a continuous one, for the reason that the massecuite and the cooling water flow in opposite directions.

The counter-current cooling performance is graphically explained in *Fig.* 535. The assumed temperatures are :

Massecuite inlet	68°C. (154°F.)
Massecuite outlet........	34°C. (93°F.)
Water inlet	30°C. (86°F.)
Water outlet	54°C. (129°F.)

Of course, the calculation can be made with any prevailing temperatures of the massecuite and cooling water.

The entering temperature difference of the water is : $\Delta_0 = 34 - 30 = 4$°C. and the discharge difference : $\Delta_1 = 68 - 54 = 14$°C. The mean temperature difference, according to formula (96) (Chapter XVIII), amounts to :—

$$\Delta_m = \frac{14 - 4}{\log_e \frac{14}{4}} = 7{\cdot}98°\text{C.}$$

This mean temp. diff. (M.T.D.) has a bearing upon the heat transmission, and the smaller it is, the greater will be the cooling surface required. The quantity of the cooling water needed, i.e., the *cooling water ratio*, is derived from :

$$W = \frac{(t_x - t_y) \times C_{mass.}}{t_1 - t_{11}} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (142)$$

in which : W = lbs. cooling water per lb. massecuite (or in kgs.).
t_x = entrance temperature of massecuite in °F. or °C.
t_y = exit temperature of massecuite in °F. or °C.
$C_{mass.}$ = specific heat of massecuite (about 0·4).
t_1 = exit temperature of water in °F. or °C.
t_{11} = entrance temperature of water in °F. or °C.

For the optimum temperatures assumed, the cooling water ratio is : $(68 - 34) \times 0{\cdot}4 \div (54 - 30) = 0{\cdot}55$.

In practice for average operation 0·8 lbs. water per lb. massecuite is required with counter-current cooling and though the gradual cooling with a maximum temperature difference of 14°C. is assumed, there is no danger of the formation of false grain, which might be produced by abrupt cooling.

Intermittent cooling is more complex, because it is not continuous. The massecuite cools gradually and the cooling water at the beginning leaves with 54°C. assumed temperature, whereas at the end of the cooling performance it leaves with only 32·52°C. temp. The water consumption towards the end of the cooling will thus be considerably higher.

In *Fig.* 536 an attempt is made to explain graphically the intermittent cooling performance. At the left side of the figure are drawn the beginning and end periods of the cooling operation, during the infinitesimal time δt. The massecuite temperature is considered constant during this minimum time limit.

The temperature differences at the beginning and the end of each period are thus :

$$\Delta_{s1} = 68 - 30 = 38°C.$$
$$\Delta_{s2} = 68 - 54 = 14°C.$$

For the starting period and for the final one :

$$\Delta_{f1} = 34 - 30 = 4°C.$$
$$\Delta_{f2} = 34 - 32{\cdot}52 = 1{\cdot}48°C.$$

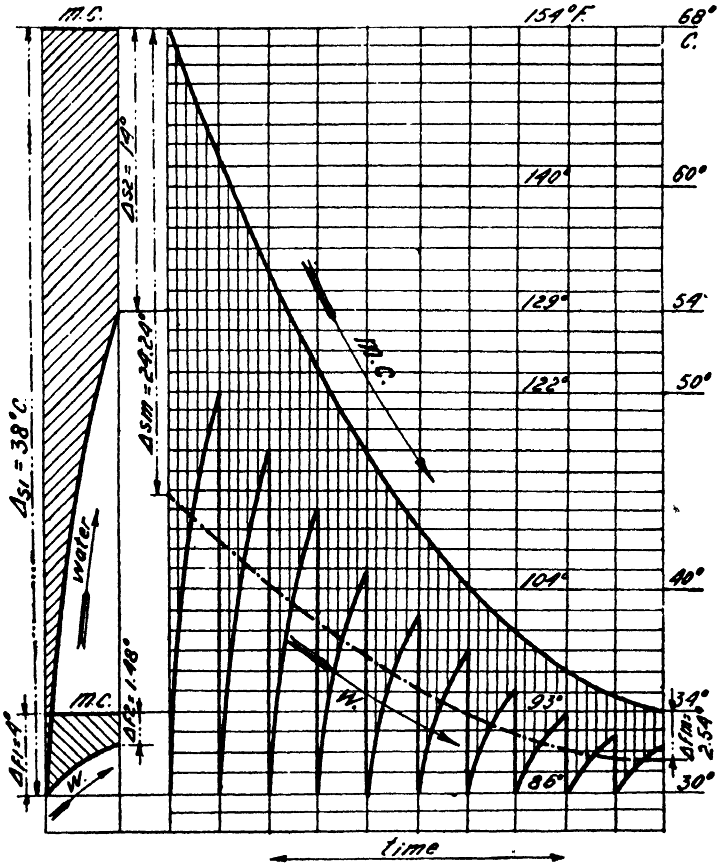


Fig. 536.—Graph of Intermittent Cooling Performance.

The proportion between these temperatures is 0·37 in both instances, which proportion is required for the continuity of the performance (thus 14 ÷ 38 = 1·48 ÷ 4). At the start the M.T.D. thus amounts to :

$$\Delta_{ms} = (38 - 14) \div \log_e (38 \div 14) = 24{\cdot}24°C. ;$$

and for the final period :

$$\Delta_{mf} = (4 - 1{\cdot}48) \div \log (4 \div 1{\cdot}48) = 2{\cdot}54°C. ;$$

and for the complete performance :

$$\Delta_m = \frac{\Delta_{ms} - \Delta_{mf}}{\log_e \dfrac{\Delta_{ms}}{\Delta_{mf}}} = \frac{24{\cdot}24 - 2{\cdot}54}{\log_e \dfrac{24{\cdot}24}{2{\cdot}54}} = 9{\cdot}66°C.$$

thus about 20 per cent. higher than for counterflow cooling, at the same assumed temperatures.

At the right-hand side of the figure the temperature drop of the massecuite from 68 to 34°C. is shown. The time co-ordinate has been divided into a number of sections and at each section the cooling water enters at 30°C. but leaves with a gradually decreasing temperature until the end of the cooling performance. The mean average temperature is shown by the chain-dotted line, and varies between 24·24 and 2·54°C.

The cooling water ratio for intermittent cooling is:

$$W_i = \frac{(t_x - t_y) \times C_{mass.}}{\Delta_m} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (143)$$

in which W_1 is the quantity of cooling water per lb. massecuite, and Δ_m the M.T.D. as explained above; t_x, t_y and $C_{mass.}$ have the same significance as for formula (141).

For the assumed temperatures, the cooling water ratio thus theoretically amounts to : $(68 - 34) \times 0{\cdot}4 \div 9{\cdot}66 \simeq 1{\cdot}40$ and in practice about 2 lbs. water per lb. massecuite will be required.

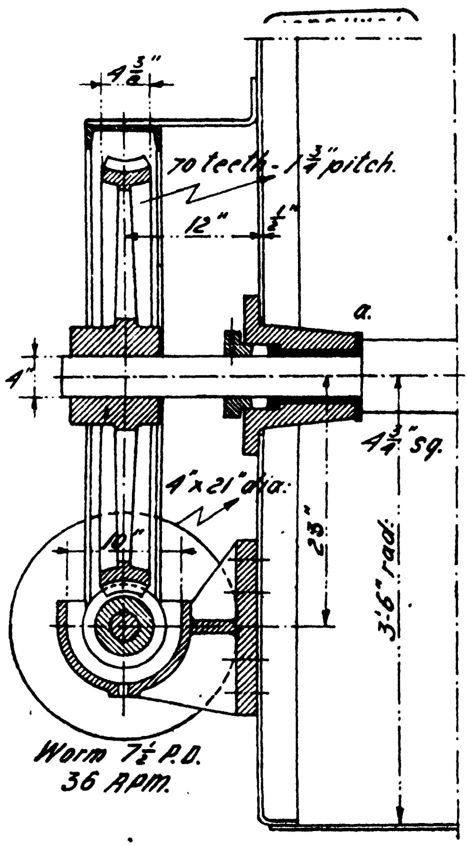


Fig. 537.—Crystallizer Worm Drive.

2.—Types of Crystallizers.

The different types of air-cooled crystallizers include circular, U-shaped or lyre-shaped arrangements, and the diameter varies between 5 and 7 ft. as an average. The length is generally between 16 and 32 ft. and the capacity from 450 to 1800 cub. ft.

When not revolving on its own axis, a crystallizer requires a stirring device for the reasons already mentioned. This stirring device is composed of a central shaft, on which the stirrer arms for the spiral ribbon are arranged, or else simple paddles are used.

The *Worm Drive* for such a stirrer is shown in *Fig.* 537 ; the worm wheel and the worm itself being machine-moulded and of cast iron. The latter is arranged below the worm wheel, and an oil or grease pan is formed which is integrally cast with the two bearings. These bearings are arranged to take up the lateral thrust caused by the worm drive. The whole is bolted to the crystallizer front plate. A guard of channel section is bent around the worm wheel for neatness and to recover any grease that may fall off the teeth. Fast and loose pulleys are arranged for belt drive, but since the crystallizer station is not always as clean as could be desired owing to massecuite drippings, etc., the worm drive is sometimes arranged superimposed on top of the worm wheel. Other manufacturers use chain drives, instead of belting.

The shaft is provided with a packing gland bearing bolted to the front plate having a brass liner of sufficient bearing surface. The spiral stirrer or flight, employed in so many crystallizers, exerts an axial thrust on the main shaft and generally the rear plate support is designed as a thrust bearing. In case the stirrer is composed of two opposed parts, thrust rings *a* are arranged at both end plate supports.

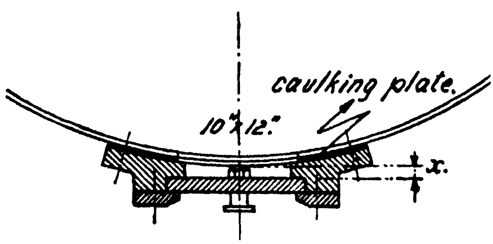


Fig. 538.—Massecuite Discharge Valve.

The square shaft has intermediate journals inside the crystallizer at about 10 ft. centre distances. The intermediate bearings are brass or white metal lined and are lubricated by a grease cup, above the massecuite level.

The arms of the stirrer are generally made of flat iron, ¾ in. × 3½ in., whereas the ribbon is made of flat iron ½ in. × 3 in. In case round shafts are used, cast iron or cast steel hub pieces with detachable cap are used, secured to the shaft by pins.

Paddles without ribbons are also used, and the centre distance between two adjacent paddles is about 18 in.

The speed of rotation of the main shaft is from ½ to ¾ r.p.m. and the power consumption for each crystallizer between 1·5 and 3 h.p. according to size. The ribbon or paddles should, of course, not rub on the shell, but clear it by about ¾ in. The shaft also has to be well aligned, as otherwise excessive power will be required.

A *Massecuite Slide Discharge Valve* is shown in *Fig.* 538, which can be operated by a common lever gear, or by a screw spindle arrangement. Care should be taken that the distance *x* remains as short as possible, to avoid dead spaces which will not share in the stirring performance. The standard size is 10 in. × 12 in. as an average.

Massecuite gutters should have an inclination of 1 in 20, or more if conditions will allow.

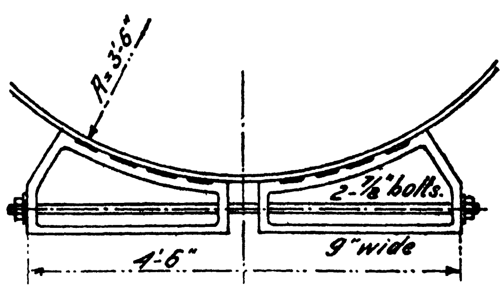


Fig. 539.—Supports of the Crystallizers.

The *Supports of the Crystallizers* should be arranged at the ends, generally forming an integral part of the cast iron or steel end plates. Moreover, intermediate supports, as shown in *Fig.* 539, should be arranged at about 6 ft. intervals and as close to the intermediate bearings as possible. The construction shown will ensure good bearing of the shell through the split design and the two throughgoing bolts.

3.—Rapid Cooling Crystallizers.

During the last decade special attention on the part of factory superintendents, as well as of manufacturing engineers, has been given to the evolution of a rapid cooling crystallizer. The quicker the cooling of the massecuite, the less storage capacity is required.

In *Fig.* 540 a *Spray-cooled Crystallizer* is shown. The cooling water is sprayed on the shell by means of two perforated pipes, one on each side. A thin film of water will dribble on about 60 per cent. of the shell surface and cooling through evaporation will take place. The angles *a* clear the bottom part of the shell from water, and these crystallizers are generally arranged on the factory floor where gutters can be laid in the concrete towards the drain. Although a very efficient cooling principle is involved, this system is not generally used. The thick shell walls (¼ in. to ⅜ in.) and possible incrustations on the inside are not favourable for efficient heat transmission.

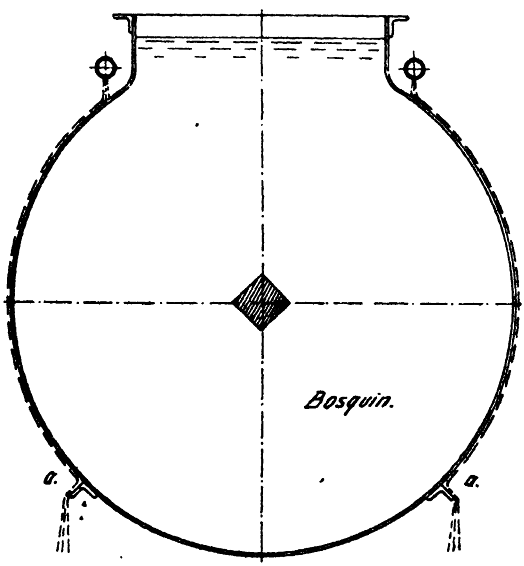


Fig. 540.—Spray-cooled Crystallizer.

It should be mentioned that actual *scraping* of the cooling surfaces has not so far been attempted with rapid cooling crystallizers. Favourable results from the periodical scraping of cooling or heating surfaces are reported from other industries, where viscous materials are handled.

Cooling by means of tube elements can be divided into two groups—the *fixed* and the *moving tube nests*. A design belonging to the first group is shown in *Fig.* 541. This arrangement has found wide application in Cuba, Hawaii and other countries. The sets of coils are arranged vertically in the crystallizer at about 16 in. centre distances. The main pipe lines for cooling water inlet and outlet have branches for each set of coils. To get good distribution of water over all the coils, a washer with an orifice of about ⅛ in. diameter is laid in the universal joint coupling of the ¾ in. branches. For good circulation, the coil windings are laid horizontally. Between the sets of coils paddles keep the massecuite in motion and there is practically no caking between the coil windings.

In a Cuban factory the same massecuite was cooled in crystallizers with and without these coils, giving after 36 hours the following results :—

	With Coils	Without Coils
Brix massecuite	96·8	96·8
Final temperature	92°F.	140°F.
Purity run-off	30·39	35·20

The surrounding air temperature was noted as 86°F.

Like most of the rapid cooling designs, it can be built into existing crystallizers at reasonable cost.

Revolving cooling elements were first used in the West Indies, being of French design. The cooling element is composed of a spirally wound copper tube of small pitch, arranged on the stirring arms and having the water entrance and discharge through the hollow shaft. This system has not found wide application due to breakage and leakage of the coils. There is now a British design, having iron pipe coils parallel to the shaft, which are used as the stirring device. This construction has found many applications in the British colonies, the cooling time being reduced to about 50 per cent. of that of the non-cooled crystallizers.

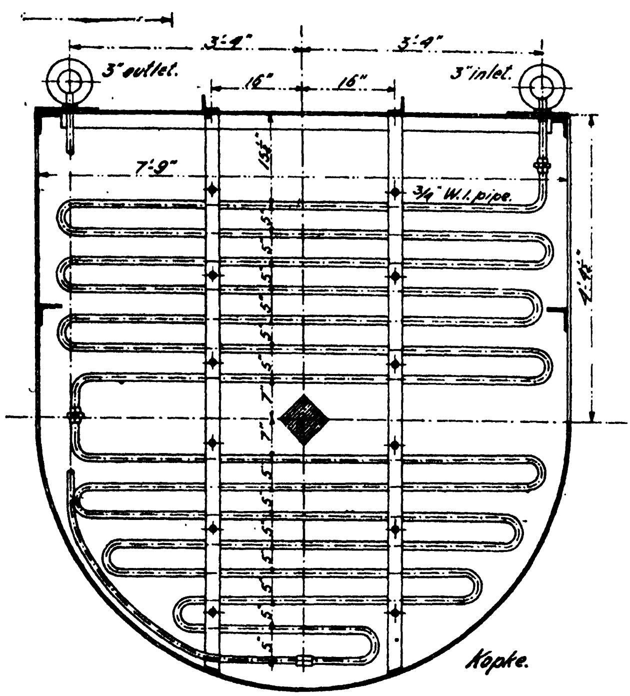


Fig. 541.—Fixed Cooling Tube Nests.

Another design is the *Revolving Disc Crystallizer* shown in *Fig.* 542. On the main shaft are arranged a number of hollow discs about 7½ in. apart and about 2 in. thick. The disc has a sector left out for convenient attachment to the shaft, and the water is charged and discharged through the hollow shaft ends and circulates consecutively through all the discs and the flow inside the disc is indicated by arrows in the drawing.

This crystallizer has the unique feature that it gives continuous operation, based on the *counter-flow principle.* The hot massecuite is charged in at one end and the cooled massecuite discharged at

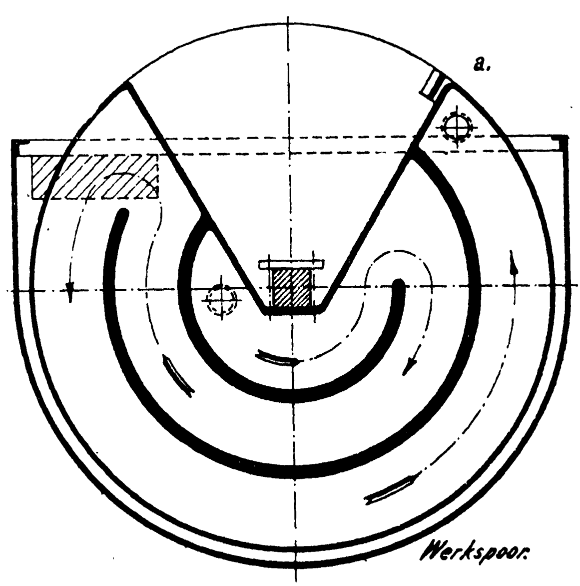


Fig. 542.—Revolving Disc Counterflow Crystallizer.

the other, whereas the cooling water flows in the opposite direction. As there is no abrupt cooling by counterflow performance, false grain formation need not be feared and free curing massecuites are reported to be obtained. The cooling time is between $1\frac{1}{2}$ and 24 hours, according to the massecuite and cooling water temperatures, the cooling surface and the purity of the massecuite; low grades requiring the longer time.

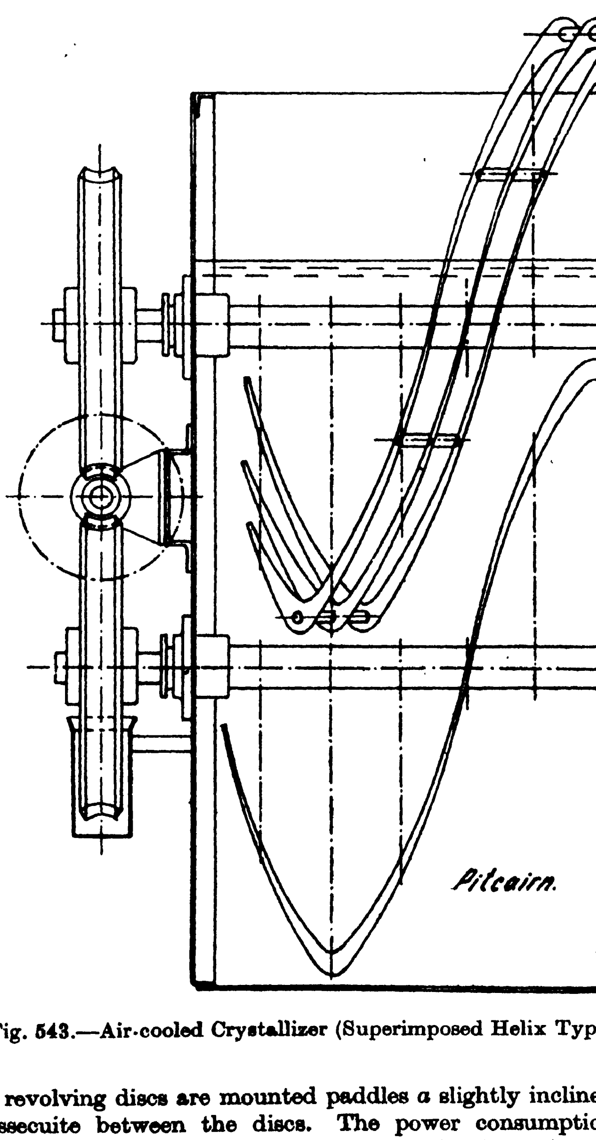


Fig. 543.—Air-cooled Crystallizer (Superimposed Helix Type).

On the revolving discs are mounted paddles *a* slightly inclined, for stirring up the massecuite between the discs. The power consumption to operate this crystallizer is about the same as for standard spiral stirrers.

These counterflow crystallizers are of great value in cooling first boilings, as it gives a greater output in first sugars and moreover reduces the number of boilings.

The horizontal rotary vacuum pan, shown in *Figs.* 470 and 471 is used as a rapid cooling crystallizer after the strike has been finished and transfer of the massecuite is thus not necessary. Very good operating results have been reported, not only as to the cooling time, but also as to the good curing qualities of the massecuite in the centrifugals. This rotary pan has the largest cooling surface in proportion to its capacity of all the intermittent rapid cooling crystallizers treated. The total cycle, boiling included, has been reported to be accomplished in 16 hours with final strikes.

The cooling of the massecuite in the same vessel in which it has been boiled requires not only the cooling of the massecuite but also of the metallic parts of the vessel. Nevertheless, these metal parts have good heat transmission.

An *Air-cooled Crystallizer* of the *superimposed helix* type is shown in *Fig.* 543; this has found application in the Philippines, and in other countries.

The two helixes revolve at the same speed in opposed directions and the upper one is made of three-fold ribbon and revolves for a large part above the massecuite level. The adhering massecuite on and between the ribbons of the upper helix is efficiently cooled, especially as the massecuite surface is constantly broken. Any foaming by air dragged into the massecuite has not proved a drawback and a reduction to 50 per cent. of the cooling time has been achieved ; 5 cub. ft. per short ton of cane per 24 hours being the total capacity required, when final strikes are cooled only.

The cooling surfaces of different types of rapid cooling crystallizers are tabulated as follows :—

Type	Capacity	Cooling Surface	Proportion
Hamakua	500 cub. ft.	100 sq. ft.	0·200
Huck (jacket)	883 ,,	275 ,,	0·311
Bosquin	1236 ,,	407 ,,	0·329
Ragot (revolving coils)	900 ,,	303 ,,	0·336
Kopke	1833 ,,	694 ,,	0·379
Herisson	300 ,,	300 ,,	1·000
Lafeuille	1160 ,,	160 ,,	1·862
Werkspoor :			
High grade	494 ,,	1480 ,,	4·000
Low ,,	550 ,,	190 ,,	0·340

The heat transmission of the very viscous material is low and depends on several factors. The Java Experimental Station has supplied us with the following data :—

Lafeuille	2·47 cal./sq. m./1°C./min.
Werkspoor	2·78 ,, ,, ,,

or in British units respectively 0·49 and 0·54 B.Th.U./sq. ft./1°F./min.[1]

1 See *Mededeelingen*, 1930, P. HONIG and W. F ALEWIJN ; *Int. Sugar Jl.*, 1931, p. 402.

CHAPTER XXIX.

SUGAR CENTRIFUGALS.

PRINCIPLES — TYPES OF CENTRIFUGALS — DETAILS OF MACHINE — CENTRIFUGAL DATA.

The sugar centrifugal machine consists principally of a perforated drum or basket, which revolves at a high speed and into which the massecuite is charged. Through the centrifugal force exerted the mother-liquor surrounding the crystals is separated or thrown off through the perforations, whereas the crystals remain within the perforated lining inside the basket. The separation is a gravity one, but the gravity force is greatly increased through the centrifugal action. The perforations, of course, have to be of such small size that the crystals cannot pass; but when false or secondary grain is formed, it may pass the perforations and thus be lost for the sugar, i.e., crystal recovering, process.

To decant liquids containing insoluble impurities of higher gravity than the former, the blind or unperforated basket is used and then the purified or clarified liquid flows over the top lip of the basket.

1.—Principles.

The centrifugal force exerted is proportional to the square of the peripheral speed of the content in the basket, according to formula (59) of Chapter V, and as the peripheral speed depends upon the basket diameter and the number of revolutions per minute, it will be clear that a smaller diameter will require a higher number of r.p.m. to obtain the same centrifugal force.

Formula (59) is written: $C = \dfrac{M \times V^2}{R}$ (see page 129) and $V = 2\pi \times R \times N \div 60$. Now $2\pi \times N \div 60$ is the angular velocity w and the formula thus might be written :—

$$C = M \times R \times w^2 \quad \ldots\ldots \quad (144)$$

For different numbers of revolutions, the value w amounts to :—

for 750 r.p.m.	78·5
for 900 r.p.m.	94·2
for 1000 r.p.m.	104·7
for 1200 r.p.m.	125·6
for 1500 r.p.m.	157·0

The symbols signify : C = Centrifugal force in lbs.
M = Mass = weight $\div$ 32·16
V = Peripheral speed in ft./sec.
N = Number of revolutions per minute.
R = Length of the centrifugalling radius in ft.

A second way to express formula (59) is easily obtained from the following :

$$C = \frac{W}{32{\cdot}16} \times \frac{4\pi^2 \times R^2 \times N^2}{60^2 \times R} = \frac{W \times R \times N^2}{2935} \quad \ldots\ldots \quad (145)$$

In a 40 in. centrifugal 1 lb. of molasses with an assumed centre of gravity lying at 18 in. (1·5 ft.) from the axis of rotation and the basket revolving at 1150 r.p.m. will develop a centrifugal force of $1 \times 1{\cdot}5 \times 1150^2 \div 2935 \sim$ 675 lbs., thus the gravity force is increased 675 times.

The shell of the basket has to withstand the full centrifugal force of the total charge, and high-class material with a number of reinforcing hoops of high tension steel has to be used. Any rupture of the basket lining will not only cause serious damage, but the monitor case might also burst and endanger the operators. Manufacturers, therefore, stamp on the spindles the allowable safe number of revolutions per minute. But it may happen that through corrosion the basket material has become thinner, and as soon as any trace of bulging occurs, it is a sure sign that the baskets must be renewed. It is equally dangerous to run centrifugals above the limit the designers allow. New centrifugals are run under full charge in the testing pit of the manufacturers at a higher speed than the normal running one.

The *number of centrifugals required* depends on three factors :—

(1) The weight of sugar to be dried per hour.
(2) The weight of each charge (in dry sugar).
(3) The time to complete a cycle, comprising charging, accelerating, drying, washing with steam or water and emptying.

For good centrifugalling *double curing* is advisable, i.e., one first separates the run-off in self-discharging centrifugals and then the sugar obtained therefrom is mixed with a liquor of high purity, after which the resulting magma is centrifugalled in a second set of centrifugals, in which also a washing operation is done. In Java this *modus operandi* is extensively used for white consumption sugars, and other countries are adopting it as well.

For a given factory, the number of boilings and the quantity of sugar contained in each of them has to be known and whether the so-called forerunners are used or not. The number of centrifugals required, x, then will be :

$$x = \frac{S \times t}{B \times 60} \qquad (146)$$

in which : S = Dry sugar in lbs. contained in the specific massecuite per hour.
t = Time in minutes for a complete cycle (average).
B = Capacity of basket in lbs. dry sugar (average).

For each battery, thus for first, second and third sugars, etc., the number of centrifugals required can be calculated. A spare machine should be provided in each battery, or two when these are large, or contain say over ten units.

The time t depends upon many factors ; the charging should be done as quickly as possible, after the centrifugal has commenced running. The acceleration can be shortened by a higher power input and this initial input in modern centrifugals may attain to several times the normal power consumption. High speed, also, will shorten the cycle, but there is a divergence of opinion amongst experts as to whether such high speed will produce a better purging action. Moreover, the drying time depends on the physical properties of the mother-liquor, its temperature and viscosity, and the uniformity of the sugar crystals which will give free flow to the liquor. Washing or steaming out will require additional time. The discharge performance can be speeded up by self-discharging, which is feasible for high-grades, or else by the use of mechanical unloaders.[1]

1 For a theoretical treatise about the time involved, see TH. J. D. ERLEE, *Het Archief*, 1931, pp. 10-18, 41-58.

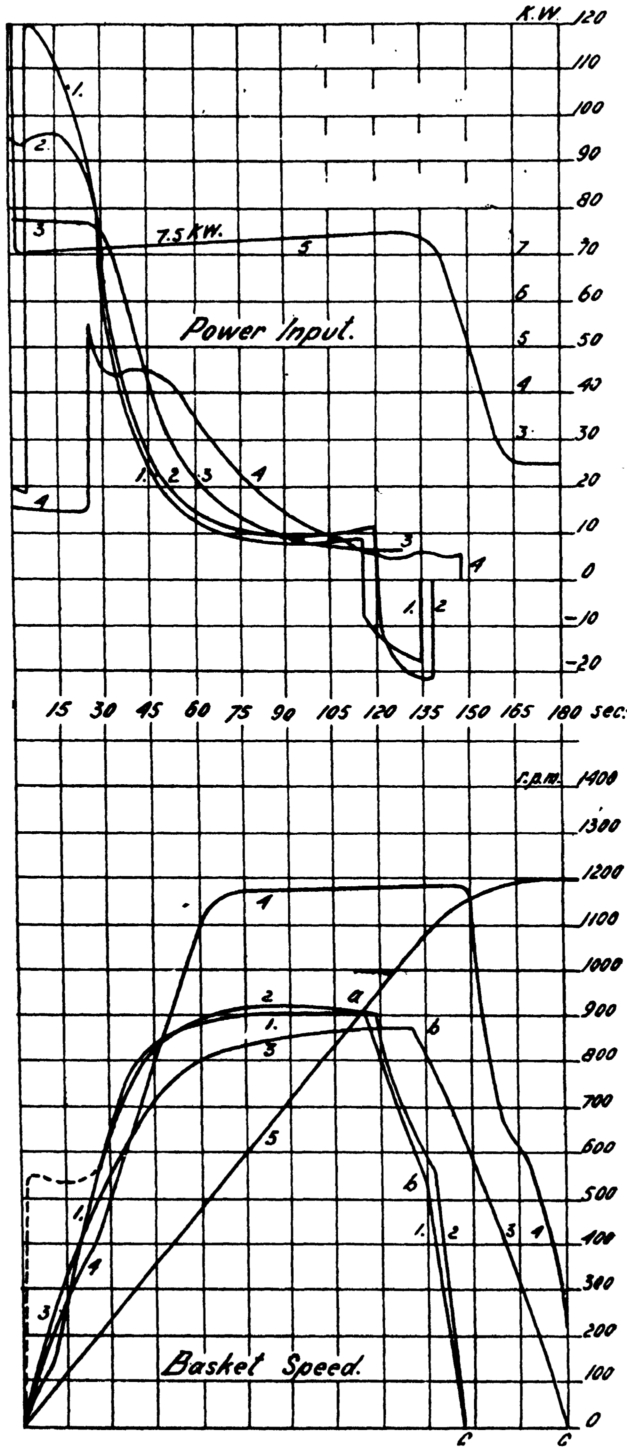
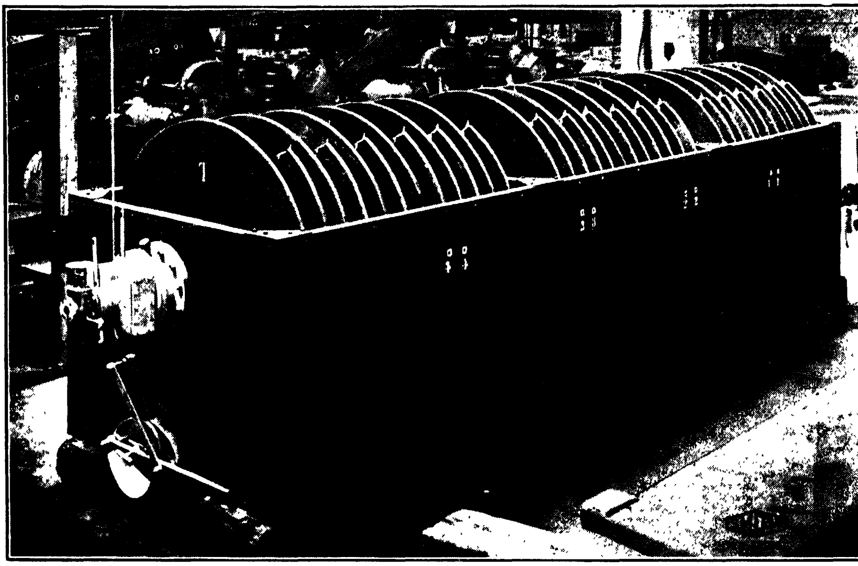
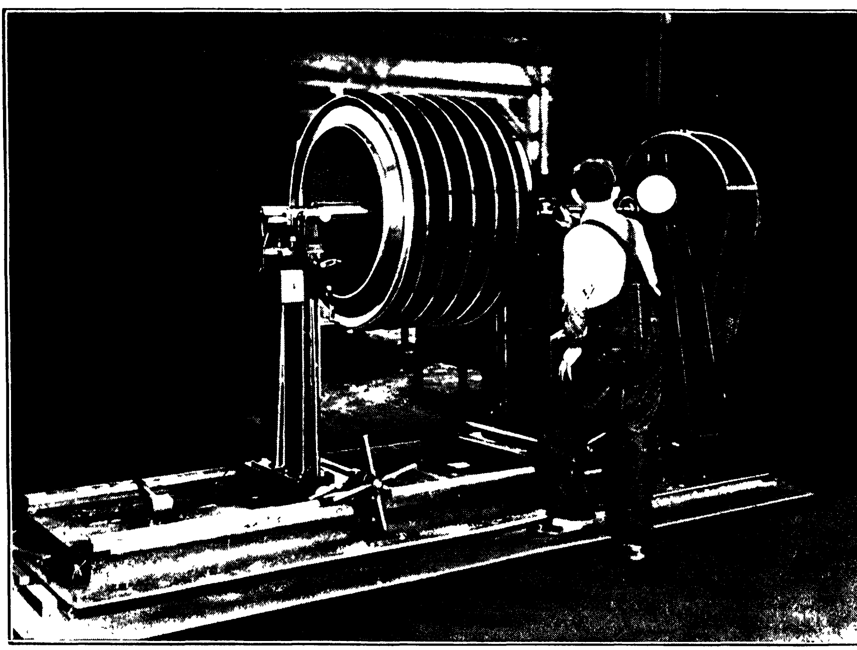


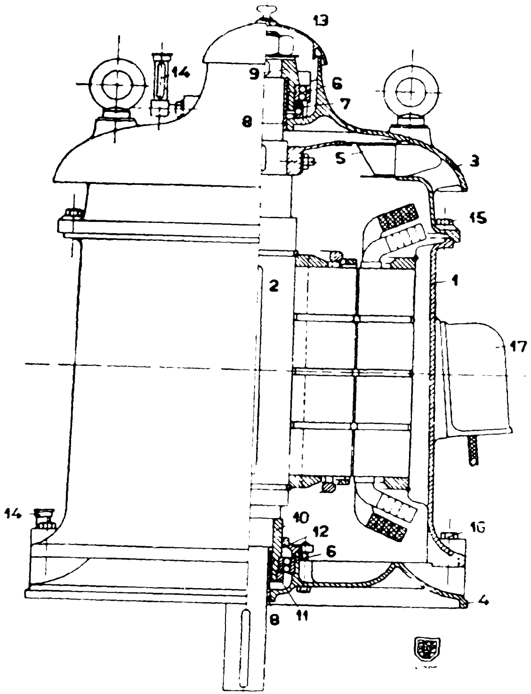
Fig 544.—Centrifugal Characteristics.



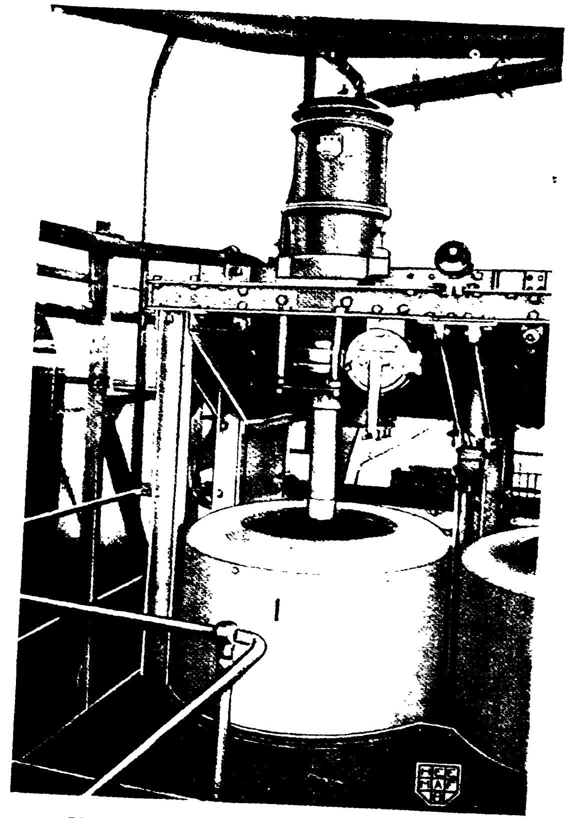
Werkspoor Rapid Cooling Crystallizer.
(*A. & W. Smith & Co., Ltd.*)



Dynamic Balancing Test on Special Machine of a Centrifugal Basket.
(*Squier Manufacturing Co., Inc.*)



Elevation and Cross-Section of S.K.A. Motor for Centrifugal Drive.
(*Heemaf N.V.*)



Vertical A.C. S.K.A. Motor for Direct Centrifugal Drive.

The average time thus can be given only as :

High grades	2·5 to 6 min.
Fore-runners	5 to 6 min.
Affination	5 to 6 min.
Low grades..............	6 to 10 min.
Molasses sugars	12 to 20 min.

With the application of the electrically-driven centrifugal a better insight into the power consumption has been obtained, as the power input in watts (or kilowatts) can be measured by recording instruments. In *Fig.* 544 the *Centrifugal Characteristics* are given for five different machines :—

Curves 1.—*Three-phase A.C.*, 8/16 *poles*, 900/450 *r.p.m.*, 40/10 *h.p.*, *direct connected to* 48 *in. self-discharging centrifugals.*

The centrifugal reaches maximum speed in 50 seconds but the peak power input is for about 20 seconds over 1000 per cent. of the normal power. Regenerative braking brings the centrifugal speed down in about 20 seconds to 450 r.p.m.; a further 10 sec. of hand braking will bring the machine to complete rest.

Curves 2.—*Same data as for Curves* 1, *but the electric motor is connected by a slip clutch to the centrifugal spindle.*

The current peak is reduced about 25 per cent. and the motor attains a speed up to 550 r.p.m. very quickly, the basket lagging behind for about 25 sec. Regenerative braking, thus supplying current to the net, is applied as in the former case, for which a special clutch construction is required. This arrangement thus consumes less peak power.

Curves 3.—*Three-phase A.C. motor, connected by slip clutch to a* 48 *in.* × 24 *in. self-discharging centrifugal.*

The total cycle is 178 sec. against 145 for the previous ones. The total time is divided as follows :—

	Seconds.
Starting	8
Charging	8
Purging	44
Washing	12
Drying	58
Hand braking	48
Total	178

Curves 4.—*Three-phase A.C. motor, connected by slip clutch to a* 40 *in.* × 24 *in. self-discharging centrifugal.*

The tests are made on first sugars of a raw sugar factory and the two-speed performance of the motor is clearly seen. The peak of the power input is very favourable. The cycle is sub-divided as follows :—

	Seconds.
Starting	10
Charging	10
Purging	80
Drying	45
Hand braking	50
Total	195

Curves 5.—Three-phase A.C. motor, connected by a slip clutch to a 36 in. × 18 in. centrifugal.

The acceleration takes about 2 minutes, but the peak of the power input is only 250 per cent. of the normal power. Moreover, an average power factor of 0·85 is obtained during acceleration, against 0·75 during normal operation. The total cycle is about 6 minutes, which is still a good practical figure.

From these five curves the average performance of the centrifugal motor can be learnt.

The *power input for a battery* of centrifugals must be based on the fact that 50 per cent. of the units are accelerating and the rest running normally. The acceleration power can be taken as three times the normal power input. For a battery of x centrifugals this will result in :—

$$HP_{batt.} = HP_{norm.} \times \frac{x}{2} + HP_{norm.} \times \frac{3x}{2}$$

$$\text{or } HP_{batt.} = HP_{norm.} \times 2x \quad \ldots\ldots\ldots\ldots (147)$$

in which : $HP_{batt.}$ = Total power input for the battery.
$HP_{norm.}$ = Normal power input for one centrifugal of the battery.

The normal power input can be taken from the following table :—

Centrifugal.	H.P.	Centrifugal.	H.P.
30 in. × 16 in.	2·5	42 in. × 20 in.	5·0
30 in. × 18 in.	3·0	42 in. × 24 in.	5·5
36 in. × 18 in.	3·5	48 in. × 20 in.	6·0
40 in. × 20 in.	4·5	48 in. × 24 in.	7·0
40 in. × 24 in.	5·0		

For accessories like the mixer and the sugar conveyor, about 15 per cent. additional power is required. For belt-driven centrifugals another 10 per cent. has to be added for the back shaft and belt drive.

Froth, i.e., the scum formed on massecuites when air has entered, will not be separated in the centrifugals and remains inside the basket as a layer on the sugar.

Incidentally, fine-grained sugar should not be accelerated too quickly when in the centrifugal basket, as it may pack together and prevent the separation of the mother-liquor.

2.—Types of Centrifugals.

The *suspended or over-driven centrifugal* is now exclusively used in new cane sugar factories ; the under-driven type is to be found only in refineries, being a special design for forming sugar slabs for turning out cube sugar.

The suspended centrifugal, invented in U.S.A. by WESTON in 1867, has the vertical spindle supported in a heavy ball or roller bearing at the top. This bearing is arranged in a conoidal rubber buffer, which will allow slight oscillations and also let the running centrifugal rotate about its true gravity axis. The charging of the centrifugal should be done at about one-fourth of its operating speed and any heavy swing of the basket through uneven charging has to be avoided, as the basket otherwise will rub the monitor case.

In *Fig.* 545 the general arrangement of a battery of *Suspended Belt-driven Centrifugals* is shown. The centrifugals are mounted on an elevated platform, which has sufficient height to allow the sugar scroll conveyor *a* and the run-off gutter *b* to be placed above floor level ; excavations, which may fill with molasses, are thus avoided, and the curing department will be kept clean with less effort.

The driving or back shaft *c* runs behind the whole battery, each centrifugal being provided with its own driving pulley and friction clutch. As regards power input, there is the favourable circumstance that the driving pulleys of the running centrifugals have in them a considerable amount of flywheel inertia, which is partly consumed during the acceleration period of the starting stage. There will be practically no peaks in total power consumption and a smooth running is assured.

As the driving wheel is located in the vertical plane and the driven pulley in a horizontal one, the belts have to be guided over idlers *d* and the pulleys have to be well outlined, as the belts otherwise will run off or only cover a part of the wheel periphery and thus become subject to premature wear. The belt ends are often jointed by steel lacing having pins with lugs of the width of the belt. For very high speeds, endless belts are used, but the idlers then have to be designed in such a way that they can take up the stretch of the belt.

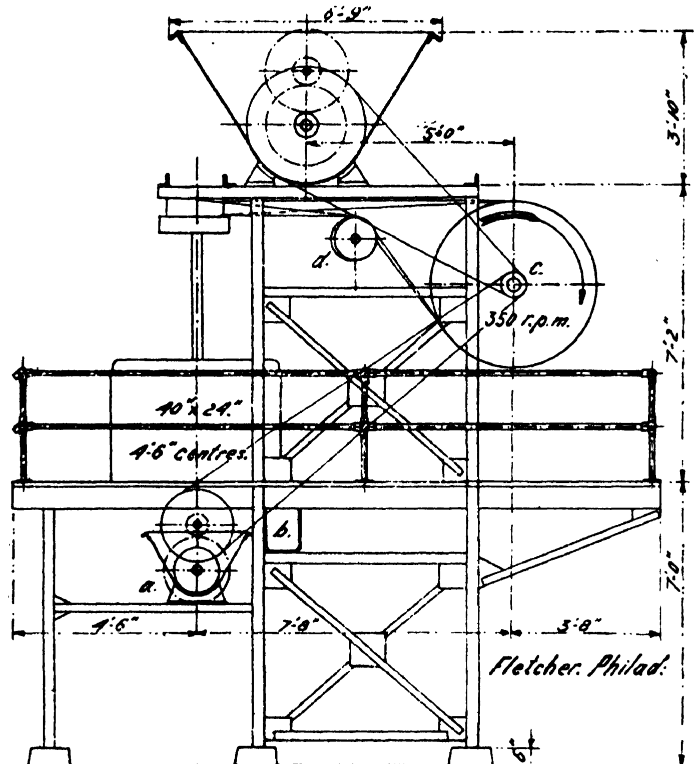


Fig. 545.—Suspended Belt-driven Centrifugals.

Belt-driven machines are very reliable in operation and have found a wide application. From a Report of the Hawaiian Sugar Planters' Association of 1931, there were, of 796 machines reported :—

661	belt-driven	83 per cent.
124	water-driven	15·5 ,,
11	individual electrically-driven	1·5 ,,

The distribution as to size and application for first sugars or low grades was as follows :—[1]

[1] See *Int. Sugar Jl.*, 1931, p. 449.

First sugars	36	30 in. × 14 in.	
	10	36 in. × 20 in.	
	171	40 in. × 24 in.	
	9	not cited	Total 226
Low grades	303	30 in. × 14 in.	
	218	40 in. × 24 in.	
	26	42 in. × 20 in.	
	23	not cited	Total 570

The smaller size centrifugals were formerly applied for low grades, but for medium-size factories the larger machines are also used, thus reducing the number of centrifugals required. The following table comes from the province of Camaguey in Cuba, where 29 large raw sugar factories are located, built or renewed for the greatest part during the last two decades :—

Water-driven	36 in. × 18 in.	20	
	40 in. × 20 in.	6	113 .... 12 per cent.
	40 in. × 24 in.	87	
Belt-driven	36 in. × 18 in.	12	
	40 in. × 20 in.	110	552 .... 59 per cent.
	40 in. × 24 in.	430	
Ind. elec.-driven	40 in. × 20 in.	16	
	40 in. × 22 in.	10	
	40 in. × 24 in.	230	274 .... 29 per cent.
	42 in. × 24 in.	14	
	48 in. × 20 in.	4	
		Total	939 ....100 per cent.

A few off-sizes are arranged under the nearest standard. From this table it will be gathered that the 40 in. centrifugal occupies a predominant position.

The *suspended water-driven centrifugal* has fewer moving parts subject to wear than the belt-driven. Its general arrangement follows the same lines of the belt-driven machine, the backshaft of course not being required. A cup ring or water wheel is mounted on each centrifugal spindle in a casing to receive the waste water. Some designers provide the water motor, which is always built according to the Pelton type, completely separated from the centrifugal spindle, the latter being driven by means of a flexible coupling. This arrangement ensures that the cup ring does not participate in the spindle oscillations.

Two nozzles direct the water jets against the wheel cups, one of them being automatically shut off when the acceleration phase has been completed. The water pressure is from 150 to over 200 lbs. per sq. in.

Operating engineers and superintendents differ in their views as to the efficiency of the water-driven centrifugal. The construction is very plain and wear is small when the jet water contains a certain percentage of lubricating oil and is not contaminated with gritty matter, which obviously will wear the nozzles and cups. Moreover, the water has to be alkaline to avoid corrosion by acidity, and the temperature must not exceed 100°F. but the efficiency and the torque-characteristic are not always accepted as satisfying the demands.

A fixed head for the water pressure is difficult to arrange ; the steam-driven duplex pump can be regulated by a water-pressure governor or throttle, but its steam consumption is not favourable. The flywheel pump, owing to the flywheel inertia, will not readily adapt itself to greatly varying conditions of water delivery, whereas the centrifugal pump has a varying output under varying head. If the latter is electrically-driven, a three-fold energy conversion takes place, thus :—

1. Steam converted in electrical energy.
2. Electrical energy into water energy.
3. Water energy into mechanical energy.

The over-all efficiency in the latter case must be low.

Moreover, the water wheel gives the highest efficiency when the peripheral cup velocity amounts to half the velocity of the water jet, and this does not take place just when the acceleration period starts, and a larger power input over a longer time period is thus required.

The water consumption of a w.d. centrifugal obviously depends upon the prevailing head and the efficiency of the Pelton wheel. One h.p. being equal to 33,000 ft./lbs. per minute, the water consumption per horse-power delivered to the centrifugal spindle amounts to :—

$$Q_{hp} = \frac{33{,}000}{10 \times H \times \eta} \qquad \ldots\ldots\ldots\ldots\ldots\ldots \quad (148)$$

in which : Q_{hp} = Water consumption in Imp. gals. per h.p. per minute.
H = Head in feet (150 lbs./sq. in. = 345 ft.)
(200 lbs./sq. in. = 460 ft.)

One Imp. gal. is equal to 10 lbs. water and the efficiency η for over-all conditions about 0·6. For each h.p. normal power input, there is required at 60 per cent. efficiency 16 Imp. gals. per minute for 150 lbs. and 12 Imp. gals. per min. for 200 lbs. water pressure.

More recently, increased resort has been had to *individual electrically-driven centrifugals,* which, like the water-driven centrifugals, do not require any power transmission as compared with the belt-driven machine, and a good torque characteristic is obtained, which is shown in the rapid accelerations in *Fig.* 544. The current input on these motors has proved heavy, but a very short cycle results, which is in accord with our present-day intensification of the centrifugal station. Regenerative braking can also be applied, so a part of the otherwise wasted power is recovered.

Originally compound motors for direct current were used, allowing for a big range in speed, but in a cane sugar factory constant speed is required or is allowable for most of the electric motors fitted, and so three-phase alternating current has come extensively into use and centrifugals are now also driven by the same current.

The centrifugal motor has to be started and stopped at frequent intervals, and complicated switchgear with resistances and the like should be avoided. The slipring A.C. motor, therefore, has not been adopted to any great extent and *squirrel cage motors,* which are switched across the line, are now almost exclusively used. The normal squirrel cage motor has a large starting current, as may be seen from the characteristics of *Fig.* 544, and it should be recollected that these curves indicate the current input in the motor and not the exact power delivery to the centrifugal spindle. A part of the starting current is converted into heat, for which reason centrifugal motors must be efficiently ventilated, that is, air-cooled.

The torque of the normal squirrel cage motor during the stage of acceleration is not very favourable and improvements have been effected by arranging more resistance in the rotor bars, but less efficiency and a reduced power factor may be the result. By using slip clutches, which are of the centrifugal type, the requirements regarding the starting torque are changed. The clutch torque varies as the square of the speed and directly as the weight of the friction blocks. Generally, a lower current peak will result from using a slip clutch.

A special design is now on the market, where the high resistance bars of the rotor of the squirrel cage are arranged in a fan-like form on the upper part of the rotor, to achieve rapid heat transmission at this point for continuous operation of the centrifugal.

In addition, the special double squirrel cage motor, as explained in Chapter IX, *Fig.* 268, has found an increasing application for centrifugal drive, generally by means of a friction clutch. A high starting torque and a low starting current are inherent features of this motor. As there is no insulation in the rotor, the construction is very simple and reliablo, as is the switch gear for operation with all squirrel cage motors.

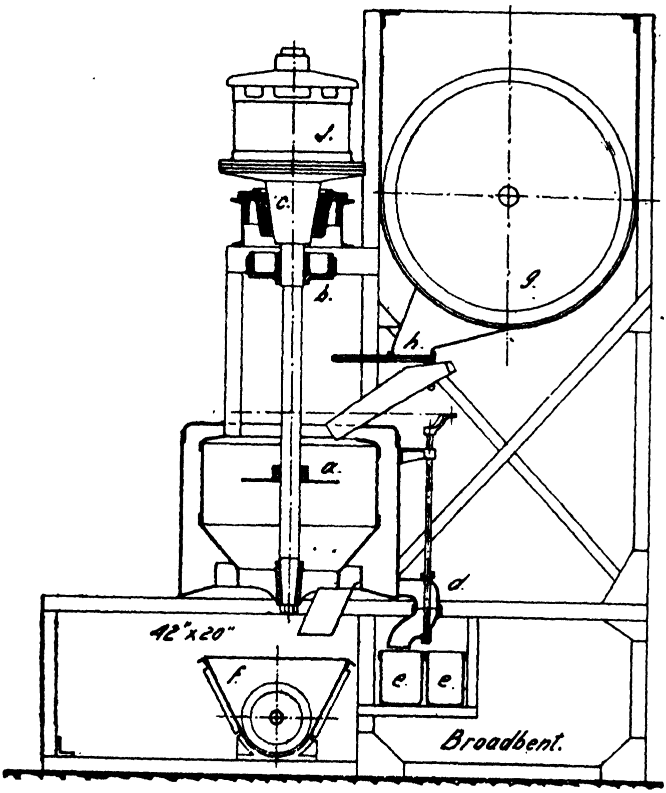


Fig. 546.—General Arrangement of Electrically-driven Centrifugals.

The motor shaft is supported in ball bearings and the upper one is a vertical thrust bearing for supporting the rotor weight.

Two-speed arrangement in the proportion of 1 to 2 is quite practicable through pole switching. The number of revolutions for A.C. depends on the frequency and the number of poles, as tabulated in Chapter IX (page 246).

In *Fig.* 546 is shown the general arrangement of *Electrically-driven Centrifugals.* The electric motor *j* is connected by a fixed coupling to each centrifugal spindle and thus oscillates with the latter. Other designers have the motor firmly mounted on the frame and the connexion between motor shaft and spindle has to be of a flexible design.

The conoidal rubber buffer *c* carries the main bearing housing, and at *b* a hand-operated brake is arranged.

The basket is of the *self-discharging type*; the massecuite being charged on a disc *a*, mounted on the spindle and is thus thrown out radially through the centrifugal force. A bottom valve thus is not required, as the dry sugar will drop as soon as the centrifugal slows down. The bottom inclination for high grades is generally 45° and for the more sticky low grades 60° with the horizontal.

The discharged sugar falls into a screw ribbon conveyor *f* and the run-off from the monitor case is led by a swivel spout *d* into one of the gutters *e*, one for the normal run-off and the other for the wash of higher purity. The mixer *g* is U-shaped, but cylindrical mixers are also frequently used, as the stirrer will prevent any caking of the massecuite against the mixer walls. Lumps of caked massecuite are not desirable in proper centrifugal operation.

The massecuite valve *h* has a tumbling gutter, which will collect the after-drippings when in raised position. The distance between mixer wall and valve should be as short as possible for high grade or white sugars, as the massecuite easily "freezes."

Another type is the *steam-driven centrifugal*: here a low-geared steam turbine is attached to the upper part of the spindle. It is used in small or moderate sized raw sugar factories and variable speed is easily obtained. Direct power generation without intermediate prime movers or transmission results. This type of centrifugal has, however, not found any wide application as compared with the other types mentioned.

3.—Details of Machines.

The *basket* is of brass or steel, reinforced by hoops. Only the very best materials are used for its construction. The walls are perforated and the basket is attached to the spindle at the lower end by a hub and arm piece. Proper equilibration is of importance for smooth running.

The draining capacity of a given centrifugal basket depends upon the perforations, and as these will weaken the basket resistance a happy medium has had to be adopted by the manufacturers.

When the draining capacity of an existing centrifugal is not sufficient, it can be improved by applying a *Special Grid Lining* as shown in *Fig.* 547. The holes in the basket wall 1 are countersunk and a square brass grid lining 2 is riveted on the inside. On this a perforated copper lining 3 with about 25 holes per sq. in. is laid, followed by a wire gauze 4, on which is fixed the inner lining 5 of brass or copper with fine perforations. An improvement up to 30 per cent. in draining capacity has been reported.

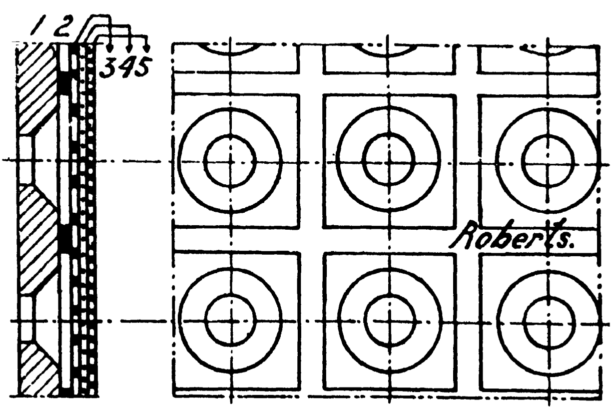


Fig. 547.—Special Grid Lining.

The *Main Bearing* of a centrifugal is indicated in *Fig.* 548. It is composed of three ball bearings *a* and *c* for a lateral thrust of 9050 lbs. and *b* for a vertical load of 5060 lbs., to be fitted to a 40 in. × 24 in. centrifugal. The ball bearings *a* and *c* are sometimes replaced by roller bearings.

The nut *e* on the spindle has to be well tightened and as the centrifugals run clockwise, the nut threads have to run contrary. A loose nut will cause the spindle to vibrate and so damage the conical bore. At *d* on the cap of the housing a grease cup is provided.

A *Spindle Head* for a belt-driven suspended centrifugal is shown in *Fig.* 549. The spindle *a* is supported by a ball or roller bearing as shown in *Fig.* 548, the bearing housing *b* resting in a rubber buffer, so that it can follow the slight oscillations of the spindle. These rubber buffers wear and have to be replaced, but usually only after many years of work. The buffer casing is suspended at both sides at *x*. The driven pulley *d* is centrally located on the mid of the bearing, so that the spindle oscillations will not have any influence on the belt drive or vice-versa. A spherical friction clutch *e* brings the spindle to rest, when applying the handle *f* of the bilateral bell crank, interconnected by the hoop *g* which runs clear of the spindle.

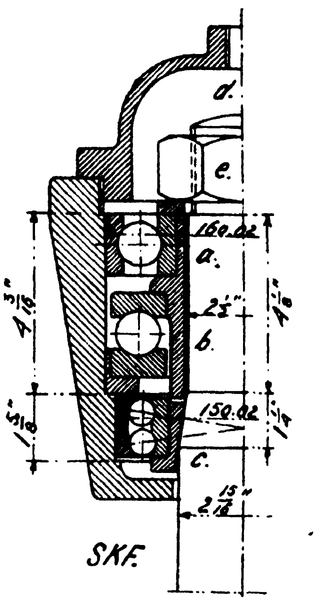


Fig. 548.—Main Bearing.

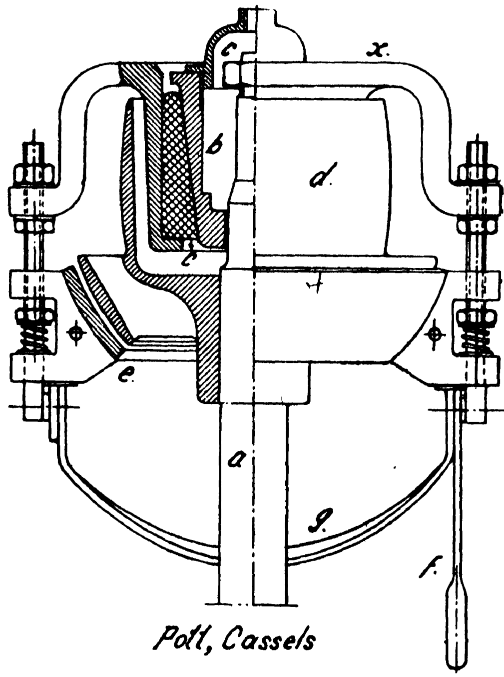


Fig. 549.—Belt-driven Spindle Head.

The driving pulleys on the back shaft ride loose on it in brass bushings. A friction clutch acts inside the rim of the pulley, this clutch being composed of a set of two arms, firmly keyed and bolted to the back shaft. The friction slippers inside the rim of the clutch, faced with leather or friction lining, are engaged or disengaged by means of a pair of toe levers through rods and flat springs attached to a moving sleeve on the shaft. The slippers are under centrifugal force and can be very well adjusted for smooth acceleration of the centrifugal, without causing undue stresses to the belt.

The author has supplied centrifugals with *Double Rim Friction Pulleys* as shown in *Fig.* 550. These pulleys are for centrifugals 950 × 450 mm. (about 38 in. × 18 in.) with the back shaft running at 265 r.p.m. The belt rim *a* has a flange to prevent the belt slipping off on the friction side. The inside rim *b* is for the friction slippers proper and although air cooling is very efficient with these clutches, the rims may get hot through repeated performance of engaging and disengaging. A better heat radiation and less effect on the belt results with this construction. The inertia of the wheel rim may require a little more braking power, but it will be of assistance for smooth running, when adjacent centrifugals have to be accelerated.

The construction is in two halves and although a one-piece arrangement is more usually applied, the latter has the disadvantage that an intermediate pulley can only be removed by dismounting the others between this one and the end of the shaft. The shaft has to be removed as well in such a case.

The drum type friction brake is also used for belt-driven centrifugals, with wide drums of about one third the pulley diameter.

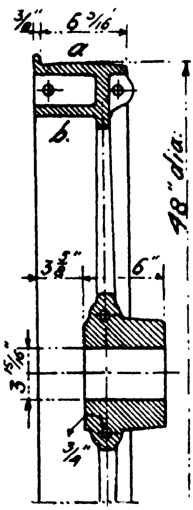


Fig. 550.—Double Rim Friction Pulley.

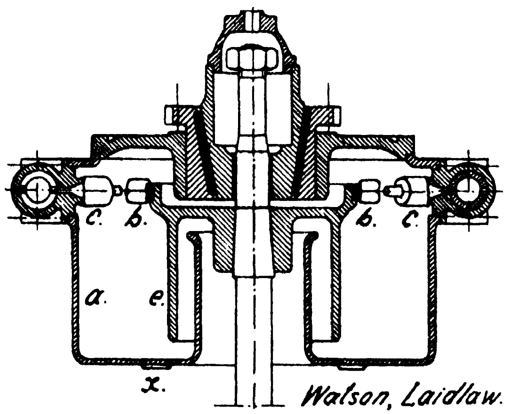


Fig. 551.—Centrifugal Water Motor.

In *Fig.* 551 a *Centrifugal Water Motor* is arranged in the cast iron casing *a*, which is mounted on the centrifugal frame at the supports *x*. The water is charged through two nozzles *c* against the cups *b* of the cup ring, which in turn is mounted on the brake drum *e*, the latter being keyed to the spindle. The main bearing and conoidal rubber buffer are of the customary type. The waste water is withdrawn from the casing by a 4 in. outlet pipe to the waste water gutter and thence to the suction pit or tank for the pump.

One of the nozzles is cut out automatically as soon as the acceleration of the basket has been completed and this is done with some designs by a centrifugal governor, or in others by the rise of the waste water in the casing, which has a restricted outlet that cannot cope with the total water output of the two nozzles. The waste water, of course, should not be allowed to reach the cup ring, as it would cause considerable friction to the running wheel.

4.—Centrifugal Data.

A number of data about centrifugals are to be found below.

The standard *basket perforations* are $\frac{3}{16}$ in. dia., having a pitch of $\frac{7}{8}$ in. in both vertical and horizontal directions.

The *centrifugal linings* starting from the basket shell are generally :—

1. Plain iron gauze, 4 mesh.
2. Plain brass gauze, 7 mesh.
3. Perforated copper sheet, 400 holes per sq. in., 0·027 in. diameter of holes, having No. 21 B. & S. thickness.

The latter may be replaced by 625 holes per sq. in., 0·020 in. dia. or by taper slot perforated plate, having slots $\frac{5}{32}$ in. long, 0·012 in. wide on the inside, 0·028 in. on the outside, and 0·032 in. thickness of the plates. Twilled or spiral woven linings are sometimes used, but not frequently, in raw sugar factories. The linings should overlap in the direction of rotation. For mechanical unloaders, a staggered interlocking joint is sometimes provided.

The *average capacity of different sized baskets* is as follows :—

Size	cub. ft.	lbs. dry sugar.
30 in. × 18 in.	3·6	225
36 in. × 18 in.	5·0	310
40 in. × 20 in.	7·5	470
40 in. × 24 in.	9·0	560
42 in. × 24 in.	9·4	590
48 in. × 24 in.	12·8	800

The *gravity factors* as per formula (145) are :—

Diameter in inches.	
30	565 at 1150 r.p.m.
36	512 at 1000 ,,
40	566 at 1000 ,,
42	538 at 950 ,,
48	553 at 900 ,,

The *mixer capacity* for 40 in. × 24 in. centrifugals ranges between 100 and 190 cub. ft. per centrifugal and for 48 in. × 24 in. is about 200 cub. ft.

The *power per battery*, as recorded from plants in actual operation, is tabulated as follows :—

Belt driven.	Size.	Power.
6	40 in. × 24 in.	75, 80 and 125 h.p.
7	,,	75 h.p.
8	,,	100 h.p.
10	,,	90 h.p.
12	,,	100 h.p.
10	38 in. × 18 in.	110 h.p. by engine 22 in. × 16 in., running at 75 r.p.m.

(This last engine drives the centrifugals, mixer, screw conveyor, magma pump, eight crystallizers, one molasses pump, one grasshopper and one sugar elevator).

Water driven.	Size	
12	40 in. × 24 in.	duplex pump 25 in. × 9 in. × 30 in.
6	40 in. × 24 in.	duplex pump 22 in. × 12 in. × 24 in.

The quantity of wash water varies and may run as high as 8 per cent. on the dry sugar weight in the basket. Superheated steam up to 660°F. (350°C.) has shown no effect on the colour of the white sugar, when oil-free and not applied for over 12 minutes.

Automatic wash water appliances are frequently used, so as to avoid excess consumption of wash water.

Insulated centrifugals for steam drying have resulted in saving about 25 per cent. in steam and 16 per cent. in time.

CHAPTER XXX.

SUGAR CONVEYING AND DRYING.

SUGAR CONVEYORS — SUGAR DRYERS — SUGAR SCALES.

The sugar discharged from the centrifugals is not totally dry, but contains about 0·5 to 3 per cent. moisture, depending upon the quality of the sugar, or, better, upon the purity and viscosity of the film of mother-liquor surrounding the crystals.

This moisture has a detrimental effect on the keeping qualities of the sugar; micro-organisms will develop more quickly in a moist medium and deterioration of the sugar may result. For consumption sugar dryness is of paramount importance, as it should flow freely from the bowl or spoon.

For raw sugar, drying is not considered essential in all countries, although it is done in some of them; but in Cuba, for instance, the raw sugar is not dried in special sugar dryers. The sugars, which are bagged, are spun hot, i.e., at about 120°F. (50°C.), and washing and steaming in the centrifugals is omitted in Cuba as a rule. The raw sugar contains from 0·6 to 0·8 per cent. moisture, this being considered a safe limit against deterioration.

Because of its hygroscopicity, sugar should not be bagged over 100°F. (38°C.), as warm sugar will easily absorb moisture and on cooling will readily cake or cement in the bag. The cooling is usually effected in the centrifugals and in the conveyors, especially when there is proper aeration, but sometimes special coolers are arranged for to ensure this aeration.

The author has noted 1·4 per cent. moisture in sulphitation sugar, it being washed with about 8 per cent. cold water in the centrifugals. Drying with superheated steam in the centrifugals is frequently done, e.g., in Java, and for good results the vapours should be withdrawn by an exhauster from the monitor cases, the latter being covered with insulating material.

1.—Sugar Conveyors.

The oldest and most widely used type of sugar conveyor is the *screw ribbon conveyor,* it being a U-shaped or V-shaped trough in which a shaft revolves with the screw ribbon attached. The screw actually pushes the sugar towards the discharge end of the conveyor, but this has the drawback that the sugar crystals may be ground between the screw and the shell. A clearance of at least ¼ in. is provided around the ribbon, but that space will easily fill with caked sugar, which becomes hard, and regular cleaning has to be done by scraping.

The capacity of the screw conveyor, which has to be filled to just below the shaft, amounts to :—

$$Q_{min.} = \frac{\pi d^2}{4} \times p \times N \times C_v \times W \ldots\ldots\ldots\ldots\ldots\ldots \quad (149)$$

the symbols signifying :—

$Q_{min.}$ = Weight of sugar transported per minute (in lbs.).
d = Outside diameter of ribbon in feet.
p = Pitch or lead of ribbon in feet (average 1 foot).
N = Number of revolutions per minute of the ribbon.
C_v = Coefficient of fullness and slip (average 0·20).
W = Specific weight of loose crystal sugar per cubic foot (average 50 lbs./cub. ft.).

With the values cited, the formula can be written :—

$$Q_{min.} = 7{\cdot}85\ d^2 \times N \quad \ldots\ldots\ldots\ldots \quad (149a)$$

For sugar the number of revolutions is about 15 per minute, **whereas for magma** or **massecuite** about half the number will apply, so as to avoid excessive slip. The V-trough has a wide mouth and is used in many instances under the centrifugals, not requiring special sugar chutes or gutters. The pitch is generally 1 ft. for spirals 12 in. in diameter and over, and the normal sizes are from 10 to 18 in. diameter.

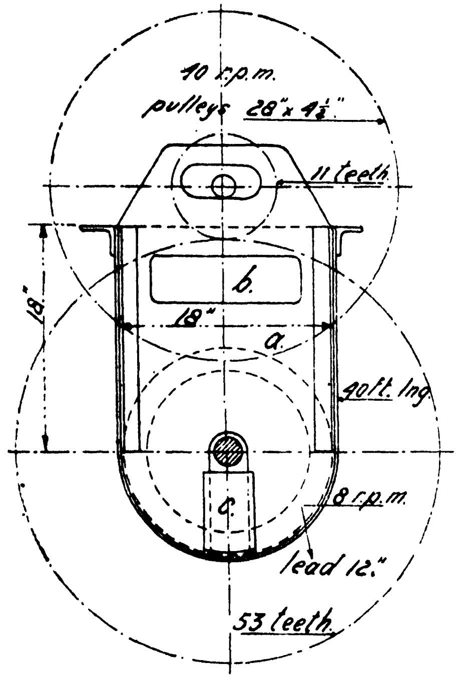


Fig. 552.—U-shaped Screw Conveyor.

In *Fig.* 552 is shown a *U-shaped Screw Conveyor* about 50 ft. in length, installed under a battery of ten 38 in. centrifugals. The low grades from these centrifugals are mixed with syrup, forming a magma of high purity, which is used as "seed" or pied-de-cuite in the vacuum pans. The mixing is done in the screw conveyor by inserting a division plate *a*, which has an opening *b* above the highest point of the ribbon, so it is totally submerged and a thorough mixing is obtained. The syrup is charged in by means of a perforated pipe, laid lengthwise over the conveyor. The slot under the shaft is covered by a plate *c* and the division plate can be removed, for which purpose a hand hole is provided.

In several cases a special mixing trough or *pug mill* is used for this operation. For double curing in the centrifugals, such mixing is necessary and the magma is pumped by a magma or massecuite pump of the reciprocating or rotary type. Some installations are equipped with chain pump elevators.

The shaft of the conveyor is 2½ in. to 3 in. diameter, sometimes made from a heavy pipe section and is supported at about 10 ft. intervals in brass-lined bearings, which are suspended from the ridges of the trough.

A newer conveying device, which came into use some 25 years ago, is the *Grasshopper Conveyor*, of which a sectional view is shown in *Fig.* 553.

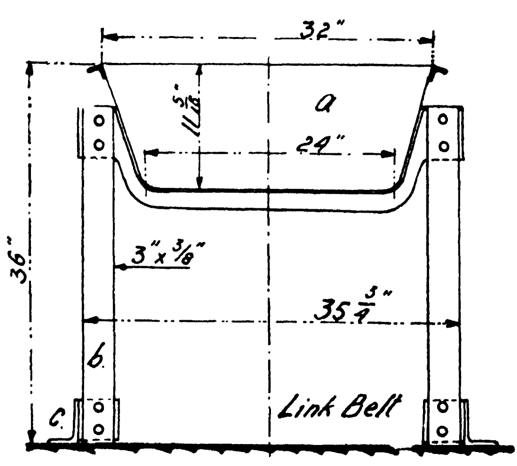


Fig. 553.—Grasshopper Conveyor.

It is composed of a trapezoidal-shaped trough a of light galvanized sheet iron (generally No. 16 B.W.G.), reinforced by angle irons and supported on wooden springs b, which are mounted at about 60° with the horizontal on the base angles c, as further shown in *Fig.* 554.

A connecting rod, sometimes made of wood with a metallic crankpin head, conveys a shaking movement to the trough, the stroke being $1\frac{1}{2}$ in. as an average, and reciprocating at 300—330 r.p.m. Through this rocking movement a particle x of sugar will be accelerated in the direction V_t, tangentially on the rocking radius R. During one half-revolution, assumed to take place in 0·1 sec. (at 300 r.p.m.) the height for unrestricted fall amounts to:

$$h = 0{\cdot}5\,g \times t^2 = 0{\cdot}5 \times 32{\cdot}16 \times 0{\cdot}1^2$$
$$= 0{\cdot}1608 \text{ ft.} = 1{\cdot}93 \text{ in.}$$

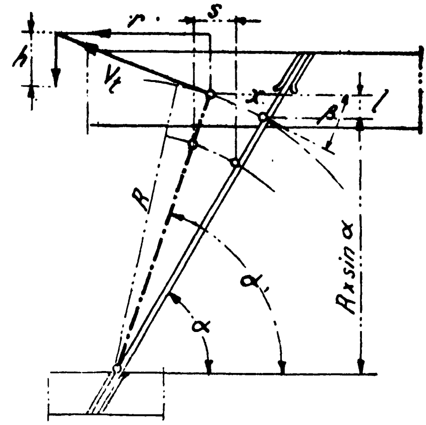


Fig. 554.—Grasshopper Conveyor Springs.

The lift l of the trough amounts to: $l = R \times (sin\ \alpha, - sin\ \alpha)$ and has to be smaller than the practical falling height, to make the grasshopper principle feasible.

Furthermore, the sugar is shot similarly to a projectile with the elevation angle β, and according to the law of dynamics the distance covered r can be derived from the formula :—

$$r = sin\ 2\beta \times \frac{v^2}{g}$$

in which v is the starting velocity $= V_t$. Without too great an error for practical calculations, this velocity can be taken as 1·5 times the mean reciprocating velocity, thus $v = 1{\cdot}5 \times \frac{S \times N}{12 \times 30} = 1{\cdot}5 \times \frac{1{\cdot}5 \times 300}{360} = 1{\cdot}87$ ft. per second, and with β assumed as 30°.

The distance covered r per one revolution thus amounts to 0·108 ft. as per given values, to which the return stroke has to be added, thus $r + S = 0{\cdot}108 + 0{\cdot}125 = 0{\cdot}233$ ft. The capacity of a grasshopper conveyor therefore will be :— $Q_{min.} = F \times r \times N \times C_v \times W$ .... (150)

in which : $Q_{min.}$ = Weight of sugar conveyed per minute in lbs.
F = Section of trough (max. fill height 0·5 ft.) in square feet.
N = Number of r.p.m. of driving crank shaft.
C_v = Coefficient of slip (average 0·5).
W = Specific weight of loose crystal sugar per cub. ft. (average 50 lbs./cub. ft.).

With the dimensions of *Fig.* 553, $r + S = 0{\cdot}233$ ft. and $N = 300$ r.p.m., the capacity amounts to $2 \times 0{\cdot}5 + 0{\cdot}233 \times 300 \times 0{\cdot}5 = 34{\cdot}95$ cub. ft., whereas the manufacturers give it as 30 cub. ft./min. For the size 18 in. × 9 in., the capacity is 18 cub. ft./min. The power consumption is respectively 10 and $7\frac{1}{2}$ h.p.

The grasshopper may be filled in heaps, e.g., when unloading a centrifugal above, for then the upper crystals will have a higher acceleration than the lower ones and an even layer is thus very soon obtained.

The length of a grasshopper conveyer is anything up to 50 ft. and sometimes even more and it provides a very good method of conveying moist sugar.

For transporting sugar from a low to a higher level, a *sugar elevator* is used, consisting of a number of buckets attached to an endless belt. Leather or balata belting is not as frequently used as detachable chains, owing to the adhesive nature of moist sugar. The elevators are provided with either one or two chains; two are to be preferred, as this allows the buckets to be bolted between the two strands of chain, to avoid the sugar caking between the chain links at the locus of the buckets. The buckets are spaced normally 12 to 18 in. and the chain speed is usually 60 to 100 ft. per minute. The capacity of the elevator obviously has to fulfil the equation :—

$$Q_{min.} = B \times N \quad \text{(151)}$$

where : $Q_{min.}$ = Weight of sugar conveyed per minute in lbs.
B = Weight in lbs. of the average bucket load (about 0·6 of the full capacity).
N = Number of buckets filled per minute.

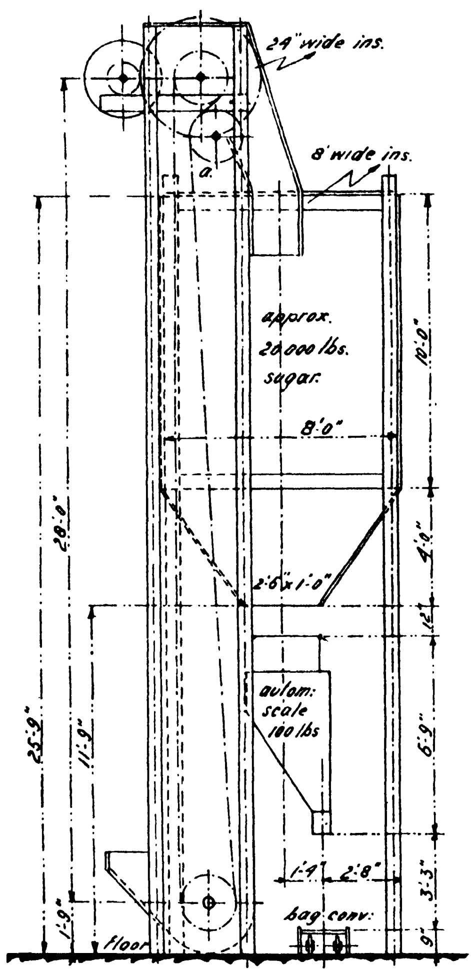


Fig. 555.—Vertical Sugar Elevator with Bin and Automatic Scale.

In *Fig.* 555 is shown a *Vertical Sugar Elevator with Bin and Automatic Scale* from an installation made by the author. The elevator has two strands of chain and the guide wheels *a* at the upper end cause the buckets to empty efficiently. For inclined elevators these guide wheels can be omitted, and sometimes in vertical elevators they are not provided, the discharge depending solely on the prevailing centrifugal force, and a part of the sugar may fall back through the elevator casing.

The drive is always at the top end and the casing can be made locally of wood as shown in *Fig.* 555, or steel plating may be used instead. A pair of inspection doors are required at the lower end for the up and down chains. This elevator has handled about 12 tons of sugar per hour. At the lower end, a spanner device is provided and it should be equipped with a stop, so as not to span the chains in such a way that the buckets may ram the chute bottom. When stretching has taken place through wear in the chains, a pair of links should be taken out.

Dry sugar is sometimes conveyed horizontally by a belt conveyor of normally 12 to 36 in. width, which does not cause any abrasion to the product. Scraper conveyors with flights of angle iron in a wooden trough have also found application, as have tray conveyors from 18 to 24 in. width, with steel trays and chains. The belt conveyor has the lowest power consumption, but is generally used only for moisture-free sugar. For moist sugar the grasshopper has nowadays found a wide application.

2.—Sugar Dryers.

Where moisture-free sugar has to be produced (with a practical moisture content below 0·25 per cent.) sugar dryers have to be installed. The moisture is evaporated by a hot air current, which has passed through a steam-heated air heater. Absorption of the moisture by the air increases with the temperature and the degree of saturation of the entering air also has a bearing upon the result. For tropical cane sugar factories it is safe to assume that the entering air is saturated, so it will not take up any more water vapour at the surrounding temperature. Moreover, it is also safe to assume that the hot air, discharged from the sugar dryer, is not yet completely saturated, but only up to 75 per cent. Accordingly, the better the dryer performance, the higher will be the saturation of the outgoing air.

From the psychrometric tables published in different engineering handbooks, the following table has been copied. The moisture content in lbs. (water) is given per 1000 cub. ft. mixture of vapour and air at 100 per cent. saturation at different temperatures :—

°F.	lbs.	°F.	lbs.
65	0·977	105	3·282
70	1·148	110	3·766
75	1·346	115	4·312
80	1·570	120	4·924
85	1·832	125	5·605
90	2·131	130	6·370
95	2·469	135	7·210
100	2·851	140	8·140

These temperatures will cover the actual range in tropical sugar drying operation.

Let it now be assumed that the air enters the heater at 85°F. and that the hot air is withdrawn at 120°F., then each 1000 cub. ft. of mixture will absorb :—

At 85°F. and 100 per cent. saturation : Moisture content .... 1·832 lbs.
At 120°F. and 75 per cent. saturation : Moisture content
$0{\cdot}75 \times 4{\cdot}924 = 3{\cdot}693$ lbs.

Absorption........... 1·861 lbs.

The capacity of the exhauster per minute has thus to be found from :—

$$V_{air} = \frac{S \times M_s}{M_o - M_i} \times 1000 \qquad (152)$$

in which : V_{air} = Capacity of exhauster in cub. ft./min.
S = Weight of sugar per minute in lbs.
M_s = Moisture content of sugar (say 0·02).
M_o = Moisture content in air mixture at withdrawal from heater in lbs. per 1000 cub. ft.
M_i = Ditto, when entering the heater.

At a higher temperature, the moisture vapours will occupy a smaller specific volume, but the volume of air will increase. In this calculation the total volume of incoming and outgoing mixture is considered equal, but the difference is of small practical interest.

The heat to be supplied to the air mixture must be calculated from the rise in temperature between inlet and outlet and the latent heat required for evaporation of the moisture plus about 20 per cent. for radiation.

The Java Experiment Station has supplied us with very useful data on sugar dryers :—[1]

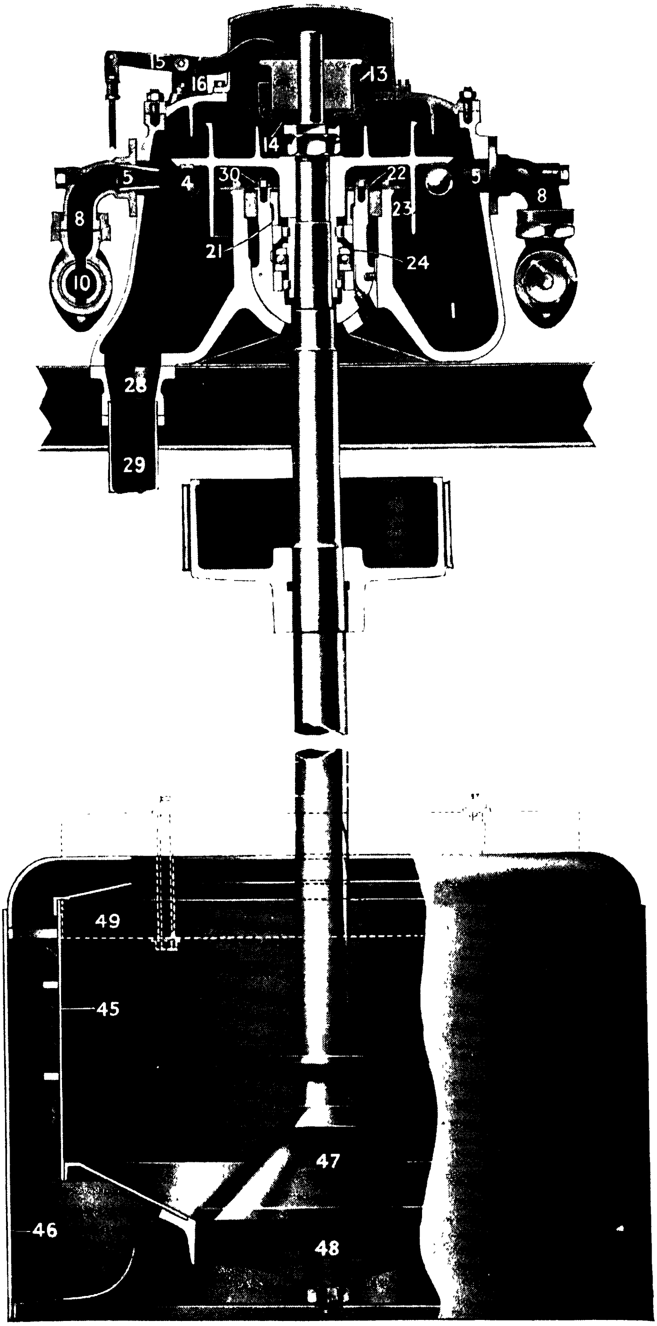
Length of drum	from 18 to 30 ft.
Diameter of drum	from 2 ft. 3 in. to 8 ft.
Number of r.p.m.	from 2 to 27.
Lbs. sugar per hour per cub. ft. drum volume..........................	from 130 to 1300 lbs.
Ditto, average mostly applied	from 250 to 375 lbs.
Scoop dimensions	from 4 to 6 in. wide.
Number of scoops in periphery........	from 12 to 24.
Proportion weight of sugar to weight of air	average 1 to 3.
Cubic feet of air per lb. sugar	average 15 to 45.
Steam consumption in lbs. per 100 lbs. sugar	average 1·8 to 3.
Hot air temp. from heater............	160 to 250°F.

The sugar will leave the dryer at a higher temperature and the author has observed a maximum of 60°C. (140°F.) in actual operation, and therefore means have to be provided to cool the sugar before it enters the bag. In *Fig.* 556 such a *Combined Dryer and Cooler Arrangement* is shown. It has a capacity of about 8000 lbs. sugar per hour with about 2 per cent. moisture when charged.

The sugar enters at *a* in the dryer drum *b*, which is provided with scoops on the inside periphery. The drum revolves on steel tyres which are each borne on two roller supports ; those at the discharge end are provided with double thrust wheels, to keep the drum centrally and axially in its proper place. For conveying the sugar inside the drum when it revolves, the latter is arranged sloping slightly towards the discharge end, the inclination generally being 1 in 20 to 1 in 25. The discharge takes place at *c* and in the same housing are arranged the heating elements *d*, the air being drawn in through the heater and the drum by the fan *e*, thus in counterflow with the sugar. As the air is drawn through the dryer, the pressure inside will be slightly below the atmospheric one, which has the advantage that sugar dust will not be blown out through the joints as is the case with a pressure type. This sugar dust is highly explosive, and any open flame, or even smoking, should not be allowed in the vicinity of sugar dryers. Several accidents have resulted from a neglect of this precaution.

The hot sugar enters the cooler, which is placed below at *f* and traverses the drum *g* in contrary direction as in the dryer above. The construction of the cooler is similar to that of the dryer, the diameter generally being smaller, as air with less temperature has less volume. At the discharge end of the

[1] See the article of L. H. DE LANGEN and H. J. SPOELSTRA, *Het Archief*, 1934, pp. 1113-1124 and 1133-1167, abridged in *Int. Sugar Jl.*, 1935, pp. 267, 308.

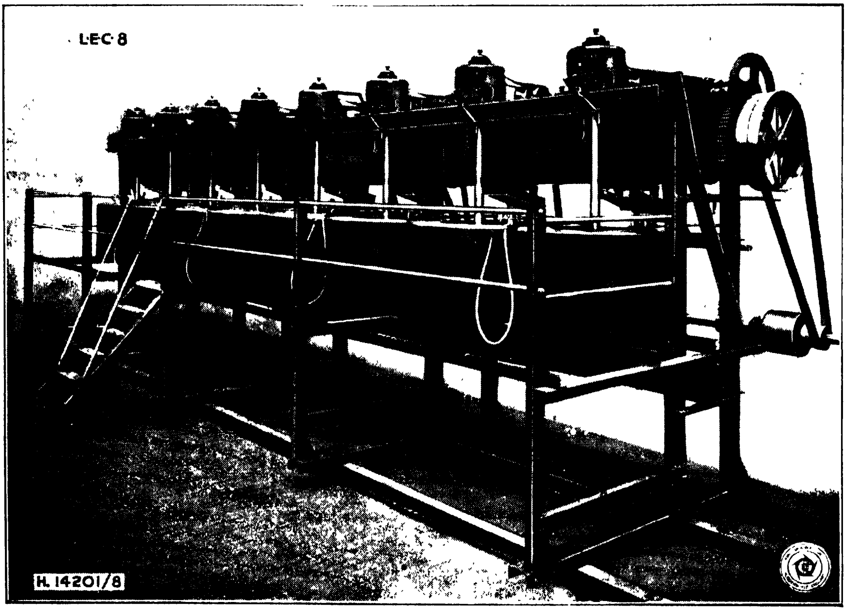


1013

SECTIONAL VIEW OF WATER-DRIVEN CENTRIFUGAL WITH SPHERICAL SEATING.

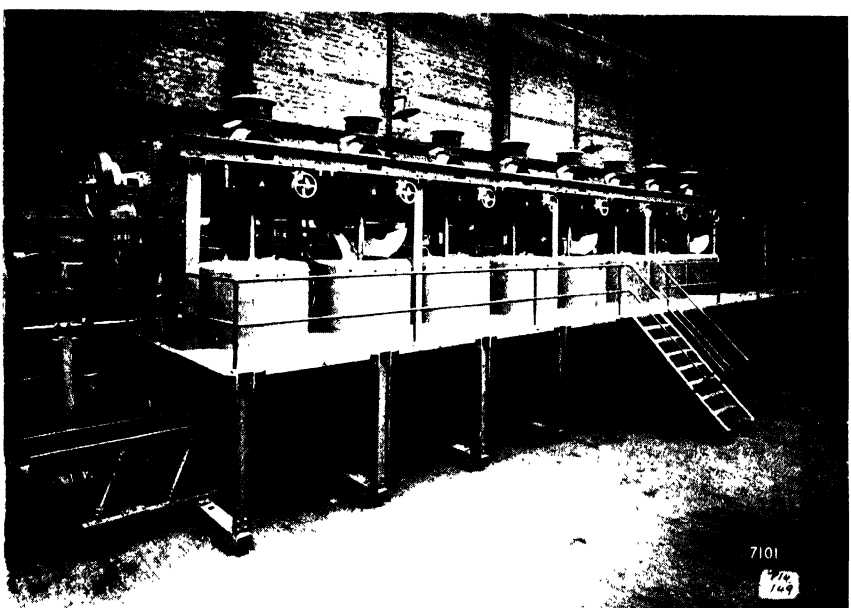
(*Pott, Cassels & Williamson.*)

PLATES 120 & 121.



BATTERY OF 36 in. × 18 in. BELT-DRIVEN CENTRIFUGALS FOR FIRST AND SECOND SUGARS.

(Thomas Broadbent & Sons, Ltd.)



BATTERY OF EIGHT ELECTRICALLY-DRIVEN CENTRIFUGALS WITH GRASSHOPPER SUGAR CONVEYOR.

(Watson, Laidlaw & Co., Ltd.)

cooler drum a circular screen is attached, having wire gauze of $\frac{1}{16}$ in. mesh for fine granulated, supported by a heavier backing gauze. The discharge box *h* has two outlets, one *j* for the lumps or tailings and the other *k* for screened sugar. The air is sucked through the cooler in counterflow by a fan or exhauster *l*. Both drums revolve at from 8 to 12 r.p.m. and the fans are driven from the geared driving wheels around the drums.

The heater elements are made up from gilled tubes, heated by live steam; and a trap is provided for draining the condensate. The air temperature, when leaving the heater elements, should not be over say 110°C. (230°F.), as a higher temperature may give a brownish tinge to the white sugar. The sugar temperature at the discharge end of the cooler is between 42 and 35°C. (108 and 95°F.).

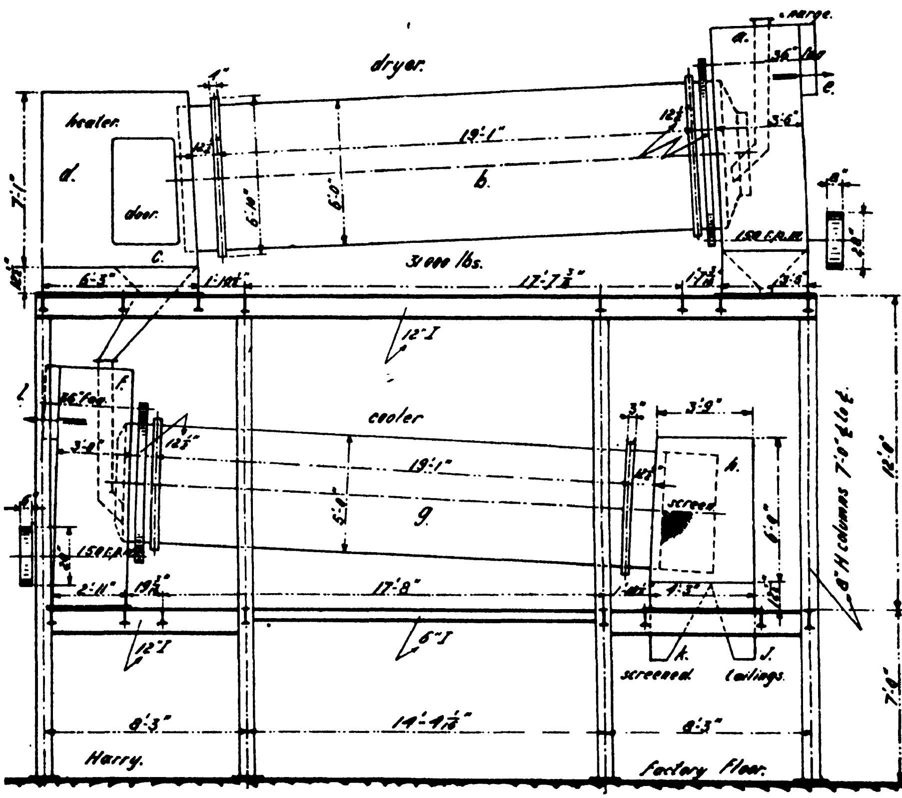


Fig. 556.—Combined Dryer and Cooler Arrangement.

Further drying takes place in the cooler, as the sugar enters at about 140°F. max. and the air is heated to nearly this same temperature. In good installations the dryer and the cooler will have nearly equal shares in the drying performance.

The *Scooping Arrangement* of a sugar dryer drum is shown in *Fig.* 557. There are 12 to 24 scoops fitted on the inside periphery, sloping at about 45° with the radius. For distributing the falling sugar over the whole dryer section and not merely over the right hand half, several designers vary the profiles of the scoops. The scoops are about 2 ft. long and the varied profiles of adjacent rows are staggered. The time the sugar will remain in the dryer depends on the inclination and the drum dimensions, as well as on the number of revolutions, 6 to 10 minutes being an average figure.

Moist sugar may cake inside the dryer drum; since this cannot always be prevented, hammers have to be provided on the outside drum periphery fitted with a rubber head to soften the noise caused. They are operated by a cam, or revolve with the drum. Scraping inside and periodical cleaning have also to be carried out. Several designers make the drums with central shafts, which revolve in heavy duty bearings, and lower power consumption has resulted. The arms which connect the drum to the hub piece on the shaft should not hinder the passage of the sugar.

A *Fan and Heater Arrangement* for a sugar dryer is shown in *Fig.* 558, it being of the pressure type. The heater elements are made of solid drawn copper tubes, No. 17 W.G., having $\frac{3}{4}$ in. outside diameter with corrugated gills of $1\frac{1}{2}$ in. external diameter. The tubes are curved before being rolled into the tube plates of the headers, so as to allow for expansion and contraction. The tube length varies from 2 to 10 ft. There are 15 rows of vertical tubes and 1 to 6 rows of horizontal ones.

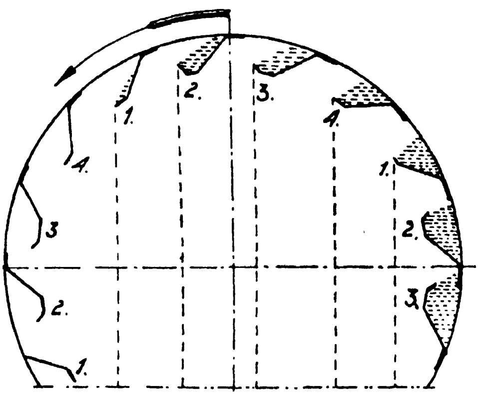


Fig. 557.—Scooping Arrangement of a Sugar Dryer Drum.

The free air area in these heaters varies from 2·36 to 11·85 sq. ft. per unit. Different units can be arranged in parallel, when larger free air areas are required. The average air velocity through the free heater area should be about 1000 ft. per min. With heating steam of 50 and 100 lbs. gauge pressure, the following heating results are obtained :—

HEATER 6 ROWS DEEP.	50 lbs./sq. in.	100 lbs./sq. in.
Air inlet temperature	80°F.	.... 80°F.
Air outlet temperature ..	216°F.	.... 241°F.
Air velocity	1000 ft./min.	.... 1000 ft./min.
Heater resistance	0·290 in. water gauge	.... 0·290 in. W.G.

The sugar dust carried along with the air leaving the dryer has to be separated and special dust collectors must be provided. Not all operators are equally pleased with the performance of these apparatus and a 100 per cent. efficiency is hardly to be expected. There are different systems; sometimes large chambers are built of brick or wood, or the air is led into finely woven cotton bags as is done in the flour industry or in vacuum cleaners; but a widely used type is the *Cyclone Dust Catcher*, shown in *Fig.* 559. This has two inlets and has been used for a combined dryer and cooler installation for 7 lbs. sugar/sec., requiring about 280 cub. ft./sec. total exhauster capacity. The air currents whirl inside the dust catcher and a considerable reduction in air velocity is obtained before reaching the chimney, which extends some 20 ft. above the apparatus. When using the water spray as indicated, the action is fairly good, but without it the sugar dust will reach as far as the hood of the chimney. Some designers put boiling water at the bottom of the dust catcher, so that the steam may moisten the sugar dust. This catcher has been used for finely granulated sugar.[1]

A new type of dryer which has been installed in cane sugar factories and other industries is the *Vertical Multi-tray Dryer*, shown in *Fig.* 560. In a sheet iron casing a vertical shaft is arranged, revolving at sufficient speed to ensure that the sugar which falls on the trays mounted on it will be thrown off by the centrifugal force. The sugar is caught in cones, which deposit it on a revolving shelf below, an operation which is repeated about 15 times. The hot air is blown in at the bottom and the feed and the dust separator are arranged at the top and thus the counterflow principle is adhered to. On the five top trays chains are attached, which sweep the surrounding cones clean

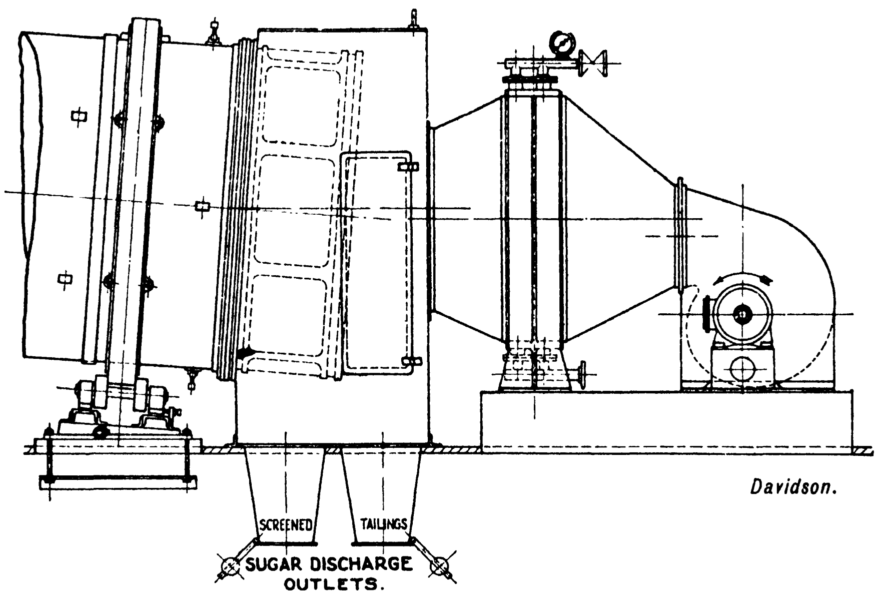


Fig. 558.—Fan and Heater Arrangement for a Sugar Dryer.

of caked sugar. A water spray for cleaning is also provided at the top. The dried sugar falls into the hopper of a bucket elevator below. About 2 h.p. is required for a dryer 4 ft. dia. by 25 ft. 9 in. high and the air saturation has been reported to be 90 per cent. at the exit. The running is silent and the lustre of the sugar (which may be impaired by the prevailing dust which adheres to the crystals in other types of dryers) here remains unimpaired, according to the makers.

To maintain the lustre of the sugar, other types of dryers are in use, some being horizontal with a grid inside the drum so that the fall of the sugar will be less; there is also the *reversing pan type dryer*, the *non-tilting pan dryer* and finally the *turbo dryer*, this last having horizontal discs mounted on a vertical

1 The theory of dust separation is based upon STOKES' Law; see also the article of P. S. ROLLER. *Ind. & Eng. Chemistry*, 1931. p. 213.

shaft, which are scraped off so that the sugar falls through slots on the lower disc. The shaft revolves at from 2 to 4 r.p.m. and the air is efficiently circulated.[1]

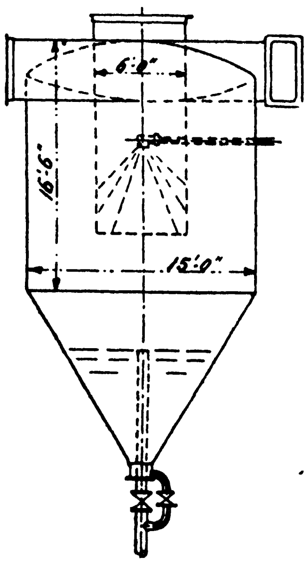


Fig. 559.
Cyclone Dust Catcher.

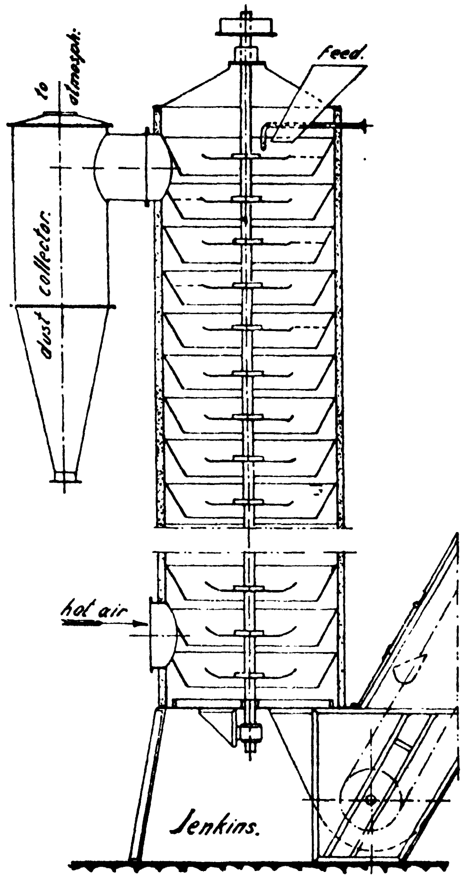


Fig. 560.—Vertical Multi-tray Dryer.

For breaking up the tailings *disintegrators* are sometimes used, these being pin mills with two discs which revolve in opposite directions, or have one fixed and one revolving disc.

3.—Sugar Scales.

Most of the sugar made nowadays is bagged, and other containers, like barrels or bamboo matting, are fast disappearing in the cane sugar industry. Proper weight is essential, not only for the factory control, but also for the commercial end of the enterprise. Hand weighing on good weighing platforms depends upon the human operator and errors in this routine work may arise, especially during the night watch. Automatic weighing has therefore been favoured, and nowadays equipment can be had with a sufficient degree of accuracy for practical use. For those sugar factories which produce white sugar for consumption, small bags of 5, 25 and 50 lbs. will render automatic weighing a necessity, but even for bags of 100 to 325 lbs. the time and labour saved will make the automatic method advantageous.

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[1] See *Facts about Sugar*, 1934, pp. 229.

Dry granulated sugar will not stick to the walls of the hoppers of the scale, but with raw sugar the hoppers should be inverted cones made of brass. In *Fig.* 561 is shown the *General Arrangement of an Automatic Sugar Scale* for bagging 325 Spanish lbs. (one Spanish lb. = 460 grams or 1·015 lbs.), of which the author has seen many in satisfactory operation. A hopper is mounted on top of the scale and a periscope is provided for observing the contents of the hopper. At the bottom of this sugar bin, a coarse wire gauze should be

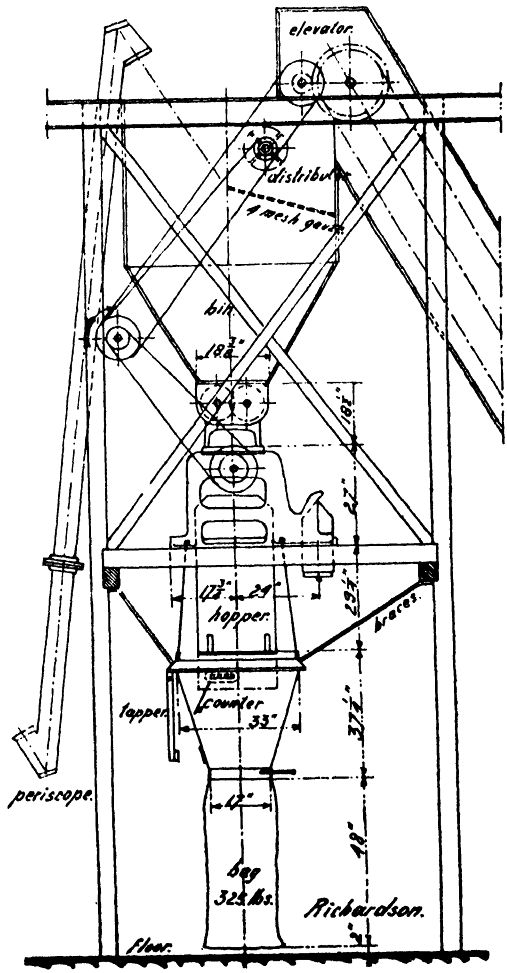


Fig. 561.—General Arrangement of an Automatic Sugar Scale.

provided to prevent bodies, such as nuts, links of elevator chain and the like, which may have fallen into the bin, from interrupting the weighing mechanism. A double set of revolving discs, running at a maximum of 30 r.p.m., will assist in the free flow of the raw sugar at the entrance to the scale; 50 weighings per hour can be achieved, but the author has seen up to 80 bags filled per hour. For granulated sugar, over 500 bags of 25 lbs. can be filled per hour, and 360 bags of 100 lbs., which rates show the ample capacity of these scales.

CHAPTER XXXI.

STORAGE OF SUGAR, MOLASSES AND FUEL OIL.

Sugar and molasses cannot be put direct into the channels of consumption just as produced, and storage has therefore to be provided. Moreover, for the better handling of small quantities, containers have to be available. Fuel oil also is obviously not burnt at the rate at which it arrives at the factory and adequate storage also has to be provided.

1.—Sugar Containers and Stores.

The most widely used container for sugar is the *sugar bag*, made of Indian cotton or jute for capacities from 100 to 330 lbs. For consumption sugars bags of smaller size made of cotton, ranging from 5 to 100 lbs. and sometimes with paper linings, are provided, which help to protect the sugar from moisture, this being the principal cause for deterioration in the sugar. But washed bags have lost the wax coating of the new fibre and are inferior to new ones. Banana fibre is also recommended, but so far has found no extensive application. The bag material has to have a certain tensile strength and test pieces of 2 in. × 8 in. should stand a strain of about 160 lbs. on the warp and about 200 lbs. on the woof in case of good quality bags.

The top seam of the bag is sewn after filling and this has to be done mechanically in case of small size bags for reasons of economy and efficiency. The stitching should be of the non-loosening type under all conditions. Hand sewing is done in case of the larger bags and that for the reason that two-ply stitching is not required, but only a butt-seam, which will result in a saving in bag material. For sewing, good quality hemp twine has to be used. A skilful operator will sew 50 to 75 bags per hour, whereas a sewing machine easily handles 500 small bags in the same time.

Raw sugar is sometimes shipped and stored *in bulk*, and in the U.S.A. concrete silos having 35 ft. dia. by 75 ft. height are in use, possessing a capacity of 260,000 lbs. Open piles of sugar, 25 ft. square and 21 ft. high, sloping at an angle of about 30° have shown less deterioration than when bagged.

Storage bins should have smooth walls, as otherwise from 0·02 to 0·10 lbs. per sq. ft. surface may get lost. *Caking of sugar* may be caused by moisture which has subsequently evaporated or by lack of uniformity of grain ; the gaps thus are more easily filled up and so the sugar is more easily cemented together. Sugar of low polarization is more hygroscopic than sugar of high test, and deterioration is thus more likely.

The absorption of moisture can be prevented if the temperature inside the warehouse is kept a few degrees above the outdoor temperature, 60°F. being considered the minimum ; this is below the temperatures prevailing in tropical countries. A large refinery in the U.S.A. is supplying pre-heated air to two warehouses, containing 140,000 bags of 100 lbs. each, at the rate of 12,000 cub. ft./min. at 0·5 in W.G. Air-conditioning is not applied in tropical countries and conditions become critical when temperatures up to 42°C. (108°F.) prevail during the day time. With this temperature a large moisture content in the air is likely and as soon as the atmosphere cools down a deposit of dew will result. This same phenomenon occurs when the temperature in the warehouse is much above the outdoor one and sweating of the walls and the sugar piled close to these will then take place.

Different views therefore exist as to how a sugar warehouse should be ventilated. One group recommends that the warehouses be ventilated by thorough circulation of air when the dewpoint outside is low, and be almost hermetically closed when this dewpoint lies at a higher temperature, thus indicating more moisture in the atmosphere. Several experienced men do not think it practicable to go by the prevailing atmospheric conditions, and consider it essential to produce the sugar as dry as possible and then to cover it up with waterproof material or tarred paper and have the doors of the warehouse only opened while handling the sugar, and this should be done in the quickest way possible. For inspection, a small doorway should be provided. The author has seen sugar keep well over two years in a closed warehouse, without any ventilation, but the sugar stored had been manufactured under the supervision of a capable factory staff.

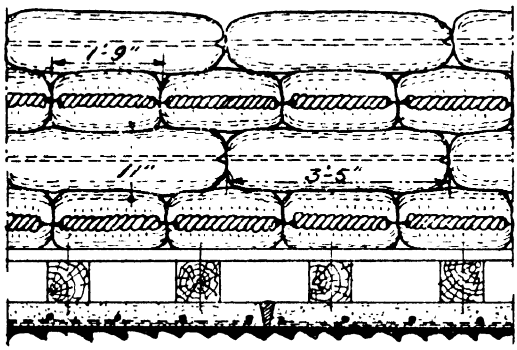


Fig. 562.
Storage of Sugar Bags on a Concrete Floor.

The floors preferably are made of concrete, laid on a good stamped sand filling well above the surrounding ground level. In *Fig.* 562 the *Storage of Sugar Bags on a Concrete Floor* is shown. The concrete is reinforced on the tension (lower) side and expansion joints 1 in. wide and filled with asphalt are provided at about every 12 ft. length and crosswise. Moreover, the top layer of the concrete is made impermeable, to avoid penetration of any eventual molasses or drippings from the bags. On top of the floor are laid wooden beams 8 in. square at about 2 ft. centres, covered by boards about 2 in. × 6 in. and 6 in. apart. The bags containing raw sugar weigh 330 lbs. each and are piled up to 26 rows high. In Java 48 rows have been reported, the bags being of 50 kgs. each.

At every 20 ft. lanes must be provided between the bags, and the sloping of the end rows is about 1 in. for each bag lying above the other. In non-ventilated warehouses, the lanes are not provided and the wooden beams underneath are also omitted, the bags being piled directly on wooden boards which are sometimes laid in double row lattice fashion.

The buildings are made of steel with corrugated galvanized roofs and sides, although brick or concrete walls cause less sweating when the outside temperature is lower than inside. The piles should lie clear of the walls.

The safe floor bearing should be 1200 lbs. per sq. ft. as a minimum, and the space occupied by the bags can be calculated on the basis of a weight of 50 to 55 lbs. per cub. ft. of bagged sugar. Low-grade sugars will drip molasses, and gutters have to be arranged where these molasses can accumulate and be removed from the building.

Sugar bags are transported by different types of conveyors, and a *Roller Conveyor* is shown in *Fig.* 563. It is made up of two strands of malleable iron chain having 3·075 in. pitch, as shown in *Fig.* 564. The ultimate strength of each strand is 9600 lbs. and attachments are arranged at every sixth link for bolting a flat iron 3 in. × 1 in. on top. These flat irons have trunnions on the ends which carry 4 in. cast iron rimmed rollers. The rollers move over flat irons attached to the conveyor body. A piece of hard wood with rounded corners is bolted to the flat irons.

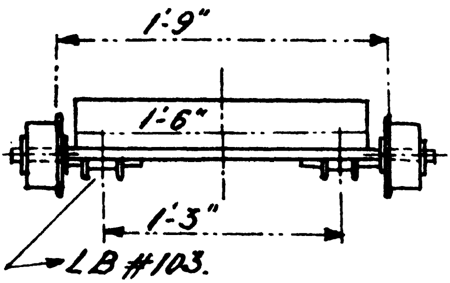


Fig. 563.—Roller Conveyor.

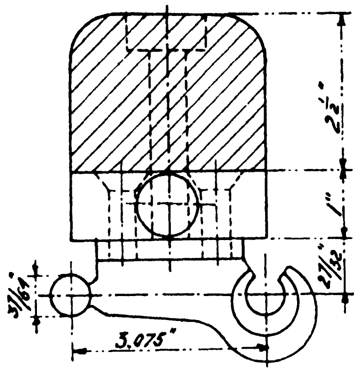


Fig. 564.—Roller Conveyor Attachment.

The *General Arrangement of a Bag Conveyor* in warehouses is shown in *Fig.* 565. The carrier slope is 30° and standard gauge cars can move freely under the conveyor between the two buildings. The brakesman can even walk freely over the top of the cars. The floor level is flush with the platform of the cars, which are loaded by two-wheeled hand carts. The speed of this carrier is up to 90 ft. per minute and the power is supplied by a 12·5 h.p. electric motor. Lintels are provided on the loading side of the building, but the warehouses if possible should only be opened when dry weather prevails.

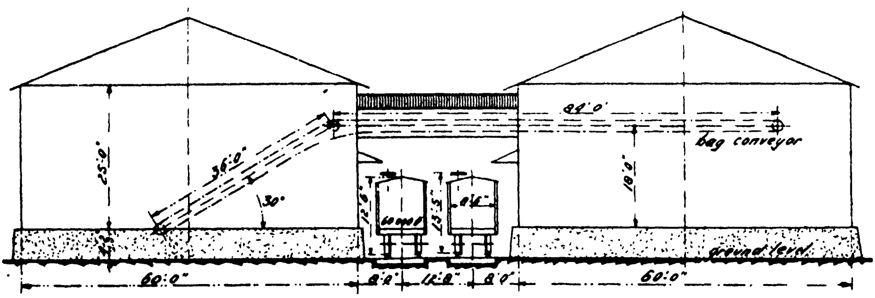


Fig. 565.—General Arrangement of a Bag Conveyor.

To pile the bags from the floor, as the conveyor will deliver the bags by chutes to the floor level, a *Transportable Bag Piling Machine* is very convenient and saves labour. Such a machine for bags of 330 lbs. is indicated in *Fig.* 566 and it will pile up to 600 bags per hour when sufficient hands are available for storing the bags in the proper place in the pile. This bag-piling machine is composed of two conveyors, each having two strands of chains and the bags supported by iron bars 1 in. square with rollers on both ends. A bottom plate is provided so that the bags will not sag between the bars. The bag piler is driven electrically and on the columns of the warehouse plug connexions have to be arranged. The maximum height to which the bags can be lifted is about 30 ft., which is in excess of the average sugar warehouse requirements, with a normal truss height not over 25 ft.

Another type of conveyor for sugar bags is the *Slat Conveyor*, 18 in. to 24 in. wide, the slats of wood being mounted on a roller chain as shown in *Figs.* 120 and 120a of Chapter IV. The upgoing as well as the returning apron is supported on sliding beams.

A novel system for conveying bags has been applied in a large refinery in the U.S.A., it being composed of two 3 in. tubes provided with welded spirals ½ in. thick, and having 5 in. lead. The two tubes are arranged parallel at 8¼ in. centre distance, having brackets at 8 ft. 3 in. intervals, and they revolve in opposed directions at 250 r.p.m. One tube therefore has right hand and the other left hand spirals.[1] Up to 1800 bags of 125 lbs. are conveyed per hour at a speed of about 104 ft. per minute. The power consumption of this *twin spiral conveyor* is 5 h.p. for 50 ft. to 70 ft. length. The bags have been transported up a slope of 30°, and as the threads are polished there is no abrasion wear reported on the sack material.

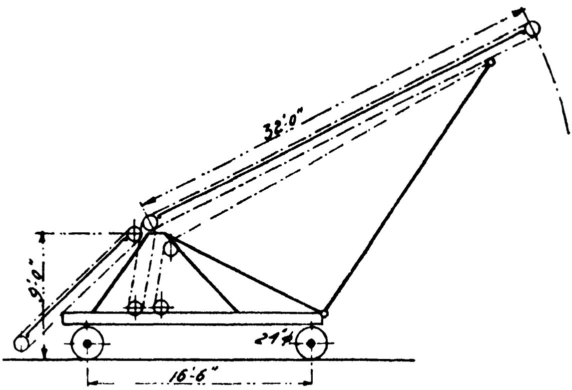


Fig. 566.—Transportable Bag Piling Machine.

2.—Storage of Molasses.

Molasses is stored in tanks, although the author has seen molasses stored in ponds dug in the soil, which should be covered to prevent dilution and the consequent fermentation and foaming. A few standard dimensions of molasses tanks are given below :

Capacity U.S. Gals.		Diameter		Height
250,000	...	42 ft. 3 in.	×	24 ft. 0 in.
500,000	....	53 ft. 4 in.	×	30 ft. 1½ in.
750,000	....	65 ft. 3 in.	×	30 ft. 0½ in.
1,000,000	....	75 ft. 4 in.	×	30 ft. 2½ in.
1,500,000	....	92 ft. 4 in.	×	30 ft. 0¼ in.
2,000,000	....	97 ft. 3 in.	×	36 ft. 0¼ in.

The tanks should be provided with a good sloping roof so as to prevent rain water from mixing with the molasses. In *Fig.* 567 the *General Arrangement of a Molasses Tank* of 180,000 U.S. gallons is shown. In countries affected by cyclones the top rim should be well reinforced as indicated by *a*, or the tank filled with water when the molasses has been shipped. In Plate No. 124 is shown what has happened to the tank of *Fig.* 567, when empty and not properly anchored and reinforced during a cyclone. The roof should be attached to the upper rim and also be properly reinforced under such conditions.

The tank is placed on a high foundation, which can also be designed in reinforced concrete, having columns and a reinforced bearing plate on top. Tank cars can be run alongside and filled by gravity. The dimensions of the tank car for standard gauge allow 8000 U.S. gals. capacity and when filled will carry about 80,000 lbs. net.

[1] See *F.A.S.*, 1928, p. 641.

Raw sugar factories produce about 1·4 to 3 gals. molasses per 100 lbs. sugar manufactured (4 to 7 Imp. gals. per ton cane ground) and sometimes the whole production of a crop is stored, all depending upon shipping facilities or market considerations, as there might be a difference in price between crop time and the dead season.

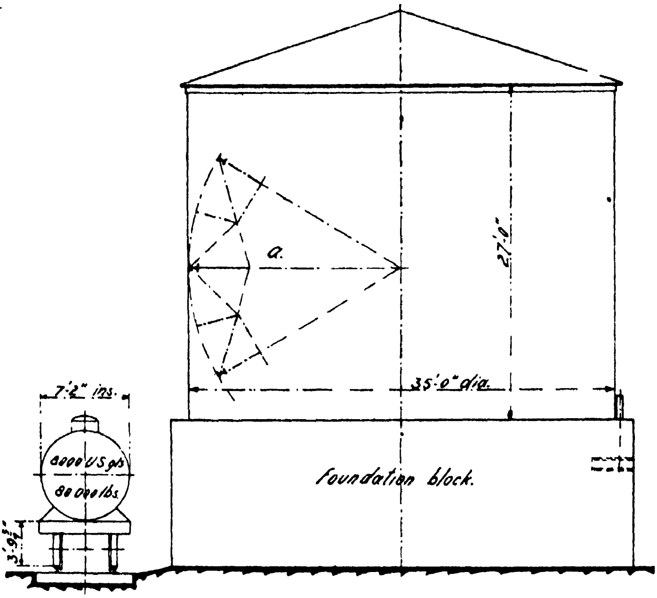


Fig. 567.—General Arrangement of a Molasses Tank.

A gate valve from 6 to 12 in. diameter is arranged at the bottom for discharge, and the connexion between the tank and the valve should be well proportioned, as the author has seen such a valve break off through corrosion, the molasses pouring out in the factory yard after attempts to close the hole with a large wooden plug had failed. A plate could not be lowered inside from the top, as the roof covering at the spot desired could not be removed in time.

Air free molasses weighs about 14·5 lbs./Imp. gal.

3.—Storage of Fuel Oil.

Fuel oil is pumped into the storage tanks and is also pumped from the storage tanks to the factory or to the small fuel tanks for the locomotive service.

The specific gravity of fuel oil is lower than that of molasses and the tanks therefore are generally built up to about 40 ft. high, whereas molasses tanks are usually 30 ft.

The soil bearing for fuel oil tanks, inclusive of the tare weight of the tank, amounts to about 2500 lbs./sq. ft., and for molasses tanks up to about 2800 lbs./sq. ft. This is not an excessive figure and thus only light foundations are required. A sole plate of concrete, well covered with asphalt, will prevent the heavy corrosion that occurs when the tank is placed without foundation on firm soil.

Standard sizes are tabulated below :—

U.S. Gals.		Diameter		Height
250,000	....	40 ft. 0 in.	..	27 ft. 0 in.
500,000	....	55 ft. 0 in.	..	30 ft. 0 in.
750,000	....	66 ft. 0 in.	..	29 ft. 10 in.
1,000,000	....	66 ft. 0 in.	..	40 ft. 0 in.
Barrels.				
5,000	....	40 ft. 0 in.	..	22 ft. 6 in.
10,000	....	55 ft. 0 in.	..	25 ft. 0 in.
15,000	....	66 ft. 0 in.	..	25 ft. 0 in.
20,000	....	77 ft. 0 in.	..	25 ft. 0 in.
25,000	....	85 ft. 0 in.	..	25 ft. 6 in.
30,000	....	85 ft 0 in.	..	30 ft. 3½ in.

One barrel is taken as 35 Imp. gals. or 42 U.S. gals. and the specific gravity lies between 0·85 and 1·00, or per Imp. gal. from 8·5 to 10 lbs. and per U.S. gal. from 7·10 to 8·33 lbs.

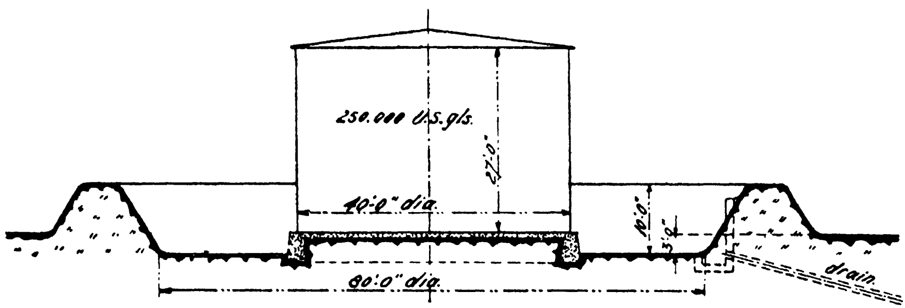


Fig. 568.—General Arrangement of a Fuel Oil Storage Tank.

In *Fig.* 568 is shown the *General Arrangement of a Fuel Oil Storage Tank* of 250,000 U.S. gals. capacity ; the tank is surrounded by an earthen wall in such a way that the annular space will have a volume equal to the tank volume. In the eventual case of fire or explosion, the fuel oil thus cannot spread over the factory yard. Oil tanks should be located at a good distance from the factory buildings. The tank is placed 3 ft. below ground level, the excavated earth being sufficient to build the dam. This practice is permissible where good drainage of the hole is possible. The annular space has to be provided with a cast iron drainage tube of about 6 in. dia., to drain off any rain water that may have accumulated within.

Fuel oil has a flash point around 180°F. according to closed test, and it is therefore not very dangerous ; premiums for fire insurance, therefore, lie within reasonable limits.

Gas-tight covers for fuel oil tanks are not required, but rain water has to be kept out, as it will impair the proper working of the oil burners. Generally, by fuel oil the residue of the oil cracking process is meant ; in some cases the crude oil as it comes from the well is used for firing, but either sort can be stored in the same type of tank.

Underground concrete tanks for fuel oil and molasses are sometimes built. The concrete has to be made impermeable for this kind of work, and air vents provided.

CHAPTER XXXII.

LUBRICATION OF EQUIPMENT.

When metallic surfaces are sliding, rubbing or rolling on each other, microscopic penetration of the metals is taking place and abrasion will result, not only causing heavy wear, but also unnecessary friction, which is equivalent to excessive power consumption. Moreover, the friction energy will be converted into heat, which may heat the metal of the bearing surfaces beyond the allowable limit. Lubrication, therefore, is essential for all those places where metallic or other friction exists.

1.—Fundamentals of Lubrication.

The purpose of lubrication is to insert between the bearing surfaces under friction load, a film of lubricant so that the metallic surfaces will not rub actually on each other, but are separated by the lubricating medium. Instead of metallic friction, the cohesive friction of the lubricant proper, which is only a fraction of the former, prevails ; and thus lubrication of the equipment in any industry becomes a necessity for reasons of economy and for reliable and uninterrupted operation.

As it would be well nigh impossible to compile data about all the lubricants now on the market, only the fundamental requirements which a good lubricant should fulfil are given here :—

1.—A lubricant should possess the necessary fluidity to reach the pores of the rolling or rubbing surfaces.

2.—The surface tension should be low, to ensure proper adherence to the bearing surfaces, and the film formed should not be broken through the pressure on it being too heavy. For high speed and low specific bearing pressure a thin lubricant has to be selected, whereas for heavy specific loads more viscous lubricants will be required.

3.—The lubricant should not lose its lubricating properties under the conditions ruling, e.g., through the heat or moisture of the steam in steam cylinders or the splashing of juice or water in the cane mill bearings.

4.—As the cohesive friction increases with the viscosity, lubricants should be selected with the lowest viscosity possible for the required task.

5.—The lubricant should not carbonize under high temperatures as in air compressors, vacuum pumps and internal combustion engines.

6.—For automatic lubrication, where the same oil re-enters the cycle repeatedly, the lubricant should not lose its body within the pre-determined time of re-filling the bearings or containers.

7.—The lubricant must not contain impurities in suspension, as these will scratch the bearing surfaces. Nor are chemical impurities allowable, such as acids, resins and water, which will attack the metal of the bearing surfaces.

8.—The lubricant should have a high resistance to decomposition, like oxidizing and hardening.

Obviously, there is no single lubricant that can meet all these requirements, and different kinds have to be employed for different purposes.

The usual lubricant consists of mineral oils which have passed the distillation process to eliminate all volatile components. Sometimes they are mixed or "compounded" with vegetable oils or animal fats which have a better adherence property, but are more easily decomposed by the oxygen in the air.

Grease, solidified or consistent oil, made with lime, soda or lead soap, replaces mineral oil in many cases. There is no fixed rule as to where oil or grease should be used, and manufacturers of lubricants should be consulted as to the proper application of their products.

Any penetration of dust into the bearings is necessarily harmful and not only requires excessive quantities of oil to wash the dust away, but it tends to scratch the bearing surfaces.

Lubricating oil should be stored in proper containers, each provided with a hand pump, and the barrels should be emptied into these containers. Batteries of containers, of 50 to 300 gals. capacity each, can be obtained, provided with cradles for easy discharge of the barrels. As lubricating oil is expensive, it will pay the factory store-keeper to keep careful control of the outgoing quantities.

The viscosity of lubricating oil is found by comparing it with the flow of colza oil through a calibrated orifice and the time in seconds required for a certain volume to flow is considered as a basis for the viscosity. The Engler viscosimeter is commonly used and the standard temperature during the test is 20°C.

The flash point is ascertained in a closed receptacle according to the Pensky-Martens method, which is also used for comparison. The average requirements for lubricating oils can be tabulated as follows :—

Lubricating oil used for :	Viscosity in degr. Engler at 20°C.	Flashpoint P.M. °C.	Tarry Matter per cent.	Congealing point °C.
Light bearings, electric motors	10—25	170—220	—	—
Turbines	9—13	180	0·10	below 0
Heavy bearings	20—60	185—220	0·20—0·50	,, —5
Air compressors, vacuum pumps	10—20	200	0·20—0·50	,, 0
Centrifugal pumps	5—10	200	0·20—0·50	--
Locomotive and car axles	25—60	>145	—	below —5
Diesel lubricating oil	8—13 (at 50°)	200	0·20—0·50	,, 0
Cylinder oil for saturated steam	23—45 (,,)	260—320	—	about 0
Ditto for superheated steam	23—60 (,,)	>280	—	,, 5
Refrigerating machinery	5—10	140—150	0·09	below —20

The rate of consumption of lubricating oil varies greatly, depending upon many factors, but a fair average is the following :—[1]

Steam cylinders	0·5 grms./h.p./hour
External engine gear	1·0 ,, ,, ,,
Crankcase	0·5 ,, ,, ,,
Heavy mill bearings	12·0 grms. per sq. ft. projected bearing area per hour.
Total consumption of factory, including agricultural machinery	0·028-0·050 U.S. gals. per ton cane ground.

(1 lb. = 453 grms. and 1 U.S. gal. oil ≃ 3500 grms.)

[1] See D. BURNS CAMPBELL, *Int. Sugar Jl.*, 1933, p. 23.

Oil recovered from drip pans, etc., can be used for inferior lubricating purposes or in the shop for thread cutting and the like. It is good practice to treat this oil in a centrifugal separator, in which the heavier particles in suspension are thrown out. For purifying oil which is in circulation in turbines, high speed or internal combustion engines, centrifugal separation can be fully recommended.

For the crank case of internal combustion engines, having trunk pistons, the oil consumption amounts to from 3 to 5 grms./h.p./hour, this being the total oil consumption.

2.—Mechanical Requirements for Good Lubrication.

The lubricant used should suit the purpose for which it will be employed, but it is equally important that the mechanical design of the bearings and frictional parts of the engine allows for the proper functioning of the lubricant, and therefore a few principles of design are given below.

As a first requirement, there should be a certain clearance between the rubbing parts of the engine, to make possible the formation of an oil film. This clearance should be about 0·01 in. for small size bearings, measured as the difference between shaft and bore diameter and may become as much as 0·02 in. in case of larger sized gudgeons.

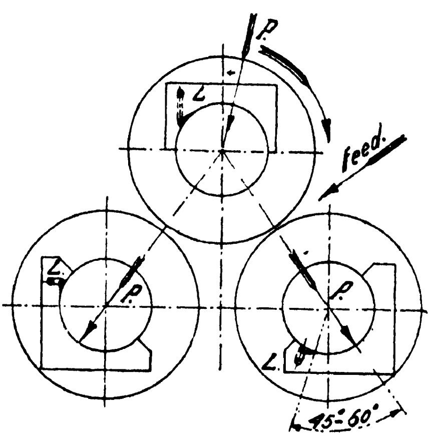


Fig. 569.—Lubrication of the Mill Gudgeons.

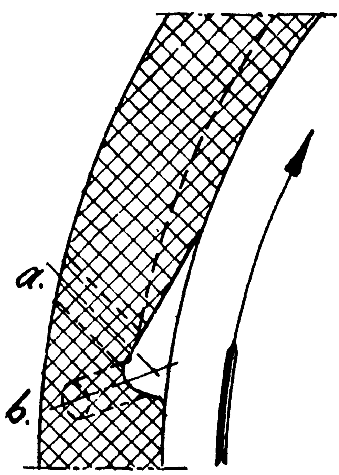


Fig. 570.—Profile of the Lubricating Groove.

Moreover, the lubricant should be applied to the bearing at the locus of reduced bearing pressure, and in such a way that it will be carried to the locus of maximum specific bearing. Sharp edges on the bearing surfaces or the oil grooves will scrape off the oil and break the film, so they should be avoided; but in internal combustion engines with trunk pistons, scraper piston rings have to be fitted on the crank side of the piston, to scrape excessive splash oil back into the sump, as otherwise the oil consumption will increase.

Oil grooves for bearings where dust is bound to enter should be cleaned at regular intervals with hot water or steam. Dust seals will decrease the consumption of lubricant and reduce the wear, whereas oil seals should be placed at those points where the lubricant may flow out of the bearing and thus be lost for use. Grease is also considered to give a good seal against dust entering bearings.

The teeth of gears and crown wheels are lubricated by dipping into slush pans or by the swab method. Totally enclosed gears will require less oil and suffer less wear, as dust or grit is kept out.

The *Lubrication of the Mill Gudgeons* in a cane sugar factory belongs to the category of very difficult problems, and in *Fig.* 569 is shown at which points the lubricant should be applied, the oil grooves being marked *L*, and the maximum prevailing bearing pressures *P*. The oil grooves are arranged from 45° to 60° before the rotation of the gudgeon reaches the locus of maximum pressure. The centre oil feed of the top bearing is therefore not considered suitable for good lubrication and reduced wear.

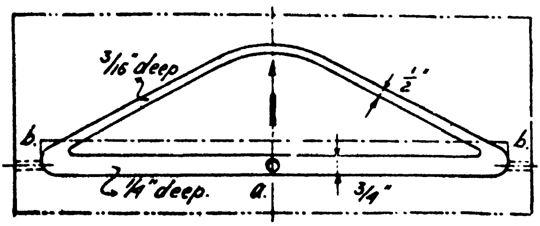


Fig. 571.—Triangular Oil Grooves.

In *Fig.* 570 is shown the *Profile of the Lubricating Groove*, which should be about ¼ in. deep and chamfered in the direction of rotation of the gudgeon, so as to facilitate the formation of film. The hole *a* is for feeding the lubricant, whereas the holes *b* at both ends of the grooves are for cleaning with hot water or steam.

The form of the grooving employed is indicated in *Fig.* 571 ; some designers prefer triangular grooves, whereas others only apply a straight chamfered groove as drawn by chain-dotted lines. The transverse groove runs to about 1¼ in. from the bearing ends. The holes *a* and *b* are equally marked as in *Fig.* 570.

Automatic or Mechanical Lubrication has the advantage that all lubricating points are centralized in one or more main lubricators, which are operated by rotating or reciprocally moving engine parts; and the lubrication action therefore ceases, when the engine to which they are attached stops. Errors on the part of the operating attendant are thus reduced to a minimum, but if the containers of these mechanical lubricators are not filled at regular intervals, the wearing parts of the engine may run dry.

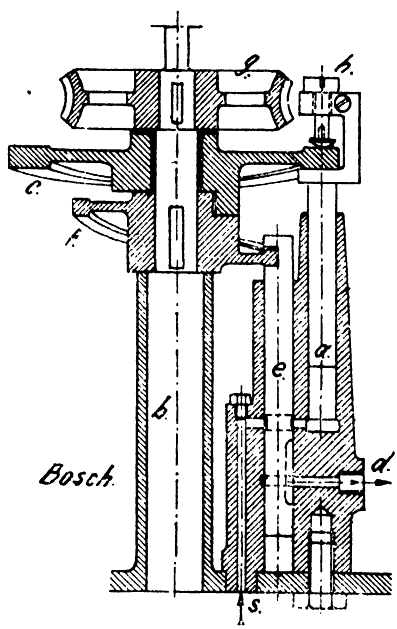


Fig. 572.—Mechanical Lubricator.

The construction of such a *Mechanical Lubricator* is shown in *Fig.* 572. It comprises a number of oil pumps *a*, mounted around the centre shaft *b*; the plungers are operated by a helicoidal disc *c* and the individual pumps have no valves, but a distribution slide *e* of cylindrical form, which is also operated by a helicoidal disc *f*. The main shaft is driven by a worm drive *g* and the whole set is mounted in a receptacle for oil (not shown); the suction of each pump is marked *s*, and the discharge shown at *d*, each of the latter connected by a steel or copper pipe, about

$\frac{1}{8}$ in. inside diameter, to the lubricating points of the engine. Ten pumps can be built as one unit and the oil pressure available is anything up to about 300 lbs./sq. in. As the whole pump mechanism is immersed in oil, there is little or practically no wear of the pump parts. The maximum volume per pump stroke is 0·2 c.c. (about 0·17 grm. or 0·0125 cub. in.). The screw *h* is for adjustment of the pump stroke for the individual requirement of each lubricating point. The maximum number of revolutions of the pump shaft is 10 per minute.

Grease Pumps for mill lubrication are also on the market, each unit having six pump cylinders for the six main bearings of a cane mill. The pistons are cam-operated and press the grease by spring pressure into the grease pipe to the bearing. For a mill running from $2\frac{1}{2}$ to 3 r.p.m., about 3 oz. grease per 24 hours is required for each bearing (about 85 grms.) and the pump volume per stroke is $\frac{1}{10}$ oz., which shows that the grease pump runs at a very slow speed.

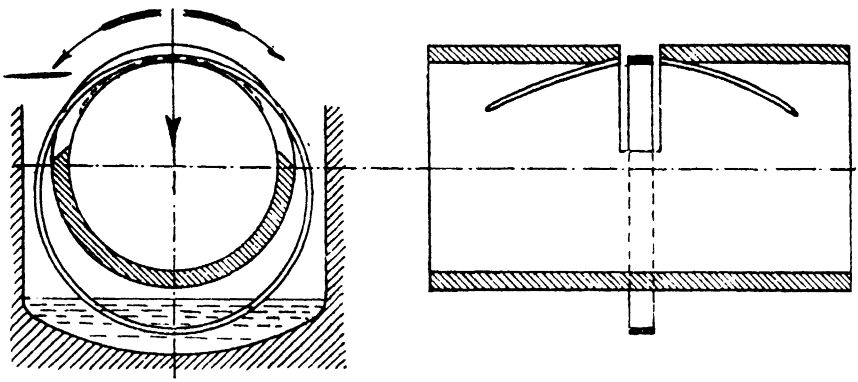
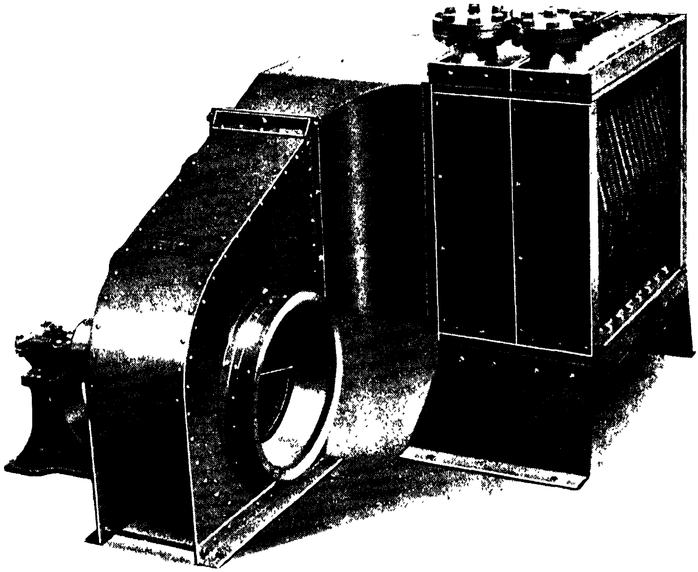


Fig. 573.—Ring Lubrication.

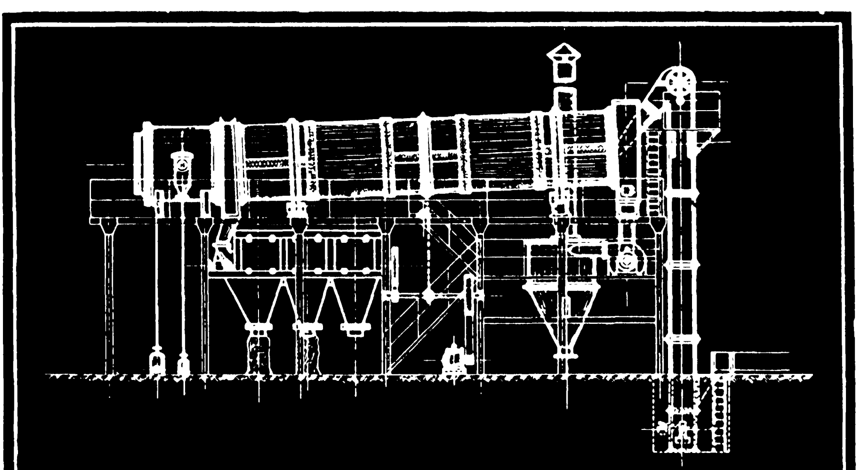
Forced lubrication is used in many high speed engines with enclosed engine frames (see *Fig.* 256) as well as in turbines and internal combustion engines. The oil is forced by a geared pump in excessive quantities to the bearings, which are literally flooded, and then returns to the crank case or bearing receptacles, from which it is pumped anew, after having been filtered, to repeat the cycle. Forced lubrication gives the lowest consumption of lubricant and generally pays for its installation in a short time. Cooling of the oil through a tubular oil cooler is also frequently provided; this cools the bearing surfaces efficiently, and thus a good deal of heat is removed. But it should be remembered that periodical tests of the body of the lubricant are necessary, as the oil finally loses its lubricating property and then the crank case needs refilling, after the stale oil has been drained off. A very good medium lubricating oil should be used in this case.

To this same principle belongs *Ring Lubrication,* shown in *Fig.* 573, where, when the load is vertical, the oil grooves have to be made on the top side as shown in the figure. The grooves are made for rotation in both directions, but if one direction rules and the bearing cannot be reversed, one set of grooves is sufficient, e.g., for left hand rotation, of course to the left.

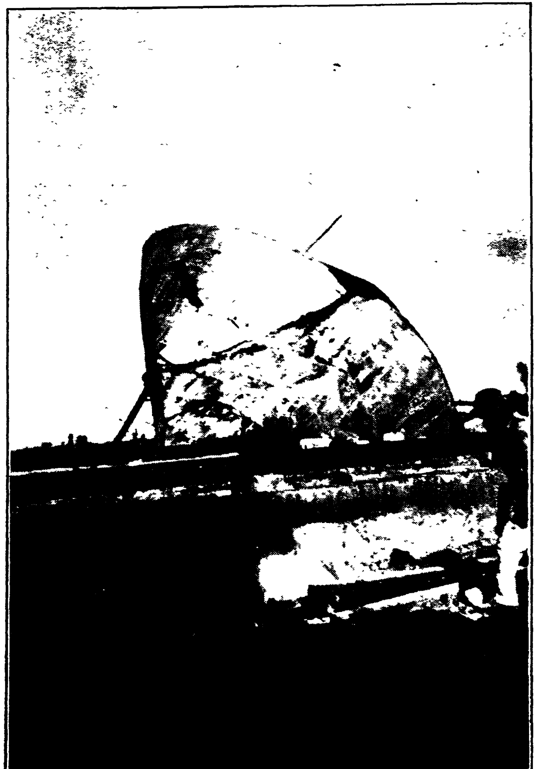
Steam cylinder lubrication is done efficiently by means of an *Atomizing Lubricator,* as shown in *Fig.* 574. The upper pipe connection *a* is attached at from 6 to 8 ft. above the steam cylinder to the live steam line *b* going to the



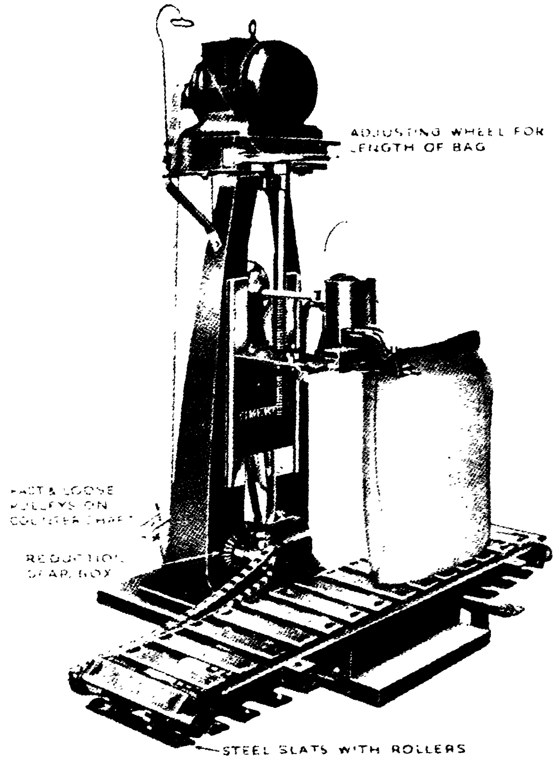
" SIROCCOFIN " HEATER, CONNECTED TO DISCHARGE OF " SIROCCO " CASED FAN.
(*Davidson & Co., Ltd.*)



SUGAR DRYING AND SCREENING PLANT.
(*U.C.M.A.S.*)



Empty 16-ton Molasses Tank swept from its Foundation over 100 ft. away.



Universal Bag Sewing Machine.

Sack-Filling & Sewing Machine Syndicate Limited.

engine; the steam in *a* condenses and enters the container *c*, which is filled with good cylinder oil, through the valve *d*. As soon as the condensate accumulates at the bottom the oil is discharged by an overflow pipe inside the container by the valve *e*, thence through the water-filled gauge glass *f* towards the drip tube *g* inside the steam line. The oil is atomized by the flow of steam and reaches the inside walls of the valve chamber and the cylinder piston course. The distance of the drip pipe *g* inside the steam pipe should be about one foot above the cylinder, as otherwise the oil might adhere to the steam pipe walls and not reach in atomized form the parts to be lubricated. Wet steam is not beneficial to this kind of lubrication, as in general a considerable part of the oil is drained with the engine condensate.

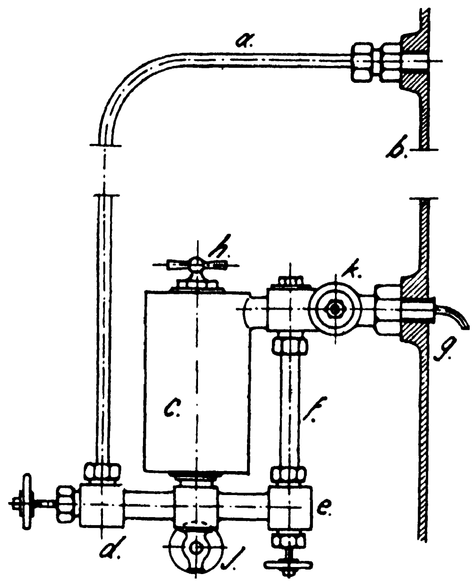


Fig. 574.—Atomizing Lubricator.

When the container *c* has to be re-filled, the valves *d* and *k* are closed and the plug *h* removed. The condensate is drained by the cock *j* before the oil is poured in. The needle valve *e* is for adjusting the flow of oil and should not be touched afterwards.

Visible drip oilers and grease cups are well known, as well as the customary oil cans and containers, so they do not need to be discussed here. They serve their purpose in those places where intermittent lubrication is required.

CHAPTER XXXIII.

MECHANICAL FACTORY CONTROL.

ESSENTIALS OF CONTROL — CONTROLLING APPARATUS — FLOW SHEETS — HEAT BALANCES.

In cane sugar factories chemical or manufacturing control has long been considered a necessity, but mechanical control has not always been viewed with favour ; and in many factories the mechanical engineer has had to satisfy himself with data from the laboratory manufacturing sheets, which contain—it should be fully recognized—many for his special purpose, but not all. In most cases mechanical control also should be embodied in the general factory control, as it may lead to savings in the operating costs of the factory and stop unnecessary losses.

There is no sharp limit to be drawn between the manufacturing control exercised by the laboratory staff and that which belongs to the mechanical engineer proper, and the author has always obtained the best results when both parties are working on a common co-operative basis. The mechanical engineer is sometimes excluded from the factory control, but this is a mistake in organization, as the functioning of many of the key apparatus is charged to this same engineer.

1.—Essentials of Control.

Not only commercially, but technically as well, great care should be taken by all concerned with the operation of a cane sugar factory, to ensure the proper control of all evolutions ; and the author has tabulated a list below of different controlling operations that do not concern the laboratory staff alone and to which the mechanical engineer should devote his full energies.

1 Weighing the cane.
2 ,, the bagasse.
3 ,, the imbibition water.
4 ,, the diluted juice.
5 ,, the ash from the boilers.
6 ,, filter-press mud.
7 ,, molasses.
8 ,, the sugars in bulk or in the bags.
9 Measuring the boiler feed-water.
10 ,, fuel oil.
11 Weighing additional fuel like wood, coke, coal.
12 ,, lime rock.
13 Measuring lime-milk.
14 Brix measurement of consecutive mill juices.
15 Brix of the diluted juice.
16 Temperature of juice entering the heater.
17 Temperature of the heated juice.
18 Brix of the clarified juice.
19 Temperature of the clarified juice entering the evaporator.
20 Brix of the syrup.

21 Temperature of the syrup, when charged to the vacuum pans.
22 Brix of the massecuite.
23 Temperature of the massecuite entering the crystallizers.
24 ,, ,, ,, leaving ,, ,,
25 Brix ,, ,, ,, ,, ,,
26 Temperature of sugar entering the bag.
27 Temperature of cooling water from bearings, jackets, etc.
28 Pressure of the live steam at the boilers and engines.
29 Pressure of the exhaust steam.
30 Pressure of the vapours from evaporators.
31 Vacuum of the evaporator condenser.
32 ,, ,, pan condenser or condensers.
33 Temperature of juice in evaporators.
34 ,, of massecuite in vacuum pans.
35 ,, of the injection water going to the condensers.
36 ,, ,, waste water from the condensers.
37 Measuring fuel for locomotives.
38 ,, lubricating oil for different departments.
39 Volumetric analysis of the combustion gases of boilers and lime kilns or sulphur furnaces.
40 Draught for individual boilers and the chimney.
41 Sugar or acidity of the injection (waste) water.
42 Alkalinity of the boiler feed-water.
43 Sugar and acidity in condensate from coils and heating elements.
44 Volumetric measurement of steam, going to the different departments.
45 Possible measurement of the condensate of each steam consumer.
46 Temperature of atmospheric air.
47 Moisture content of air.
48 Temperature of heated air, entering sugar dryer.
49 ,, ,, air released from dryer and cooler.
50 Moisture content of air from dryer and cooler.

There are several more measurements to be made in special cases but it is not essential to have continuous recording for all these operations, as many can be measured intermittently. But it is of importance for the factory engineer to have, e.g., a few recording thermometers and pressure and vacuum gauges at hand, which can be placed at any desired spot, so as to make a proper investigation in the department under consideration. These will yield a good insight to the prevailing fluctuations. Indicating steam engines, vacuum pumps, etc., will also be of valuable assistance to proper factory control.

2.—Controlling Apparatus.

Most of the controlling apparatus may be deemed to be known, such as recording or non-recording pressure and vacuum gauges, thermometers of the stem or dial type, as well as recording ones, Orsat apparatus for combustion gas analysis, dry and wet bulb thermometers of the sling type (psychrometers), hydrostatic meters for measuring the draught and the liquid level height in tanks, weighing scales for cane, juice, water and sugar, and indicators, etc. It would be impossible to treat all these apparatus within the scope of this book and only a few are dealt with in detail.

Water for imbibition should be weighed, but for boiler feed-water V-notch meters (see *Fig.* 442) are used to advantage. Molasses are generally measured by the tank volume they occupy, but weighing will give more exact figures.

Fuel oil is also measured by the tank volume, but for more precise results, ***Fuel Oil Meters*** are used, which differ from the common water meters on account of the viscosity of the material. In *Fig.* 575 such a meter is shown ; it is composed of two pistons or trunks, which by means of the oil flow put a small crankshaft in rotary motion and the number of revolutions of this crankshaft is recorded by a counter. As the friction of such a meter is very small, there is no detectable drop in pressure before or behind the meter. It will be seen from the figure that there are no valves in this hydraulic motor and both trunks carry interior ducts for proper oil distribution from the inlet to the discharge of the meter.

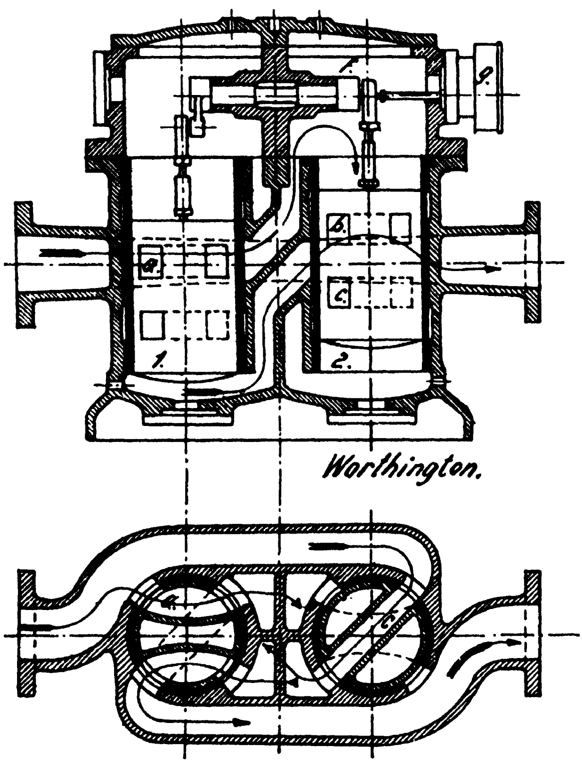


Fig. 575.—Fuel Oil Meter.

In the position of the trunks as drawn, No. 2 is pushed downwards by the oil through the upper port holes *a* of trunk 1 as indicated by the arrow. At the same moment the oil below trunk 1 is pressed via the centre conduit through the upper ports *b* of trunk 2 towards the discharge connexion of the meter. As soon as the dead centre of trunk 1 has been reached, the conduit *b* is closed and *c* opens, so as to give communication with the suction end of the meter. For the opposite ends of the trunk, the working is *vice versa*.

It will be obvious that by counting the number of revolutions of the crankshaft *f*, a measurement of the pump displacement has been obtained and the dial of the counter *g* is divided into the corresponding number of gallons.

These oil meters are made for capacities from 150 to 400 U.S. gals. maximum capacity per minute and can be used for hot, as well as for cold oil. Gas pockets under the top cover are released by pet cocks in these covers. The meters are designed for pressures up to 300 lbs./sq.in.

For measuring the flow of liquids, gases or steam, the flow meter has found a wide application in power plants and is also used in cane sugar factories for measuring the quantities of steam required at the different stations. The working of these meters is based upon the Venturi principle and an ingenious ***Electric Flow Meter*** is shown in *Fig.* 576.

In the pipeline which carries the liquid or steam to be measured a blind monel plate *a* with a small orifice is built in between two adjacent flanges.

The orifice is of such a size that sufficient passage exists with a drop in pressure from 1 to 2 lbs. per sq. in.

At *b* the full hydrostatic pressure is maintained, whereas at *c* there is less hydrostatic pressure, caused by the suction effect of the nozzle of the steam or liquid jet. The difference between these pressures depends on the rate of flow and as such differences are small, the sensitiveness of the meter is increased by the electric action of cutting in or out more resistances in the circuit of the measuring current. In a closed container *f*, filled with mercury till the seal pipe lower end is covered, the pressure from *b* is admitted. Within the seal pipe, the pressure *c* and the mercury will thus rise the more, the greater the difference in pressure between *b* and *c*. The higher the mercury rises, the more contact bars *g* are immersed and the more resistance *h* is cut out, as the mercury partly short-circuits the current from *x*-*y*. The intensity now of the current serves as an indication of the rate of flow and is measured. *i* is the indicating dial, *t* the totalizer and *r* the recorder.

For investigating the steam consumption of the different departments, these flow meters render valuable service ; the steam pressure should be recorded simultaneously, as the volume of the steam varies with the pressure. Accuracy within a few per cent. of the measured volume is generally obtained.

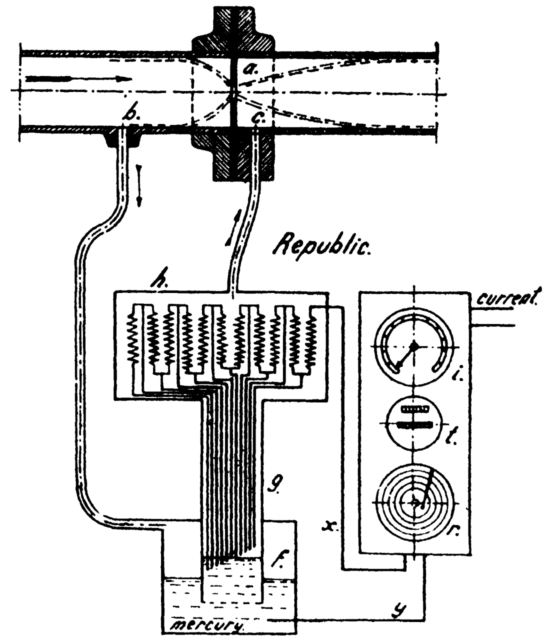


Fig. 576.—Electric Flow Meter.

Proper recording of the control data is always to be preferred to intermittent readings from an indicator, as in the latter case fluctuations may escape notice This also applies to the analysis of the combustion gases, especially that for the CO_2 content. There are now different CO_2 recorders on the market and they are generally based upon the absorption of CO_2 by chemical reagents. This absorption causes a difference in the pressures ruling before and after passing through the absorbing cartridges, and this difference is the measurable indication of the CO_2 content of the combustion gases. Instruments based upon the electrical conductivity of combustion gases are also frequently used.

A different principle is applied in the *Mechanical CO_2 Indicator* and recorder, as shown graphically in *Fig.* 577. The CO_2 gas is heavier than atmospheric air at the same pressure and temperature, and combustion gases with a high CO_2 content will also be heavier than those with a low one. A mechanical recorder, therefore, is based upon the specific gravity of the gas.

A small electric motor *a* in *Fig.* 577 drives by compound belting the two fan-shaped rotors *b* and *c*, aspirating respectively combustion gases and atmospheric air. The gases are thrown against impellers *d* and *e*, and as the rotors are driven in opposed directions, the impellers will also be driven oppositely, and the greater the gravity difference, the higher will be the torque difference of the impellers. Both impeller shafts are interconnected by levers *f-g* and the torque difference will move the arm *h* on the dial. At *j* there are humidifiers for both air and combustion gases, and the latter are aspirated through a porous filter *k* from the last boiler passage, and are sucked through the steel wool filter *l* and also a fine filter, before reaching the instrument. Below the filter a connexion for compressed air for cleaning it is provided and a water seal closes the bottom end of the gas line, thus protecting the instrument from excessive pressure or vacuum, which may cause damage.

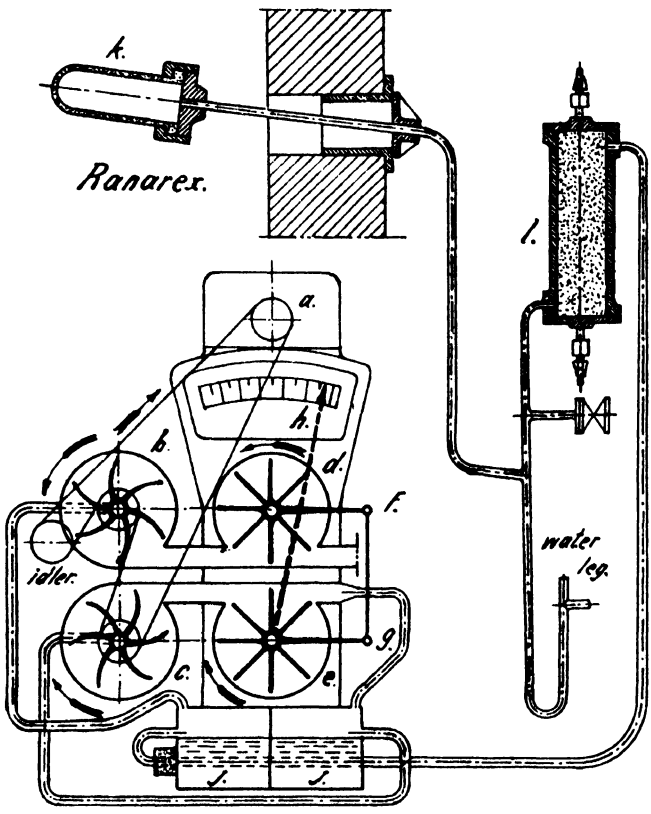


Fig. 577.—Mechanical CO_2 Indicator.

For heating bodies there are thermic control instruments, which will open or close the valve of the heating steam, in case the temperature of the heated medium drops or rises above the pre-determined limit.

Vacuum pan operation has received marked attention lately and the tendency is to put the boiling performance under the full control of the superintending staff, as empirical routine work at this station is not in accord with the requirements of control in the other stations of the factory. The author is of opinion that the operating as well as the designing engineer of a cane sugar factory should be as fully concerned with this station as are the chemical or manufacturing staff.

Several types of control apparatus have already been applied and have deepened the insight into the crystallization performance within the vacuum pan. These can be sub-divided into three groups :—

(a) Optical apparatus (brasmoscopes).

(b) Apparatus for determining the rise in boiling point of the mother-liquor (refractometers).

(c) Apparatus based upon the electric conductivity of the mother-liquor.

With all of these instruments a continuous record is generally not achieved, but only intermittent indications, which may result in unobserved fluctuations. Recording, therefore, has its definite merits. Moreover, these indicators are subject to the fluctuations in the prevailing vacuum;[1] and also the fact should not be overlooked that the field of observation or control is close to the pan wall and it is doubtful if the controlled sample represents the average condition of the pan contents, as the maximum circulation is to be expected in the pan downtake, where the interior parts of the instruments are not located as a rule.

The instruments mentioned give indication of the state of concentration of the mother-liquor in the pan ; the optical observation changes, the boiling point rises and the electro-conductivity decreases with increased concentration. For the latter, other components in the mother-liquor, especially ash, will influence the conductivity contrarily at increased concentration. The results, therefore, may prove less exact towards the end of the strike.

Moreover, the fluidity of the massecuite (i.e., the property which causes good circulation and therefore a constant growth of crystals in the mother-liquor) is not indicated by these instruments, and any inefficient design of the vacuum pan is difficult to trace from reading them.

A new instrument, recently invented and patented, which the author considers to have merits, is based upon the principle of heat transmission, and is influenced by the fluidity of the massecuite, while it is devised to compensate fluctuations in the prevailing vacuum.

This is the *Transmission Boiling Recorder* which is shown in *Fig.* 578. At the point of maximum circulation, i.e., the downtake and well below the graining massecuite level, there is located a small copper vessel *e*, connected by copper tubes *d* to the pan wall and bottom.

Reduced steam of about 60 lbs./sq. in. gauge pressure is admitted by the stop valve *a*, passes through a strainer *b* to a precision or needle valve *c*, the latter being set in a throttling position. It will now be apparent that the heaviest condensation in the vessel *e* will occur at lowest concentration and optimum circulation, as the heat absorption of the massecuite then will be highest. But the steam supply through the needle valve *c* is not sufficient to keep pace with the condensation, and the steam pressure will drop and thus the corresponding temperature of the condensate, which is measured by a mercury bulb at *f*, connected by the thermo-tubing *j* to the drum-type recorder *k*, which produces diagrams of about 4 in. × 19 in. for 8, 12 or 24 hours run. The recorder has also a vacuum coil, connected to the top of the pan by copper tubing which is provided with a cock *l*. The recording pen of the instrument is thus combinedly guided by the temperature of the condensate from the condensing vessel inside the pan and the prevailing vacuum.

[1] See the article of ALFRED L. WEBRE : "Design and Use of Pan Control Instruments," in *Int. Sugar Jl.*, 1936, p. 21.

The condensate from the vessel *e* is continuously removed by a trap *h*, having automatic de-aeration, after having passed a strainer *g*, which will keep out impurities that may deposit on the valve seat of the trap.

With higher concentration and less fluidity of the massecuite, the curve on the diagram will rise, and a fall is due to intake of syrup and increased circulation or fluidity. An interesting record of all the pan strikes is thus obtained and boilings can be efficiently finished, by merely observing the tracing of the graph, without the use of a proof-stick or taking observation at the sight glasses. The inventor has rebuilt inefficient vacuum pans, according to the analysed records of his instrument.

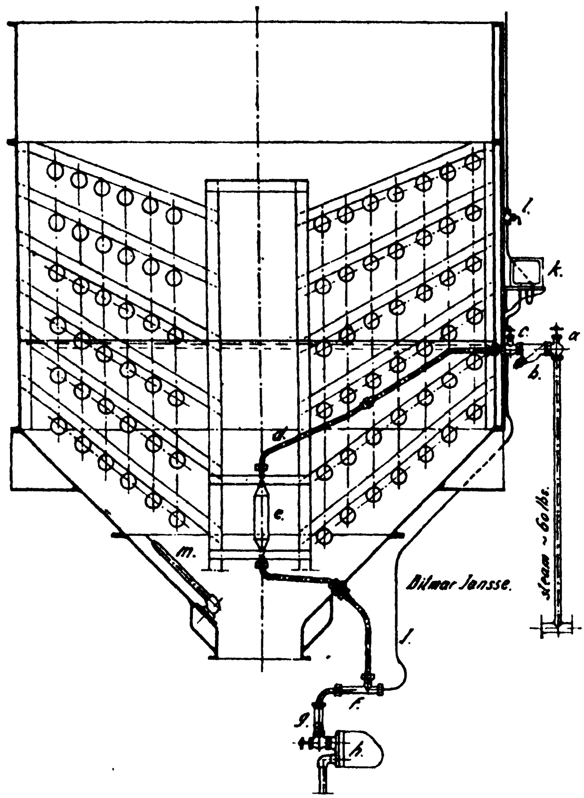


Fig. 578.—Transmission Boiling Recorder.

The valve *c*, once set, need not be touched again and for different boilings, different records are drawn and from the curves it can be ascertained whether high or low grades are being boiled. The steam consumption is very low, as the throttling valve only admits a very small quantity, while incrustations inside the condensing vessel need not be feared, as live steam which is available at any pan station, is used.

It is of importance for this kind of pan control, that the pan feed be not close to the condensing vessel, but below the heating elements, as shown by the distributing piping *m* in the figure.

In Java several of these instruments are already in successful operation.

3.—Flow Sheets.

The flow sheet is an efficient aid for the easy supervision of the factory procedure and all that belongs to it. It will enable the manager and the superintending staff to observe at a glance the *modus operandi* of the factory or the special department concerned. Some engineers prefer a large scale flow sheet, but for filing, a size of 11 in. × 24 in. should not be exceeded. On folding threefold it will just equal the dimensions of commercial paper.

Several flow sheets should be made and this work should be entrusted to the chief engineer and his staff, who possess the technical knowledge required for drawing them. The author has made very many of these flow sheets and a few particulars are tabulated below :—

(*a*) Live steam producers and consumers with inter-connecting pipe lines.
(*b*) Exhaust steam producers and consumers with ditto.
(*c*) The juice flow from the mills to the centrifugals.
(*d*) The condensate producers and storage.
(*e*) The hot and cold water systems.
(*f*) The condenser system with vapour, air and water pipe lines.

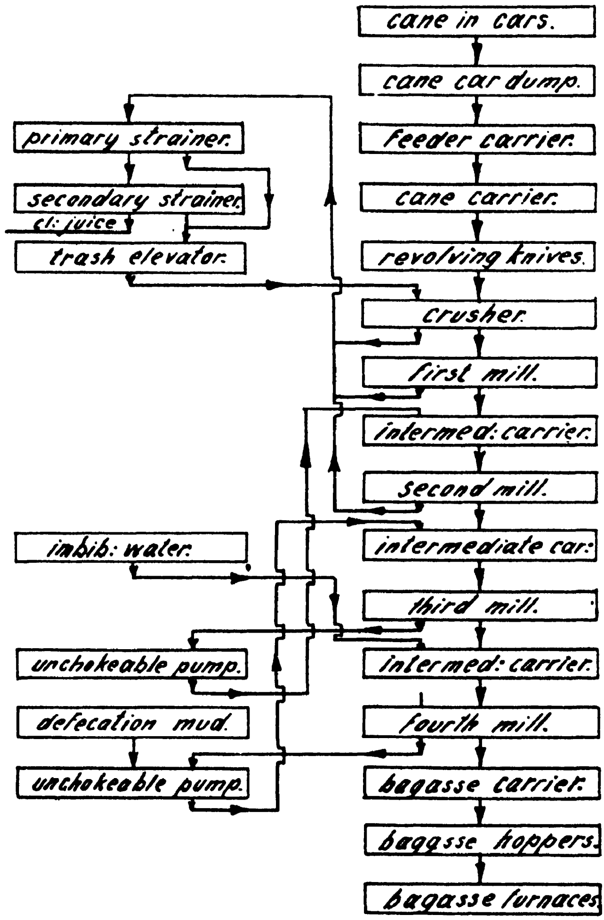


Fig. 579.—Flow Sheet of Milling Station.

The principal dimensions of the apparatus, etc., should be entered whenever possible. In *Fig.* 579 a flow sheet of a milling station is drawn and the maceration performance can be learnt from it. A flow sheet for a carbonatation factory is given in *Fig.* 580, indicating the lime and sulphur stations.

Alterations in the existing equipment should be clearly noted on the flow sheets, and the chief engineer should supply a copy or blue print, not only to the manager, but also to the superintending staff.

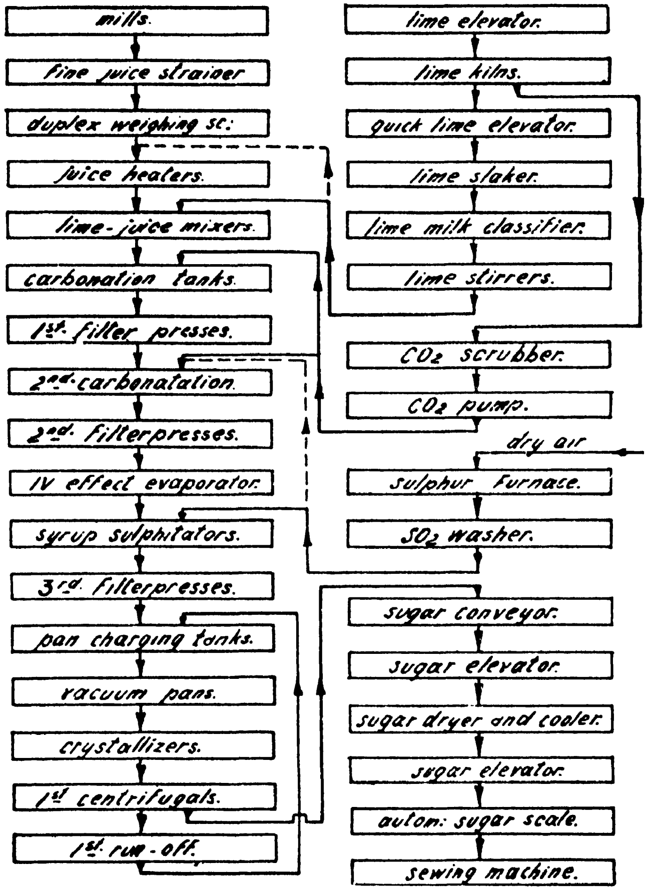


Fig. 580.—Flow Sheet of a Carbonatation Factory.

4.—Heat Balances.

Equally important for mechanical factory control are the heat balances, but generally the bagasse or fuel condition is so favourable, that they are often completely neglected. Moreover, operating mechanical engineers sometimes have such a dislike for compiling data and extensive figuring, that the author has seen very little done in practice in this matter, if not done by himself. All the same, it is logical to assume that while in business periodical balances of income and expenditure, which are universally considered a necessity, are compiled, this same necessity will arise in thermic accountancy, when keen competition leads to intensified sugar manufacture. Operating engineers should pay attention to this very interesting detail of heat balance, as it may lead to a saving in fuel and the day may not be far distant when bagasse will possess another value than just that for fuel, or other possibilities as to power supply may develop.

Heat balances are not difficult to compile and the average student should start at the bottom, by making *partial heat balances* of the different factory departments and later on compile those giving a general heat balance. It may be superfluous to mention that the incoming heat at any station is equal to the outgoing heat, which is the fundamental rule for the heat balance.

Proper analysis of temperatures, densities, gas analyses, etc., will influence the exactness of the heat balance, but, to start with, even estimated values may result in our finding some existing deficiencies.

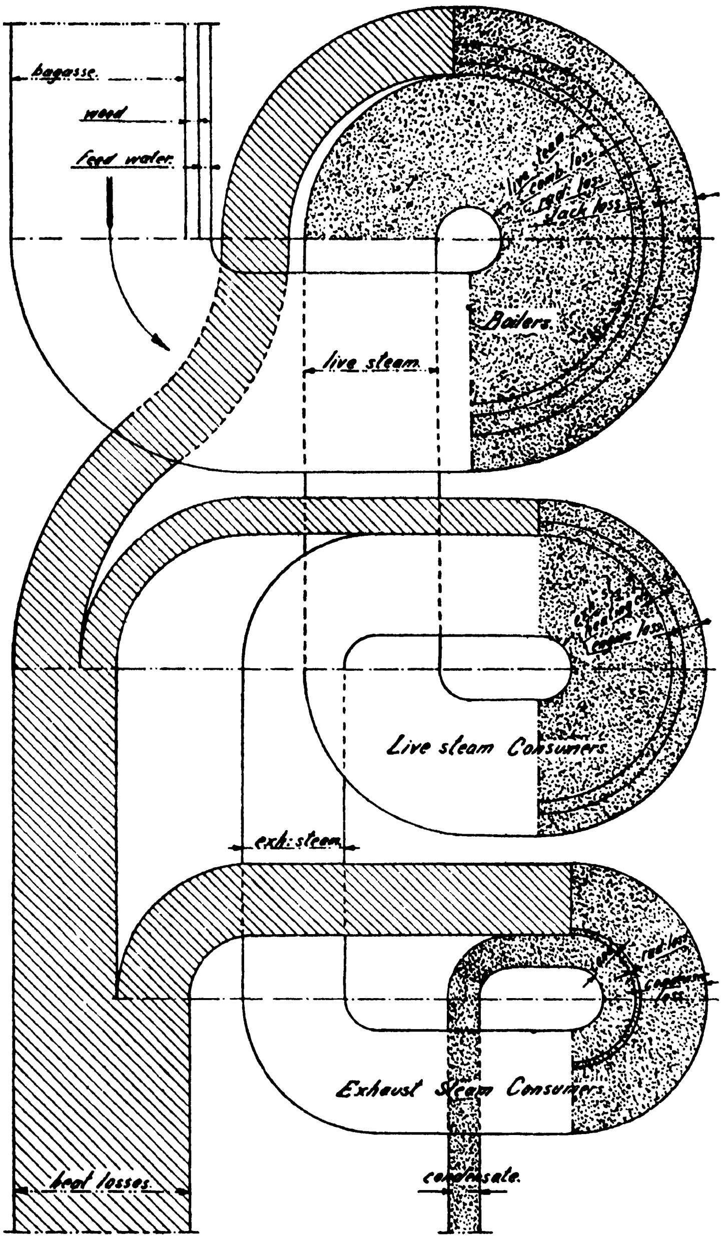


Fig. 581.—Steam Heat Balance.

In *Fig.* 581 a partial heat balance is shown, i.e. a *Steam Heat Balance* which is easily drawn, as each previous station supplies the required heat for the next one. Three definite stages exist :—

(a) The boiler heat balance.
(b) The heat balance of live steam consumers.
(c) The heat balance of exhaust steam consumers.[1]

Fig. 581 is also called a heat flow diagram ; it was made for a raw sugar factory, having 2250 tons grinding capacity per 24 hrs. The data were taken from the laboratory report and by intermittent or recording measurements. The heat balance has been made for the overall conditions prevailing.

In the previous Chapters it was shown how to calculate the quantities of heat, and the balances were given in calories and kgs. (1 kg. = 2·2 lbs. and 1 cal/kg. = 1·8 B.Th.U./lb. hourly quantities).

	Per cent.	Outgoing Cal.	Incoming Cal.
(a) *Heat Balance of boilers.*			
22,750 kg. bagasse at 1987 cal./kg. with complete combustion			45,200,000
1,200 kg. dry firewood at 3000 cal.			3,600,000
51,800 kg. boiler feed-water at 62°C.			3,212,000
51,800 kg. dry steam at 8 atm. abs. at 662 cal./kg.	66	34,320,000	
Combustion loss	6	3,120,000	
Radiation loss	10	5,201,000	
Stack loss	18	9,371,000	
Totals	100	52,012,000	52,012,000
(b) *Heat Balance of live steam consumers.*			
51,800 kg. live steam (see above)			34,320,000
Energy, automatic expansion engines, 960 i.h.p.	1·2	609,000	
Condensation losses ditto	1·7	890,000	
Energy, turbo-alternators, 1130 i.h.p.	1·4	718,000	
Condensation losses ditto	2·9	1,496,000	
Energy, turbine pumps, 212 i.h.p.	0·3	134,500	
Condensation losses ditto	0·7	365,000	
Energy, atm. discharge, engines, 190 i.h.p.	0·2	120,500	
Condensation losses ditto	0·7	327,000	
Defecators	1·5	761,400	
Condensate ditto 80°C., 1320 kg.	0·2	105,600	
Cachaceras,[2] 750 kg...	1·0	498,000	
Molasses blow-ups, 350 kg.	0·4	232,000	
Vacuum pans	3·3	1,715,000	
Condensate ditto, 80°C., 2890 kg.	0·5	231,000	
Exhaust steam, 46,490 kg.	50·0	26,117,000	
Totals .. 51,800 kg.	66·0	34,320,000	34,320,000
(c) *Heat Balance of exhaust steam consumers.*			
46,490 kg. exhaust steam with 15·2 per cent. moisture (see above)			26,117,000
First body of evaporators	34,500 kg.	19,389,000	
Vacuum pans	11,990 kg.	6,728,000	
Totals	46,490 kg.	26,117,000	26,117,000

[1] See also the author's paper read before the Association of Sugar Technologists of Cuba, Proceedings, 1934, page 220
[2] See page 372.

Similarly, the following partial heat balances have been made :—

(*d*) First juice heaters.
(*e*) Defecation tanks.
(*f*) Cachaceras.
(*g*) Secondary juice heaters.
(*h*) First body of evaporators.
(*j*) Second body of evaporators.
(*k*) Third body of evaporators.
(*l*) Fourth body of evaporators.
(*m*) Vacuum pans.

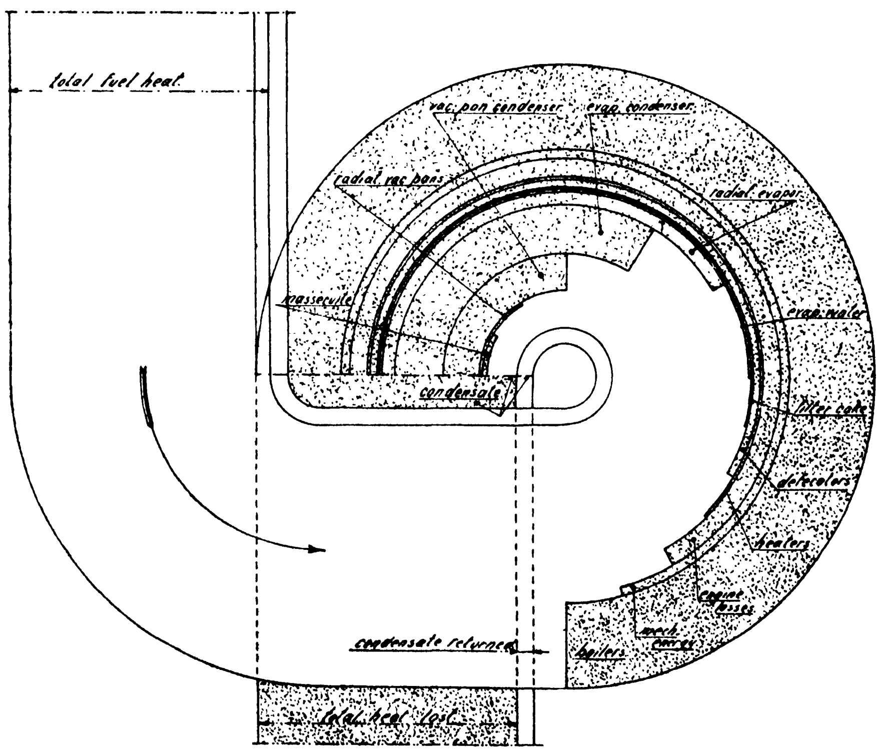


Fig. 582.—Total Heat Flow Diagram.

From these data, now, the general factory heat balance can be compiled, and the *Total Heat Flow Diagram* of it is shown in *Fig.* 582, which is drawn to scale, just as the previous one.

The general balance is now as follows :—

General Heat Balance of Factory.

	Outgoing.		Incoming.
Bagasse, total combustion heat		..	45,200,000
Firewood, additional fuel		..	3,600,000
Boiler feed water, re-entered condensate		..	3,212,000
Raw juice, initial heat		..	2,820,000
Dilution water in cachaceras and tank washing		..	284,000
Heat in returned molasses		..	81,000
Losses.			
Boiler losses	17,692,000	..	
Mechanical energy	1,582,000	..	
Engine and steam pipe losses	3,078,000	..	
Juice heater losses	577,000	..	
Defecation losses	1,724,000	..	
Loss in filter-press mud	96,000	..	
Losses through evaporation in defecators and cachaceras	256,000	..	
Losses in evaporators and connecting pipe lines	3,173,000	..	
Condenser losses for evaporators	9,450,000	..	
Condenser losses for vacuum pans	7,402,000	..	
Losses in vacuum pans and connecting pipe lines	621,000	..	
Heat losses in massecuites	598,000	..	
Heat in condensate, of which only a part is returned into process	8,948,000	..	
Totals	55,197,000	..	55,197,000

Summarizing, it will be seen that heavy losses occur at the boiler station and considerable heat destruction takes place in the condensers. The power requirements in heat value are not even 3 per cent. of the total heat employed, whereas radiation consumes also a considerable quantity of heat. A heat balance for every factory may mean an indication at which station the heat efficiency can be improved.

CHAPTER XXXIV.

POWER PLANTS.

GENERAL DATA — STEAM POWER — DIESEL POWER.

1.—General Data.

The centralization of power supply to the cane sugar factory is as important as in any other industry, but at one time, for practical reasons, it did not receive the attention it deserved; the different power requirements, using steam-driven units, made any centralization impracticable, as the machinery had to be split up among different departments of the factory and power transmission by shafts and belting is not well adapted for a cane sugar factory.

Formerly the power plant of a cane sugar factory only contained the prime mover, generally of small size, for driving a dynamo solely for lighting purposes. Even the largest factories did not require over 100-150 kw., which is equivalent to 200,000 to 300,000 candle-power in present-day incandescent bulbs.

With the introduction of electricity as the means of power transmission, the situation altered and large prime movers connected to alternators, or in some cases to generators, have been arranged in a separate building, the power plant or power house. The centralization doubtless gives a better arrangement; large units with higher thermo-dynamic efficiency, the use of steam of higher pressure and superheat, and the reduced outlay for pipelines in the factory are amongst its advantages. Electrical power transmission, of course, fulfils the requirement of high reliability under operating conditions.

It will moreover be obvious that machinery in general can be cared for more adequately in a clean and spacious power house, than if scattered over the factory, and subject to moisture, drippings or dust.

The location of the power house should be as centrally as possible within the building complex of the factory, and although this may not be possible in all instances, a few advantages for doing so are :—

(*a*) Proximity to the boiler house means short steam lines.
(*b*) Proximity to the boiling house means short exhaust lines.
(*c*) Proximity to the factory condensers means location of the vacuum pumps, etc., inside the power plant.

For the sake of efficient maintenance there should be located in the power house of a newly designed factory as much machinery as can be arranged, and neatness of arrangement should not be overlooked.

The machinery which can be so located is tabulated below :—

1. Prime movers for generating electrical current for factory operations and lighting (steam engines or turbines).
2. Ditto for the dead season (steam or Diesel engines).
3. Vacuum pumps for central condensers.
4. Injection and waste-water pumps for ditto.
5. The pumps for the hot and cold water service of the factory and what belongs to it.
6. The air compressors for shop and sulphitation.
7. The CO_2 pump for carbonatation service.
8. The switchboard for electrical power distribution and lighting.
9. Electrical sundries like transformers, converters and exciters.

The power house should be well separated by walls from the other factory departments, to keep out dust, smoke, sulphur and other gases. The building should be well ventilated and daylight ought to have free access. Artificial lighting during night time should also be ample.

A traversing crane, hand or electrically driven, should be arranged overhead with a lifting capacity well above the weight of the heaviest pieces of machinery inside the building.

The floor should be laid from 6 to 12 feet above the ground level ; this will keep out moisture from the soil and moreover cables and piping can be laid beneath the floor level. The remaining ground floor can be arranged as a store room for electrical supplies, as well as for the electrical repair shop. The floors are generally of concrete and in some instances tiles are used, which will increase the amenities of the place.

The power requirements should be studied in each instance, and a few data from Cuban cane sugar factories are given below :—

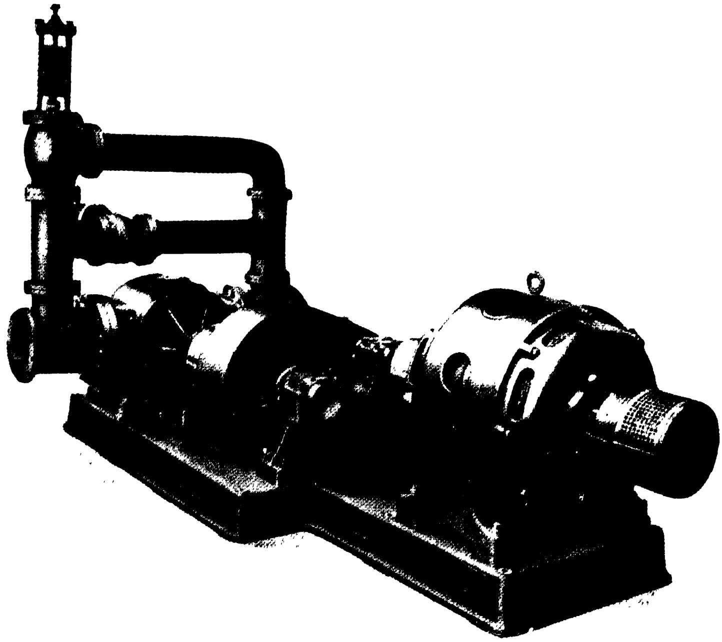
Capacity per 24 hours		
11,000 tons cane	12,100 kw.	for factory operation, totally electric.
6,000 ,, ,,	2,500 kw.	electrified except mills.
	180 kw.	for dead season.
3,000 ,, ,,	250 kw.	for lighting and water service only.
2,500 ,, ,,	75 kw.	for lighting only.
2,200 ,, ,,	90 kw.	,, ,, ,,
2,200 ,, ,,	3,000 kw.	totally electrified.
	50 kw.	for dead season.
1,950 ,, ,,	70 kw.	for lighting only.
1,800 ,, ,,	30 kw.	,, ,, ,,
1,650 ,, ,,	1,250 kw.	electrified except mills and vacuum pumps.
	95 kw.	dead season.
1,650 ,, ,,	60 kw.	for lighting only.
850 ,, ,,	600 kw.	electrified except mills.
	15 kw.	for dead season.

The dead season lighting equipment is generally for the dwellings and living houses of the factory staff, and when a factory is newly designed a liberal allowance for future extensions has to be made.

Two kinds of prime movers are to be found in a power plant of a cane sugar factory ; the steam drive through engines or turbines is of course the most common, but for dead season requirements, the Diesel engine has special claims for economy in first and operating costs.

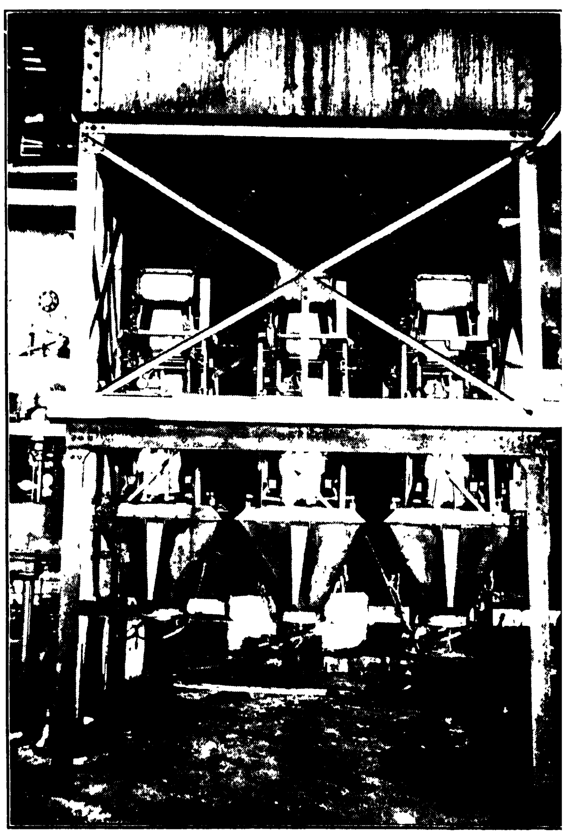
2.—Steam Power.

The prime movers of the cane sugar factory are generally of the single back-pressure type, where the expansion performance of the steam is achieved in one operation. Piston engines as well as steam turbines may be used, the latter for larger power outputs of above 500 h.p. approx., which produce an oil-free exhaust steam. The former generally have a higher thermo-dynamic efficiency for the lower steam pressure of 100 lbs. per sq. in. and a back pressure of 7 lbs. per sq. in., which pressures may be considered standard ones for the average cane sugar factory. The expansion performance within the steam cylinder or turbine casing will cause a heat drop, equivalent to the mechanical power delivered by this expansion performance, and for saturated steam of the above-mentioned pressures, this heat drop amounts to about 122 B.Th.U. per lb. of steam as measured from the entropy chart.



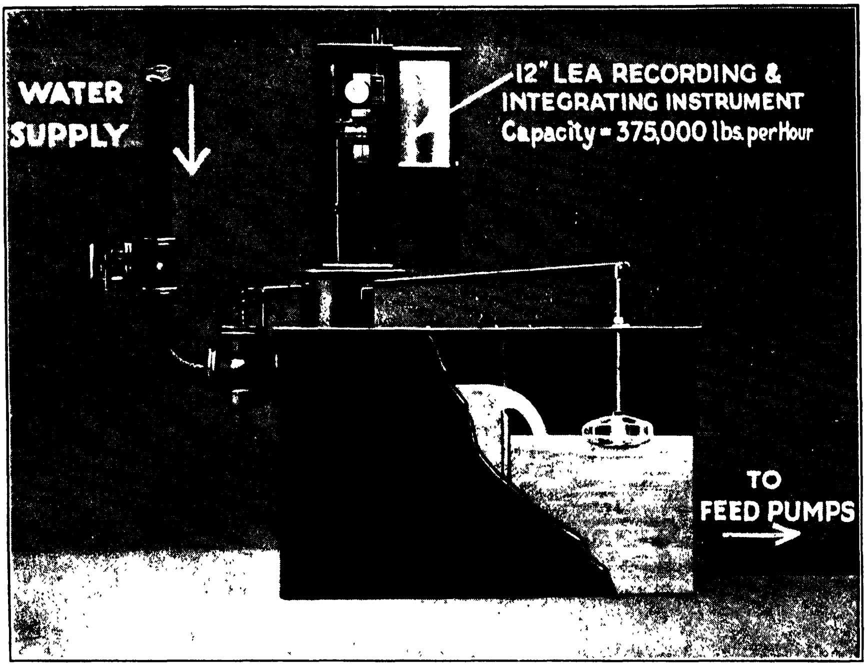
Above—

Electrically - driven Molasses Storage Pump, 250 tons/hr. with Bye-pass and Relief Valves.
(*Stothert & Pitt, Ltd.*)



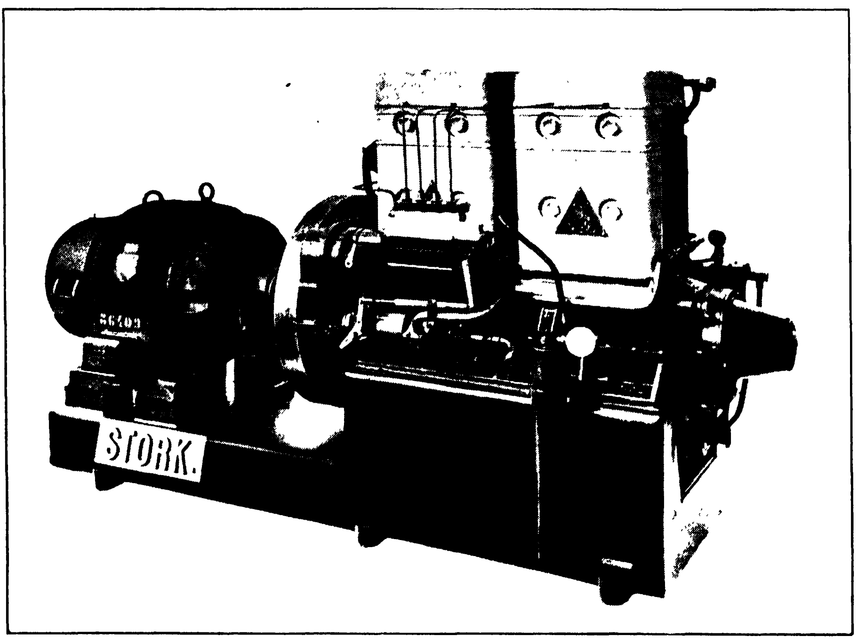
On Right—

Three Automatic Raw Sugar Bagging Scales in a large Cuban Central.
(*Richardson Scale Company*)



BOILER FEED MEASURING RECORDER.

(Lea Recorder Co., Ltd.)



50 KW. DEAD SEASON HIGH SPEED DIESEL GENERATOR SET.

(Gebr. Stork & Co.)

As exhaust steam is used for heating purposes in the processes of sugar making, the back pressure engine so far has been considered the most feasible power generator for sugar factories. The steam produced by the available bagasse has generally been in excess of the power demand, and no special attention has had to be paid to the steam economy of the prime movers.

Process work nowadays requires, all the same, more power than has been the case hitherto and outside power demands for pumping plants, for irrigation, for electric traction or shunting, as well as lighting and power supply to the vicinity, may cause a shortage of bagasse fuel, as the steam requirements become heavier. Steam economy of the prime movers, therefore, will become of paramount importance, all the more as the demand for exhaust steam in the boiling house has latterly decreased thanks to the use of pre-evaporators, which supply heating vapours to heaters and evaporators or pans; and thus a higher efficiency in this department has resulted, so far as the demand for steam is concerned.

For the reasons explained, the condensing prime mover (where the exhaust steam is led into a condenser under vacuum and thus condensed) will become of some interest for those factories with a heavier power demand than the available bagasse can now cope with. In the condenser, nevertheless, the latent heat of the steam is transferred to the cooling water at too low a temperature to be of any use for the heating purposes of the boiling house, and this will result in a direct heat loss for the process work.

With a live steam pressure of 100 lbs. as before-mentioned and a vacuum of 27 in. for 30 in. barometric pressure, which may be considered a fair obtainable average for tropical conditions, the expansion of the steam will cause a heat drop of approx. 281 B.Th.U. for saturated steam and isothermic expansion, instead of the 122 B.Th.U. when the back-pressure amounts to 7 lbs. per sq. in. instead of 27 in. vacuum. It will thus be seen that more than twice the amount of power can be obtained by equal thermo-dynamic efficiencies of the prime movers, using the same amount of steam, having the same initial pressure.

As not all the exhaust steam can be condensed in the engine condenser on account of the process work in the boiling house necessitating always the largest amount of the exhaust steam produced, engine types have been developed, where the larger part of the steam is extracted at the normal back pressure as required for the boiling house while the excess steam is led to a condenser with the corresponding additional power generation.

These extraction or bleeder engines are generally based on the compound working of the steam in the engine and this feature calls for our preliminary consideration.

The compound engine has been created to divide into two stages the expansion performance of the steam, each stage generally accomplished in a separate cylinder or turbine casing. The total heat drop of the steam through expansion is thus divided into two parts and radiation as well as cooling of the steam will be less, since maximum and minimum temperatures in each cylinder will show less difference, and thus heat losses will be lower, giving a higher thermo-dynamic efficiency and a lower steam consumption per h.p./hr.

But it should be borne in mind that there is no difference in the expansion performance, when it is divided over two stages, and no additional power output is to be expected, but only a more economic use of live steam.

The author has "indicated" a cross-compound Corliss engine in one of the older Cuban sugar factories; this had a H.P. cylinder 12 in. diameter and a L.P. cylinder 22 in., both cylinders having 36 in. stroke. The engine runs at 76 r.p.m. for driving 12 centrifugals of the belt-driven type with baskets 40 in. × 20 in. In *Fig.* 583 are shown the diagrams for H.P. and L.P. cylinders under friction load, i.e., with empty centrifugal baskets.

The total power output amounts to 61 i.h.p., the steam distribution of the low pressure cylinder being wrongly set and the high pressure cylinder developing about 25 per cent. more power than the L.P. cylinder, which might indicate that this engine was not bought for the steam conditions prevailing at this sugar factory. Moreover, the live steam pressure only amounted to 70 lbs., with a back pressure of about 2 lbs., thus with too low a heat drop to justify the installation of a compound engine.

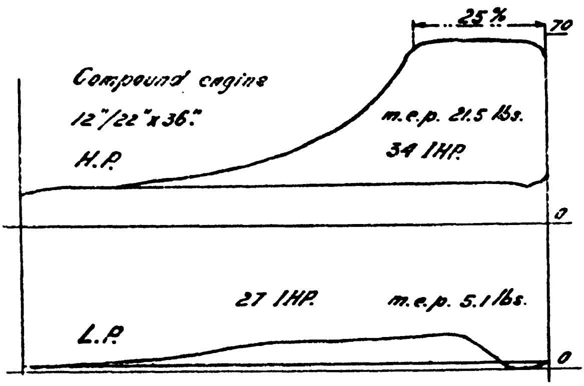


Fig. 583.—Indicator Cards of Compound Engine.

In *Fig.* 584 the diagrams for normal load are shown, the difference in power output of each cylinder being still heavier and 124 i.h.p. in all delivered. The steam admission of the H.P. cylinder covers 75 per cent of the stroke and the engine is really at the maximum of its capacity.

In *Fig.* 585 the author has drawn the theoretical diagram for this engine. As the volumes of H.P. and L.P. cylinders have a ratio of 0·3 : 1, a vertical line is drawn at 0·3 of the stroke, indicated by "H. Press," thus reducing the diagram of the H.P. area to the same scale as that of the L.P. diagram area. The admission for *Fig.* 585 will amount to 75 per cent. from 0·3 of the H.P. stroke, thus 0·75 × 30 = 22·5 per cent. of the full diagram stroke. The expansion line is drawn as an isotherm for saturated steam of 70 lbs. per sq. in. pressure, the end of the expansion reaching a pressure slightly above the prevailing back pressure. The compression curve is drawn for a 2·5 per cent. dead space, indicated by *D.S.* in the diagram, which can be obtained with good Corliss design.

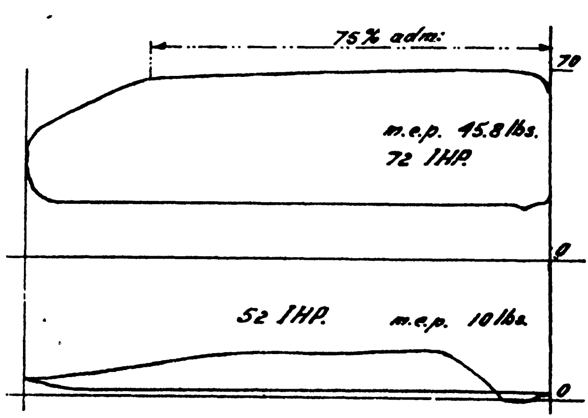


Fig. 584.—Normal Load Cards of Comp. Engine.

In this theoretical diagram—which is shown in full lines—the H.P. and L.P. diagrams from *Fig.* 584 are indicated by dotted lines, and the shaded part represents the loss in diagram area, which is not favourable in this case, indicating that the engine had been designed as a condensing engine. It is clearly seen that the H.P. cylinder covers the pressure drop from 70 down to 22 lbs., whereas the L.P. cylinder covers the range from 19 to 2 lbs. per sq. in.

The compound engine, therefore, should be used where superheated steam of high pressure is available and condensing is applied, thus where a large heat drop prevails. For cane sugar factories the compound engine will have only a rare application, but in the form of an extraction or bleeder engine, which is a compound engine of special design, it might be used to a larger extent in the future.

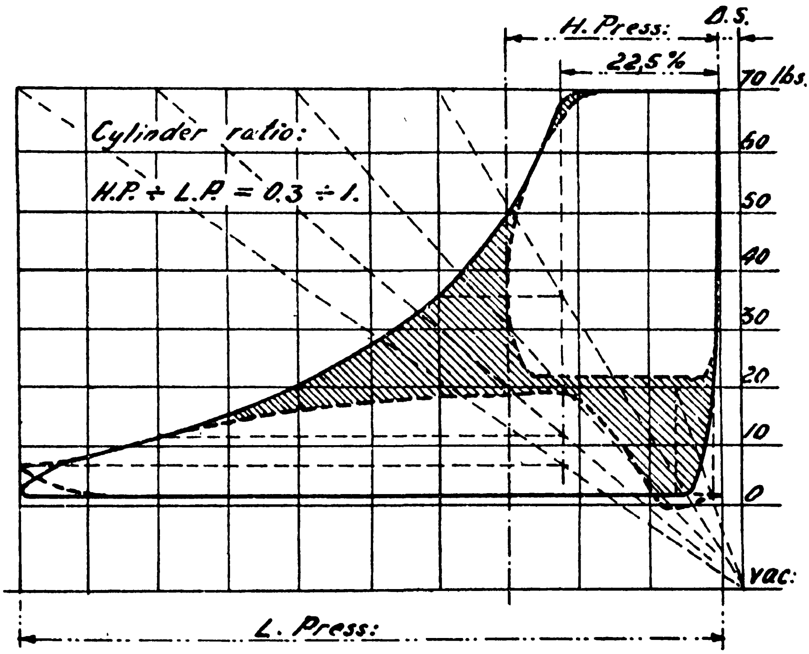


Fig. 585.—Combined Compound Engine Diagram.

Where a part of the exhaust steam is not required for heating purposes, this part may be condensed, thus yielding a larger power output. A very ingenious type of engine for this kind of work is the mixed pressure engine, which is a one-cylinder engine in its simplest form, in which the expanded steam on one side of the piston is discharged into the exhaust steam line with the prevailing back pressure and the steam on the other side is led into a condenser under vacuum.

In *Fig.* 586 is shown the theoretical diagram for this double performance. The assumed steam pressure is 150 lbs. per sq. in., whereas the back pressure is 7 lbs. per sq. in. Saturated steam is considered but it should be remembered, to start with, that superheated steam will further increase the advantages.

With an admission *b* of 20 per cent. of the stroke on the right hand side of the diagram, a mean effective pressure of 70 lbs. per sq. in. is achieved. On the left hand side, being the condensing side, a 10 per cent. admission *a* will produce a mean effective pressure of 67 lbs. per sq. in., thus nearly equal to the one on the right hand side and a heavy flywheel will not be required for such an engine. The vacuum is taken as 27 in., the working figure under tropical conditions.

Leaving out other factors for the moment, the right hand side of the diagram, assumed to represent the crank side of the engine cylinder, will have twice the admission and thus twice the steam consumption of the left hand or cover side of the cylinder. Of the total steam consumption, two-thirds will be available as exhaust steam for heating purposes, while one-third is led to the condenser and thus is lost for the boiling house of the sugar factory.

The steam going to the condenser will thus deliver about twice the amount of power per lb. supplied as the steam discharged or exhausted under the mentioned back pressure of 7 lbs. per sq. in. This shows clearly the high power-developing capacity of the condensing side of the engine.

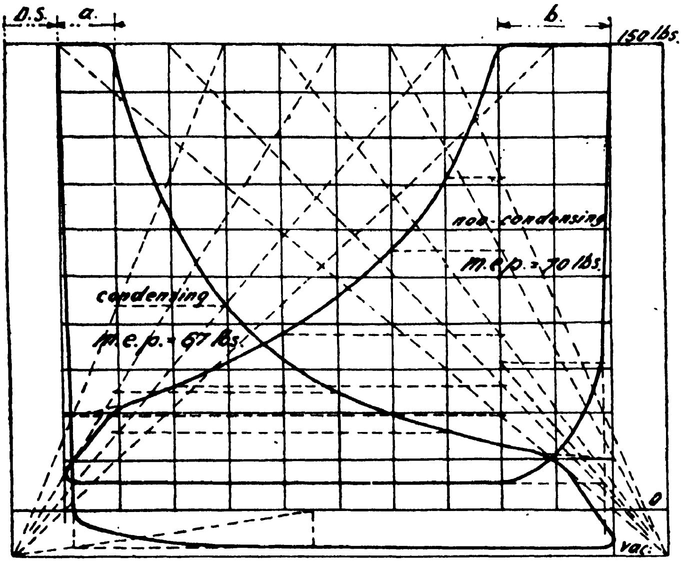


Fig. 586.—Diagram for Mixed Pressure Engine.

In practice these results cannot be obtained to the full extent, as the thermodynamic efficiency will have a higher value for the non-condensing side. For overall calculations this T.-D. efficiency will amount to about 0·7 for the non-condensing and about 0·55 for the condensing side with well designed and insulated cylinders.

The dead space, *D.S.*, for this engine has been taken as 10 per cent. of the net cylinder volume, but for small size engines may be more, reducing the thermal efficiency and thus increasing the steam consumption per h.p.

The design of the cylinder of such a mixed pressure engine, as used in other industries where the extraction or bleeder performance is required, is shown diagrammatically in *Fig.* 587. The engine has two distribution piston valves and three branches, one for live steam on the admission valve chamber, one for the bleeder or back-pressure connexion and one for the condenser. The two latter are both on the exhaust valve chamber.

The steam admission valve *a* charges on the inside edges of the steam ports, so the high pressure will not act on the stuffing boxes of the valve rods. This piston valve is operated by its own eccentric, regulated by the speed governor of the engine.

The discharge or exhaust valve *b*, also of the piston valve type having a division arranged in the centre, so as to separate the back pressure from the condenser chamber, is likewise operated by a separate but fixed eccentric. The valves at both ends discharge also on the inside edges, and the outside spaces between the valves and the valve chamber covers are generally connected to the back-pressure pipe line, so as to take up any leakage that might

The cover side of the piston valve *a* will have a larger steam lap than the crank side, as a smaller admission or cut-off is required. The discharge valve *b* has to give a larger compression on the condensing side, than on the non-condensing, and sometimes both have to be sacrificed a little to make this feasible.

As will be noted, this design is not complicated and for larger power outputs above 150 to 200 h.p., the two-cylinder construction can be used to advantage, giving a still greater regulation of the amounts of steam to be condensed.

Instead of piston valves, poppet or Corliss valves can be applied, the former being especially adapted for superheated steam, and great independence of steam distribution by the four valves is achieved.

The extraction or bleeder engine is a specially designed compound engine in which steam is extracted from the receiver or connexion between the H.P. and the L.P. parts of the engine, after having developed power in the H.P. section.

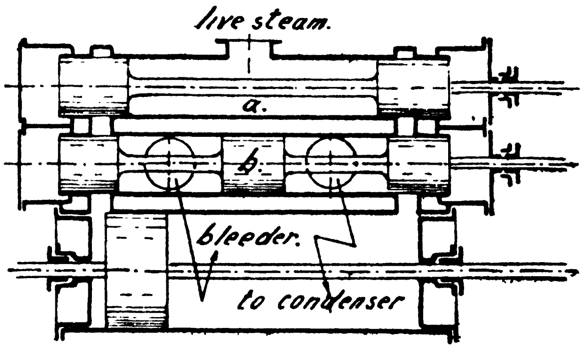


Fig. 587.—Mixed Pressure Cylinder Design.

It will be obvious that the cylinder ratio of an extraction engine has to be different from that of a straight or cross compound engine, as the volume of the steam going to the L.P. cylinder is reduced by the extracted quantity. Similarly, the L.P. section of a bleeder turbine has to be bladed according to this smaller amount of L.P. steam available, and for regulating purposes the H.P. part has to be separated by a division wall from L.P. part of the turbine.

The indicator diagram is only obtained from piston engines, but as it indicates so clearly the expansion and bleeder performance, it has been used throughout this chapter. The same *modus operandi*, nevertheless, may take place within a steam turbine and turbines should be preferred for the larger power outputs as already explained.

Two systems of steam extraction can be used in cane sugar factories: the high pressure type for back pressure exhaust, and the lower steam pressure in connexion with the condensing type. Both types will be discussed below.

In *Fig.* 588 is shown the indicator diagram for the high pressure extraction performance. The live steam pressure is assumed to be 300 lbs. per sq. in. and although superheat is not considered in these examples, it nevertheless can be used to full advantage. The bleeder steam is extracted at 120 lbs. per sq. in. and is to be used in the engines or turbines, designed for this lower pressure. The arrangement is thus suitable for all those installations where a pressure of 120 lbs.—or any pressure around this figure—is the normal boiler pressure, and obsolete boiler capacity has been replaced by new equipment designed for a higher steam-pressure, i.e., in this case 300 lbs./sq. in. As the generation of steam at 120 lbs. per sq. in. requires 1191 B.Th.U. per lb. produced, against 1203 B.Th.U. per lb. of steam having 300 lbs. per sq. in. pressure, this will indicate that the higher pressure will yield great advantages in respect to power output, as for its generation only a small additional amount of heat is required.

The H.P. prime movers for this layout, therefore, are called leader-engines or leader-turbines as they are arranged in advance of the existing prime movers, and a higher power output is obtained with the same amount of steam.

For the assumed back pressure of 7 lbs. per sq. in. a steam admission *a* of about 8 per cent. is required, so as to release the exhaust steam slightly above this mentioned back pressure. The H.P. cylinder ratio has been taken as 0·4 of the low pressure cylinder and the H.P. admission or cut-off will thus amount to 0·08 ÷ 0·4 = 0·20, or 20 per cent. By increasing the H.P. cut-off to 45 per cent. (the distance *b* in the diagram = 18 per cent. of the L.P. diagram stroke) about 50 per cent. of the expanded steam can be drawn off at 120 lbs. pressure, the other 50 per cent. being expanded in the L.P. part of the extraction engine. The extracted steam can be used in the existing low pressure engines of the factory. The shaded part in the diagram indicates the increase in diagram area, which is equivalent to an increase of power output.

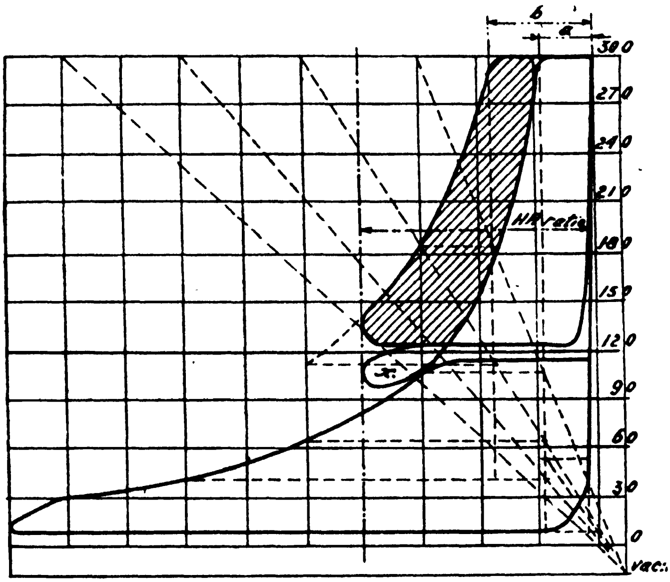


Fig. 588.—High Pressure Extraction Diagram.

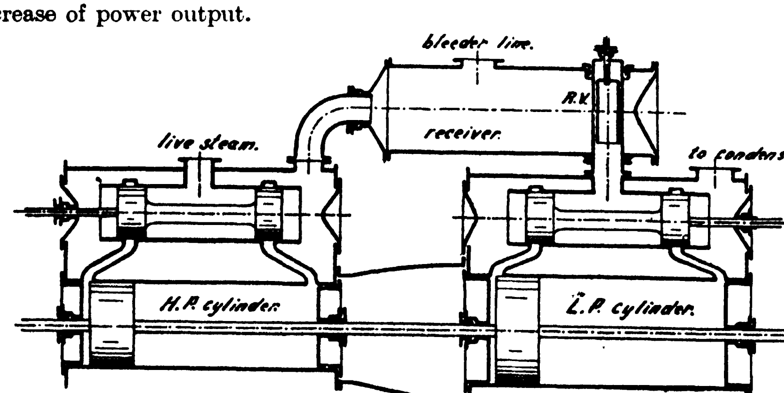


Fig. 589.—Two-Cylinder Extraction Steam Engine.

When there is no bleeder steam required and the cut-off is reduced by the governing mechanism to the dimension *a*, a loop *x* will be produced in the H.P. diagram and a small power loss will be the result, a matter of only secondary importance. But this loop can be avoided, as will be shown later in *Fig.* 591.

The essentials of the construction of a piston extraction engine, the H.P. and L.P. cylinders arranged in tandem, are shown in *Fig.* 589, this having piston valves for the steam distribution. The live steam enters at the H.P. valve chest and the cut-off is under control of the speed governor of the engine, regulating the eccentricity of the H.P. piston valve.

The exhaust of the H.P. cylinder is discharged into a receiver, to which the bleeder line is connected. The engine, of course, may be designed for any bleeder pressure required. The excess steam from the receiver will produce additional power in the L.P. cylinder, which latter has also a piston valve, operated by a separate but fixed eccentric. The regulating valve *R.V.* is a simple throttling valve, controlled by a pressure regulator, which will close this valve when the pressure in the bleeder line drops below the pre-determined limit.

The L.P. cylinder releases into a condenser or into the back pressure steam line of the factory, as may be desired.

A two-crank arrangement is also quite feasible, but in both instances, tandem or two-crank arrangement, there will be great flexibility, as the cylinder ratio can be designed to suit any required or prevailing condition. Moreover, the H.P. as well as the L.P. cylinders are double acting, so a uniform power output may be achieved without the necessity for a very heavy flywheel.

An alternative arrangement can be made by having the bleeder pressure acting on a H.P. throttle valve, so as to close this valve when the bleeder pressure becomes too high and opening the L.P. throttle *R.V.*, so as to allow the L.P. cylinder to draw steam from the bleeder line. Corliss or poppet valves are very suitable also for this kind of engine, and a very ingenious design, where the performance is done in one cylinder, is that of the MISSONG patents,[1] similar to the mixed pressure engine mentioned already in this chapter, but with the difference that compound working is achieved. Poppet valves are used in this design, the H.P. part being at the cover end and the L.P. part at the crank end of the cylinder. The cylinder ratio therefore is 1 : 1, thus reducing the flexibility of the conditions for which it can be used. A heavy flywheel, moreover, will be required, as the power output at both cylinder ends may be of varying magnitude.

From tests on such a Missong engine by Prof. Dr. C. PFEIDERER[2] a thermodynamic efficiency of about 70 per cent. has been obtained for non-condensing performance and about 75 per cent. for the condensing one. The steam pressure was about 150 lbs., with about 100°F. superheat and the bleeder pressure about 30 lbs. per sq. in., the vacuum for the condensing performance being about 27 in. The power output of the engine was about 200 h.p. max. The efficiency is thus higher than with a condensing engine alone.

Fig. 590 shows the H.P. extraction performance diagrammatically with a bleeder turbine. At the left side, the high pressure boilers supply steam at 300 lbs. per sq. in., which enters the turbine through the governor throttle *G* and is released from the H.P. part of the turbine into the bleeder line at 120 lbs. per sq. in. The low pressure boilers at the right hand side of the scheme are connected to this bleeder line and supply steam to the existing low pressure prime movers of the factory. The excess of this low pressure steam is led into the low pressure part of the turbine by the regulating valve *R.V.* Both low pressure engines and turbine exhaust into the back pressure line, having

1 German Patent, 240,718 of 1912. 2 See *Zeitsch. des Ver. Deutscher Ing.*, 1913, p. 2030.

7 lbs. per sq. in. pressure. The bleeder line pressure influences the main governor of the turbine as well as the regulating valve *R. V.*, and in case this pressure goes above 120 lbs., steam is drawn from the bleeder line into the low pressure turbine, the governor *G* being closed or throttled.

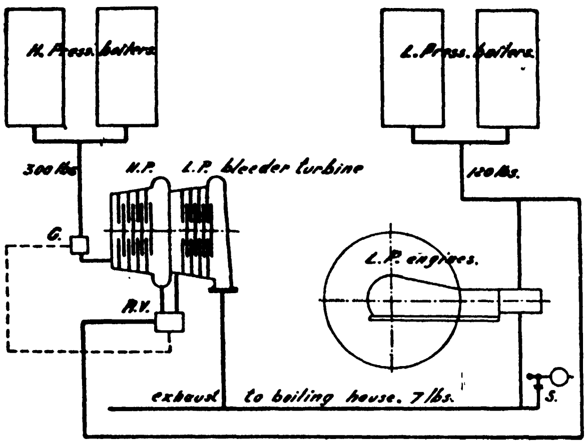


Fig. 590.—H.P. Extraction Performance.

A very flexible arrangement is thus achieved, but the back pressure of 7 lbs. is not controlled and a safety valve *S* is provided to relieve the exhaust line from excessive pressure, say of above 10 lbs. per sq. in.

A further arrangement of the bleeder principle, which has already successfully been applied in Hawaii[1] is shown in the diagram of *Fig.* 591. Use again is made of the indicator diagram, but the same principle applies equally for steam turbines.

The steam pressure of the boilers in the sugar factory is taken as 150 lbs. per sq. in., and a very small admission *a* of about 3 per cent. of the full stroke, being about 6 per cent. of the H.P. cylinder, which has an assumed ratio of 0·5 : 1, will release the steam at the end of the expansion performance slightly above the assumed vacuum of 27 in. prevailing in the surface condenser.

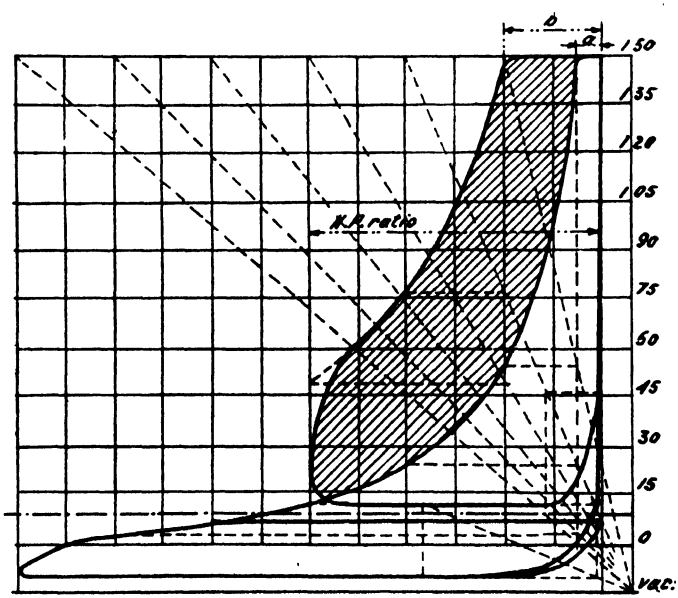


Fig. 591.—Condensing Extraction Diagram.

These small admissions or cut-offs are not very effective, as their regulation by the governor will be difficult at low loads, and therefore the bleeder principle has advantages, as the cut-off on the H.P. cylinder may be increased to the dimension *b*, being 33 per cent. of the H.P. stroke. The increase in power

1 See the paper of R. B. JOHNSON, read before the Association of Hawaiian Sugar Technologists in 1934.

output is equivalent to the shaded diagram area, whereas 80 per cent. of the total steam approximately is released into the back pressure line at 7 lbs. gauge pressure and the remaining 20 per cent. will enter into the L.P. cylinder, which is connected to the above-mentioned surface condenser. As assumed in all the previous diagrams, isothermic expansion performance is also adhered to in *Fig.* 591, although the author does not under-estimate the value of superheated steam and adiabatic expansion. It may be added that as the vertical H.P. division line in the diagram is at the intersection of the expansion curve belonging to the cut-off *a*, and the H.P. exhaust (at about 10 lbs. per sq. in.) there will be no loop formed, as indicated by *x* in *Fig.* 588, but the back or bleeder pressure may rise when the demand for exhaust steam is low.

The schematic layout is shown in *Fig.* 592, the boilers supplying steam at 150 lbs. per sq. in. to the H.P. section of the bleeder turbine as well as to the mill and factory engines of the piston type, which both exhaust at 7 lbs. per sq. in. As soon as there is any rise in pressure above 7 lbs., the governor *G* of the H.P. turbine will throttle and low pressure steam will be extracted from the back pressure line by the regulating valve *R.V.* to the L.P. turbine and condensed in the condenser, after having developed a considerable power output.

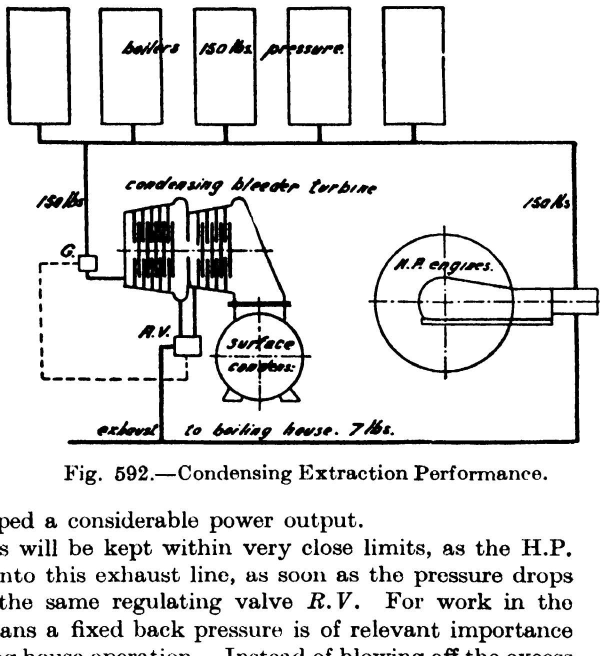


Fig. 592.—Condensing Extraction Performance.

The back pressure thus will be kept within very close limits, as the H.P. turbine will deliver steam into this exhaust line, as soon as the pressure drops below 7 lbs. by means of the same regulating valve *R.V.* For work in the evaporators and vacuum pans a fixed back pressure is of relevant importance in the efficiency of the boiling house operation. Instead of blowing off the excess exhaust through the roof, an appreciable power output has been added.

The turbine condenser is of the surface type and the injection water required for the barometric condensers of the boiling house is circulated through the surface condenser, before being pumped to the barometric ones. As the percentage of the total steam condensed is small, only a slight rise in temperature of the large amount of circulation water will result, and the water can be used, without any serious interference to the boiling house condensers.

The air extraction of the surface condenser can be achieved by means of a reciprocating or rotary air pump, as well as by steam jet extractors with intermediate and after coolers.

A small steam supply is always available for the low pressure turbine, so as to have a feed for the condensate pump. A by-pass of fresh water should also be provided for this condensate pump.

A connexion to the factory condenser, generally of the barometric type, has not been attempted, as it will impair the independence of the prime mover; moreover, the long exhaust lines would make such an arrangement less effective. The condensate in such a case would be lost as boiler feed-water and the separate condenser so far is the most practical solution.

As it will be of interest to know something about the efficiency and power output of the different systems just discussed, the author has drawn for this purpose several heat flow diagrams, all having an initial heat value of 5,000,000 B.Th.U. and dealing with hourly rates of steam. This information should assist in the proper selection of prime movers under the particular conditions ruling.

The first diagram, *Fig.* 593, refers to the normally used back pressure engines. Saturated steam is assumed in all the graphs and the following steam data are considered :—

Live steam pressure 100 lbs. per sq. in. gauge pressure
Exhaust steam 7 lbs. „ „ „

From the entropy chart for saturated steam, expanding from 100 down to 7 lbs. per sq. in., a heat drop of 122·4 B.Th.U. per lb. of steam used is found. The thermo-dynamic or total efficiency of the prime movers is assumed to be 60 per cent., a figure which might be higher for back pressure engines and lower for condensing engines, but as an overall assumption it represents a fair average. Turbines have a very varying thermo-dynamic efficiency, depending upon the construction, and the low-priced type of turbine used in sugar factories will not always give a total efficiency of over 50 per cent.

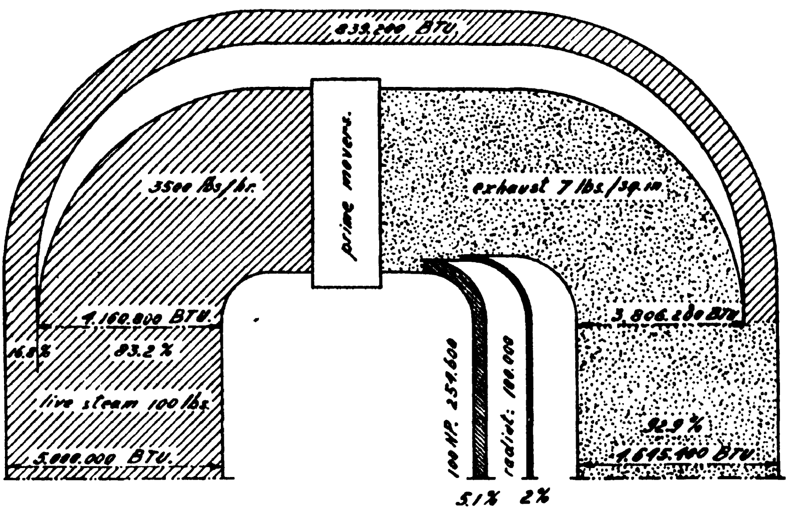


Fig. 593.—Back Pressure Heat Flow Diagram.

As the energy of one i.h.p. per hour has an equivalent heat value of 2546·2 B.Th.U., the steam consumption for these back pressure engines under the given steam conditions will amount to :—

$$2546{\cdot}2 \div (122{\cdot}4 \times 0{\cdot}6) \simeq 35 \text{ lbs. per i.h.p./hr.}$$

Assuming furthermore that 100 i.h.p. is to be developed, 3500 lbs. live steam will be required per hour for power purposes.

One lb. steam of 100 lbs. gauge pressure contains a total of 1188·8 B.Th.U./lb. and the 3500 lbs. steam thus contain :—

$$3500 \times 1188{\cdot}8 = 4{,}160{,}800 \text{ B.Th.U.}$$

thus leaving 5,000,000 — 4,160,800 = 839,200 B.Th.U. for heating purposes in the boiling house with live steam.

The engines do not deliver all the heat contained in the live steam, as there are the corresponding heat losses for power output, and moreover the radiation and cooling losses of the steam within the engine, so the exhaust steam will contain less heat than the live steam and the losses will be as follows :

Energy of 100 i.h.p = 100 × 2546·2 =	254,600 B.Th.U.
Radiation and cooling (average)	100,000 ,,
Total	354,600 ,,

Thus, in the exhaust steam 4,160,800 — 354,600 = 3,806,200 B.Th.U. are available, and the total amount of heat of 5,000,000 B.Th.U. has been used as follows :—

Energy....................	5·1 per cent.
Radiation	2·0 ,, ,,
Available for heating	92·9 ,, ,,
	100·0 ,, ,,

As one indicated horse-power requires 35 × 1188·8 = 41,608 B.Th.U./hr. and as there are 839,200 B.Th.U. going direct to the boiling house, so in the latter case 839,200 ÷ 41,608 ≃ 20 i.h.p. could be developed in addition to the 100 i.h.p. assumed.

The calculated figures are shown and drawn to scale in the diagram of *Fig.* 593.

For the mixed pressure engine (vide *Figs.* 586 and 587) no diagram has been prepared, but the calculation of the different values is given herewith, as well as the prevailing steam data :—

Live steam	150 lbs. gauge pressure
Exhaust steam.........................	7 lbs. ,, ,,
Vacuum in condenser	27 in. (barometric pressure at 30 in.)
Heat drop between 150 and 7 lbs. gauge	149 B.Th.U.
Heat drop between 150 lbs. gauge and 27 in. vacuum..	302 B.Th.U.
Steam consumption for non-condensing end	28·5 lbs./i.h.p./hr.
Steam consumption for condensing end	14 lbs./i.h.p./hr.

With the single cylinder mixed pressure engine, an equal load on both ends of the cylinder is advisable for smooth operation, but the condenser losses will be heavy. For highest efficiency the bleeder performance, shown in diagram in *Fig.* 591, is to be preferred. The two-cylinder arrangement will give greater flexibility, but the heat drop in the condensing cylinder is high and a lower thermo-dynamic efficiency will be the result. The uniflow engine, nevertheless, can be used to advantage for this condensing part, having a higher efficiency than the current counterflow engine.

Taking 100 h.p. again as the desired power output, of which 50 h.p. is to be developed in the non-condensing part and 50 h.p. in the condensing one, the steam requirements for this power output will be :—

Non-condensing, 50 h.p.	50 × 28·5 =	1425 lbs. steam/hr.
Condensing part, 50 h.p.	50 × 14 =	700 lbs. steam/hr.
Heat in steam for power,	2125 × 1195 =	2,539,400 B.Th.U.
Steam available for direct heating		2,460,600 B.Th.U.
		5,000,000 B.Th.U.
Energy loss, 50 h.p. non-cond.	127,300 B.Th.U.	
Radiation, non-cond.	50,000 B.Th.U.	
Energy and cond. loss 50 h.p.	836,500 B.Th.U.	
Total losses		1,013,800 B.Th.U.
Available for heating		3,986,200 B.Th.U.

Tabulated, the following percentages are obtained :—

Energy..................	5·1	per cent.
Radiation	2·4	„
Condenser loss	12·8	„
Available for heating	79·7	„
	100·0	„

This table shows clearly that the mixed pressure engine should not be used for equal loads on the condensing and non-condensing ends, as the available heat in the exhaust steam is considerably less than with the straight back-pressure engine.

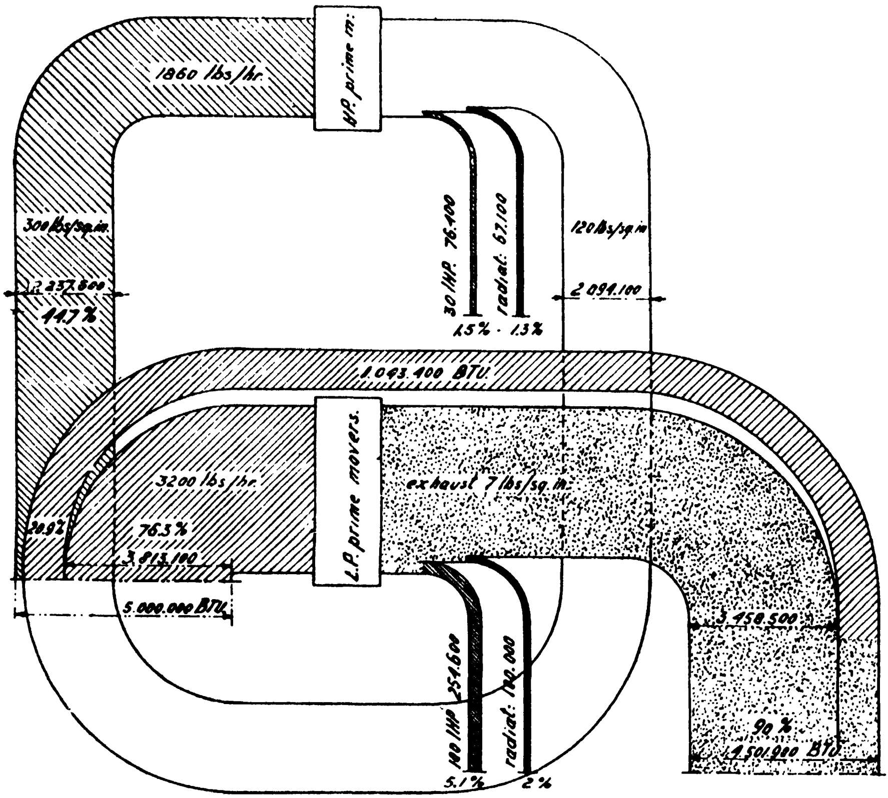


Fig. 594.—H.P. Extraction Heat Flow Diagram.

The two-cylinder mixed pressure engine will nevertheless be more efficient in those cases where a great power demand exists, and assuming 104 h.p. load on the non-condensing side, against 26 h.p. on the condensing side ; thus with an increase of 30 per cent. in developed power, the heat percentages are :—

Energy..................	6·6	per cent.
Radiation	2·9	„
Condenser loss	6·9	„
Available for heating	83·6	„
	100·0	„

On page 591 a table is given showing the maximum amounts of energy available and the corresponding available heat for the boiling house.

The high pressure extraction performance has a high efficiency, as will be seen from *Fig.* 594. Here steam conditions are as follows :—

Live steam		300 lbs. gauge pressure
Bleeder pressure		120 lbs. gauge pressure
Exhaust steam		7 lbs. gauge pressure
Heat drop between 300 and 120 lbs.		68·4 B.Th.U.
Heat drop between 120 and 7 lbs.		135 B.Th.U.
Steam consumption, H.P.		62 lbs./i.h.p./hr.
Steam consumption, L.P.		32 lbs./i.h.p./hr.
H.P. power, 30 i.h.p. — 1860 lbs. steam/hr.		2,237,600 B.Th.U.
H.P. energy loss	76,400 B.Th.U.	
Radiation	67,100 B.Th.U.—	143,500 B.Th.U.
Heat in H.P. exhaust		2,094,100 B.Th.U.
L.P. steam, 5,000,000 — 2,237,600 =		2,762,400 B.Th.U.
Heat available at 120 lbs. gauge		4,856,500 B.Th.U.
L.P. power, 100 i.h.p. — 3200 lbs. steam/hr.		3,813,100 B.Th.U.
Available for direct heating		1,043,400 B.Th.U.
L.P. energy loss	254,600 B.Th.U.	
L.P. radiation	100,000 B.Th.U.	
	354,600 B.Th.U.	

For heating in the boiling house, there remain available : 1,043,400 + (3,813,100 — 354,600) = 4,501,900 B.Th.U.

With this H.P. bleeder performance 30 per cent. additional power has thus been developed and there still remains about 90 per cent. of the total steam heat available for the boiling house or practically the same amount as with the back pressure system (*Fig.* 593). In maximum power output it cannot reach the limits of the mixed pressure scheme, but the performance will be useful in those instances where old boilers have to be replaced and the new boilers are of a higher working steam pressure, so as to be able to deliver the additional power required.

Finally, in *Fig.* 595 is shown the heat flow of the condensing bleeder performance, having the following steam values :—

Live steam		150 lbs. gauge
Exhaust steam		7 lbs. gauge
Vacuum		27 in. (at 30 in. barometric pressure)
Heat drop between 150 and 7 lbs.		149 B.Th.U./lb.
Heat drop between 7 lbs. pressure and 27 in. vacuum		173 B.Th.U./lb.
Steam consumption, non-condensing part		29 lbs./i.h.p./hr.
Steam consumption, condensing part		25 lbs./i.h.p./hr.
H.P. power in turbine	50 h.p.	
H.P. power in factory engines	70 h.p.	
Total	120 h.p.—3480 lb. steam/hr., equal to	4,158,600 B.Th.U.
H.P. energy loss	315,600 B.Th.U.	
H.P. radiation	120,000 B.Th.U.	
		435,600 B.Th.U.

Heat available in exhaust steam of 7 lbs. gauge	3,723,000 B.Th.U.
Condenser loss 25 h.p. for L.P. turbine, 625 lbs. steam/hr.	723,800 B.Th.U.
Heat available in exhaust steam for boiling house	2,999,200 B.Th.U.
Direct heating steam 5,000,000 — 4,158,600 =	841,400 B.Th.U.
Total heat in steam for boiling house	3,840,600 B.Th.U.

The total initial heat of 5,000,000 B.Th.U./hr. has thus been distributed as follows :—

Energy..................	7·6 per cent.
Radiation	3·4 ,,
Condenser loss	12·2 ,,
Available for heating	76·8 ,,
	100·0 ,,

So a 45 per cent. power increase has been obtained by reduction of about 16 per cent. of the available heat for the boiling house, as compared with the back pressure scheme (*Fig.* 593).

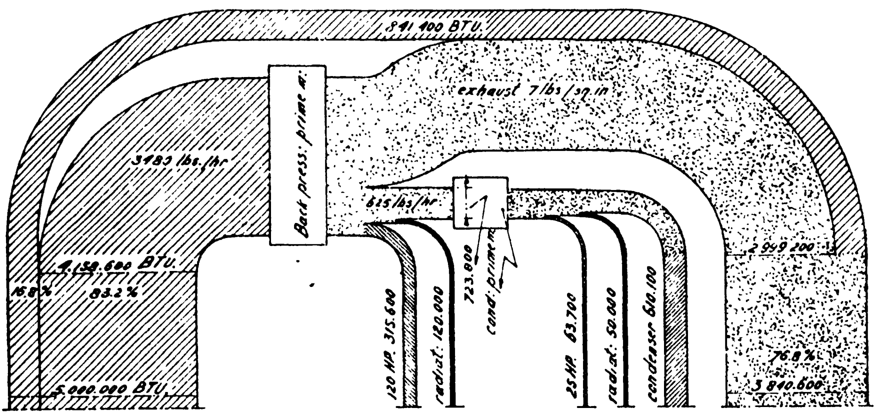


Fig. 595.—Condensing Extraction Heat Flow Diagram.

For increased power demand in the cane sugar factory, the mixed pressure, H.P. and L.P. bleeder performances with the condensing of a part of the steam in surface condensers, have been compared. The H.P. bleeder as well as the condensing L.P. bleeder performances have been put into successful operation in such factories, and wherever excess power is required or bagasse fuel has to be saved, the installation of this kind of prime mover should be considered.

In the following table the average results obtainable in respect to power output and available heat for the boiling house are arranged for the different systems as calculated in this chapter. The five vertical columns represent the following :—

A = Back pressure power generating (considered as common practice in cane sugar factories).
B = Mixed pressure power generating, half condensing, half with back pressure.
C = Ditto, 20 per cent. condensing, 80 per cent. back pressure.
D = H.P. extraction power generating, non-condensing.
E = Normal pressure, extraction power generating, condensing.

	A	B	C	D	E
Normal power developed, i.h.p. ..	100	100	130	130	145
Per cent. of total heat available for heating purposes in the boiling house	92·9	79·7	83·6	90	76·8
Additional available power with 7 lbs. exhaust pressure, i.h.p...	20	75	31	27	24
Maximum power obtainable from 5,000,000 B.Th.U./hr., i.h.p. ..	120	175	161	157	169
Per cent. of total heat available for the boiling house, when maximum power is developed	91	74·4	81·4	88·2	75·1

It will be seen that *A* has the smallest power generating capacity and *B* the largest. As regards the heat available for the boiling house, *A* is most favourable, but is closely followed by *D*, whereas *B* and *E* are less favourable, as a larger part of the steam is condensed in a surface condenser and the latent heat of the steam is thus lost for use in the factory.

D is most favourable as to both power generation and available heat, but it requires higher pressure boilers, when new boilers are required for extension or replacement of obsolete ones.

Where bagasse has a sales value, as for board making or kindred industries, the last four schemes will reduce the fuel requirements in bagasse and might thus result in direct profit.

The speed of piston engines should be between 150 and 240 r.p.m. and piston speeds of 12 ft. per second are now standard practice. In some instances even higher figures are allowable and uniflow engines are now built up to 18 ft./sec. piston speed.

In *Fig.* 596 the arrangement of a *Poppet Valve Engine*, direct-connected to an A.C. generator of 350 kw., 440 volts, 60 cycles and 180 r.p.m. is shown, this being supplied and installed by the author for the partial electrification of a 1200-ton cane sugar factory. The steam pressure is 150 lbs. at the main valve of the engine.

The engine floor is laid only 6 ft. 7 in. above the prevailing ground level because the soil at the site did not prove very firm, there being about 20 ft. of black earth underneath. It is therefore essential that the piston engine used be well balanced in respect to inertia forces acting in the reciprocating motion; for otherwise unpleasant vibrations will occur.

For flexibility, generally two units are installed, each having half the required power output, and a third unit of the same size is kept as a spare or for future extension. The governing mechanism of these prime movers has to be of such a type that the engines can be run for parallel or synchronous switching of the alternators.

Where condensing is practised, as explained before for extraction engines, or for the power unit for dead season operation when the exhaust steam is not required, then the *uniflow steam engine* can be applied. For extraction engines, only the L.P. cylinder should be designed according to the uniflow principle. For back pressure or non-condensing performance, the compression of the uniflow steam engine is too high in most cases; moreover, the mean effective steam pressure falls (see Chapter IX) and hence larger cylinder dimensions are required, while the thermo-dynamic efficiency falls also.

The general features of the uniflow performance can be learnt from *Fig.* 597, this being a *Sectional View of a Uniflow Steam Cylinder.* The piston *a* has a width of about 0·9 of the engine stroke and when at dead centres, the exhaust

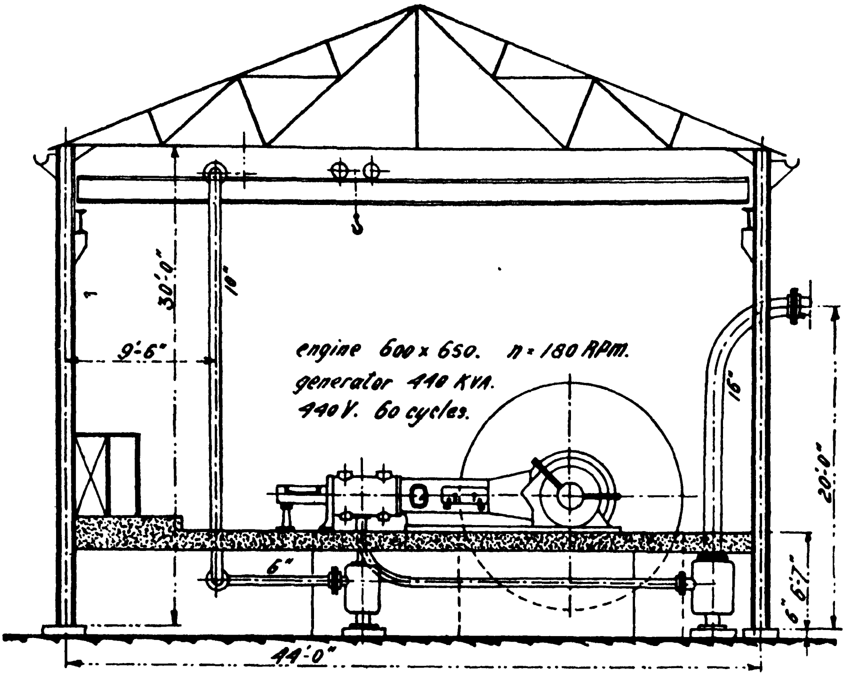


Fig. 596.—Power Plant with Poppet Valve Engine.

slots *c*, arranged midway of the cylinder course *b*, are opened. Around these slots a ring-shaped jacket *d* is provided, which connects with the discharge *j*. The exhaust steam is thus released by the piston and special discharge valves are not required. Moreover, the steam passes the cylinder in one direction only, from which fact the name "uniflow" has been derived. The admission valves, these being poppet valves, are arranged in the covers. By-pass valves *f*, which can be hand or automatically operated, are arranged for increasing the dead space of the cylinders by connecting the compression chambers *g*. In case the vacuum is not sufficient and the

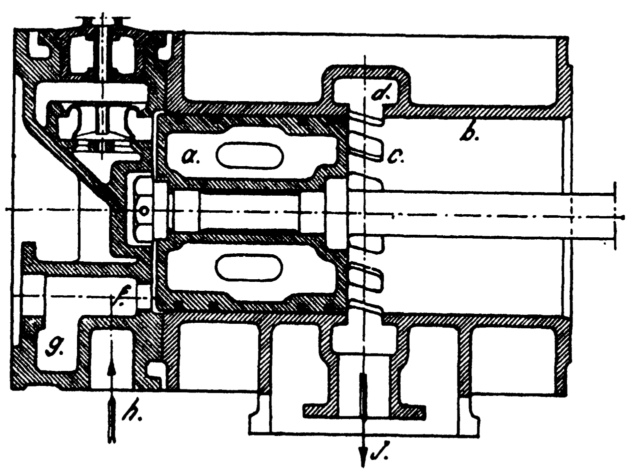
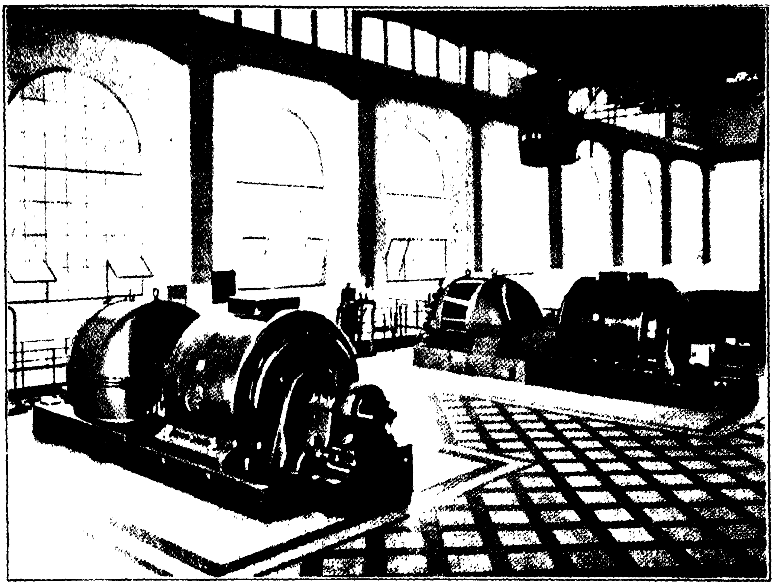
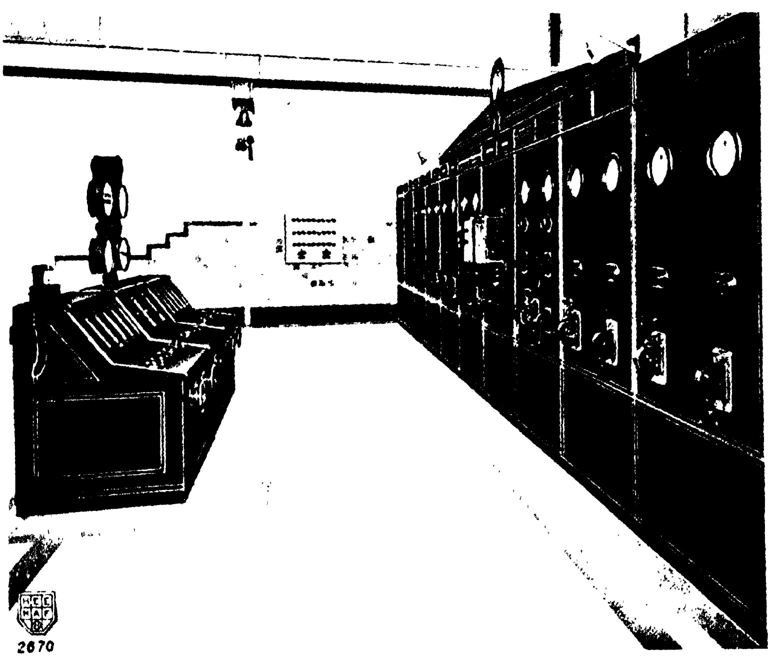


Fig. 597.—Cylinder of Uniflow Steam Engine.

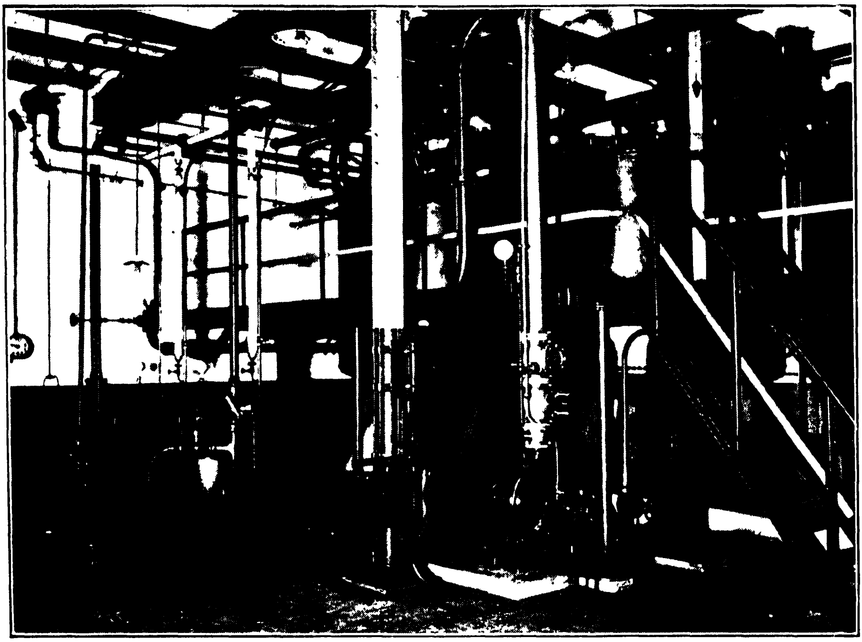


TURBO-ALTERNATORS IN POWER PLANT.
(U.C.M.A.S.)

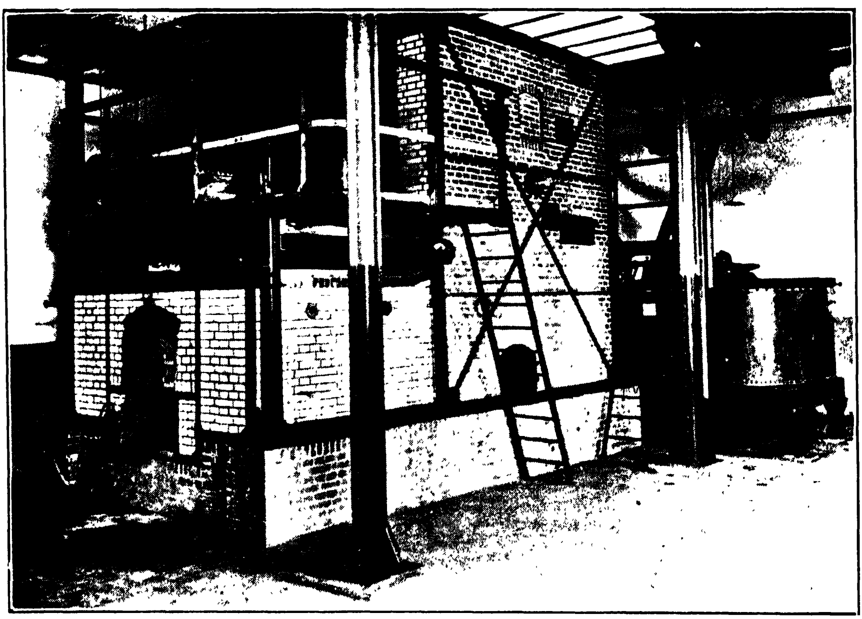


SWITCHBOARD AND OPERATING DESK FOR THE POWER PLANT OF A SUGAR FACTORY.

PLATES 132 & 133.



NORIT PASTE MIXING AND SODA TREATMENT IN A REFINERY.
(N.V. Norit-Vereeniging)



NORIT REVIVIFYING FURNACE.
(N.V. Norit-Vereeniging)

compression thus may become too high (which would stall the engine), the compression by-pass valves are opened. The live steam is admitted at the bottom side of the covers at *h*.

The uniflow engine is built as a piston or poppet valve engine ; its use in the Corliss type has not come to the author's knowledge. High speed and superheat can be used to advantage with this type of engine and a favourable thermo-dynamic efficiency is obtained in the condensing performance. Owing to their simple design, the cost price of these engines is lower than that of the four-valve type. For dead season power supply, the uniflow steam engine has been applied already in cane sugar factories.

Because of the space occupied by an engine and electric generator, which has a bearing upon the initial cost price, a steam turbine has an advantage in many cases, and sizes from 350 to 2000 kw. per unit are now built for cane sugar factories. (1 kw. = 1·341 h.p. = 1·360 metric h.p.). The number of revolutions varies between 3000 and 3600 per min. for A.C. of 50 and 60 cycles per sec. respectively.

For back pressure performance, these turbines are built according to the *impulse system*, where the expansion of the steam, thus the conversion of potential into kinetic energy, only takes place in the stationary nozzles or guide wheels. In the rotor blades there is no expansion and thus no pressure drop, the pressure on entrance and discharge sides of the turbine rotor blades being equal. The rotor of the impulse type turbine therefore is not subject to axial thrust.

With the *reaction type* the expansion takes place in both the stationary and the revolving blading, and axial thrust has to be balanced by special devices.

In our present-day turbine designs, both principles are not employed in their original form, but generally combined, the H.P. part being of the impulse type and the L.P. part connected to a condenser, sometimes of the reaction type.

Three famous designers of turbines and the principles they have selected are :—

LAVAL ....Impulse type of turbine, originally using one impulse wheel at high speed.

PARSONS ..Reaction type of turbine with a high thermo-dynamic efficiency for condenser performance at lower speed.

CURTIS ..Impulse type of turbine with several pressure stages.

In *Fig.* 598 is shown a diagram of an *Impulse Three-Stage Turbine*, used for back pressure, thus non-condensing, performance in cane sugar factories. The live steam at the first turbine nozzle 1 is assumed to have 150 lbs. gauge pressure and is expanded in the first stage to about 80 lbs. The steam velocity *V* increases from *m* to *o* and as this velocity cannot be converted into mechanical energy in one row of revolving blades, without impairing the efficiency, two rows of rotor blades 2 and 4 are provided with an intermediate row of stationary return blades 3. In the row 2, the velocity drops from *o* to *p* and in the second row (also called a velocity stage, differentiating from a pressure stage) from *p* to *q*. In the second (pressure) stage the velocity of the steam is raised from *q* to *s* in the nozzles or stationary blades 5. The kinetic energy is converted into mechanical energy in the rotor blades 6 and the steam velocity drops from *s* to *t*. In the final stage (third pressure stage) the steam velocity is raised again in the stationary blading 7 from *t* to *u*, to drop in the rotor blading 8 from *u* to *w* with delivery of mechanical power. The steam pressure has dropped in the second stage to about 45 lbs. and in the third to about 10 lbs. gauge, this being the assumed back pressure.

As there is no axial thrust in the impulse turbine, the thrust bearing serves only the purpose of keeping the rotor in place between the guide or return blading; rubbing will be detrimental to the safe operation of the tuibine.

Speed regulation of turbines is done by a centrifugal governor, which is combined with an oil pressure system for operating the governing valves. Throttling is mostly resorted to, but independent closing or cutting out of different nozzles or groups of nozzles will result in higher efficiency, as the nozzles have been designed for certain steam conditions, which cannot be altered withou altering the efficiency.

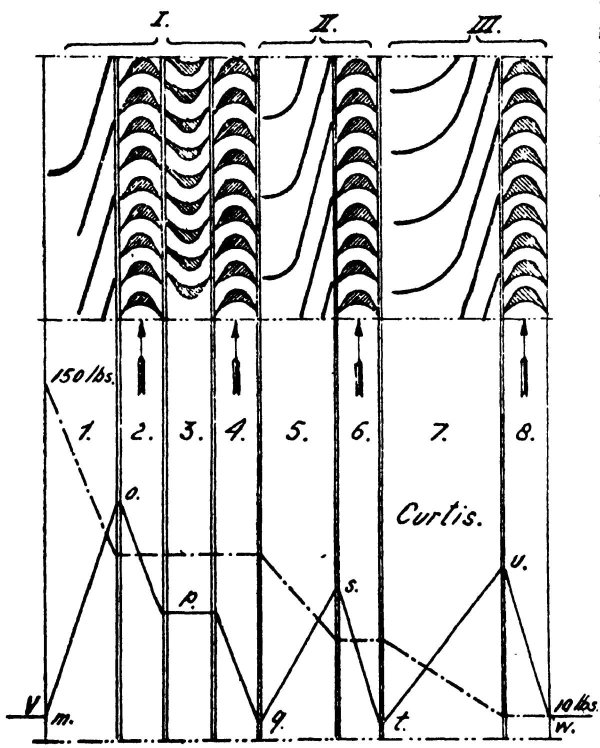


Fig. 598.—Three-Stage Impulse Turbine.

Any failure of the forced oil lubrication to the turbine bearings or any overspeeding will bring the turbine to a stop by automatic devices.

The oil in circulation is usually cooled in an oil cooler to ensure reliable operation, and filtration of the oil is also arranged.

By a special spring-tightening attachment the governor can be set for *variable speeds* and thus variable frequency of the alternating current, this being a patented feature of a well-known American manufacturer for obtaining varying mill speeds with electrically driven mills, as explained in Chapter IX.

The turbine blading should be made of stainless material to avoid corrosion. A moderate superheat is always advisable so as to obtain dry steam. The turbine shaft has packing glands with carbon rings, labyrinth rings or a water seal; friction, of course, has to be very low at these high speeds, as otherwise overheating will result.

When a turbine is ordered, the steam consumption, or the so-called water rate, as well as the thermo-dynamic efficiency should be known, as it may often happen that a little less efficiency is off-set by other advantages and a difference in price. When superheat is applied, the steam consumption in practical operation can be approximately ascertained by the steam temperature and pressure at the main valve and the steam pressure, temperature and moisture at the exhaust end (see also Chapter XII for measuring the steam moisture), and thus the heat drop measured. The thermo-dynamic efficiency of the prime mover has to be known in this case.

3.—Diesel Power.

During the dead season in cane sugar factories Diesel power is frequently used, as it does not require a boiler, and moreover the thermo-dynamic efficiency of this prime mover ranks very high as compared with the non-condensing steam performance and even with the condensing one. The initial cost as compared with a condensing steam plant is low, and often there is no fuel available at the time other than what can be bought on the market.

The efficiency account of the non-condensing, the condensing and the Diesel power should therefore be compared :—

Non-Condensing Steam Performance :

Let the following conditions be assumed : (metric measurements)

Live steam pressure .... 11 kg./cm^2 abs. (142 lbs. gauge)
Temperature superheated steam .... 250°C.
Exhaust steam pressure (exh. in free atm.). . 1·1 kg./cm.^{2} abs. (free exhaust)
Heat drop .... 101 Cal./kg.
Thermo-dynamic efficiency of engine .... 0·75
Effective heat drop $0{\cdot}75 \times 101 =$ .... 76 Cal./kg.
Steam consumption per metric h.p., $632 \div 76 =$ 8·32 kg./hr.
Heat content superheated steam .. 736 Cal./kg.
Heat in boiler feed-water .... 90 Cal./kg.
Heat to be produced in boiler .... ———— 646 Cal./kg.
Heat required for 1 mtr. h.p., $8{\cdot}32 \times 646 =$ .. 5365 Cal.
Boiler efficiency .... 0·75
Heat value in fuel for 1 metric h.p., $5365 \div 0{\cdot}75 = 7153$ Cal.
Total efficiency non-condensing perf. $632 \div 7153 = 8{\cdot}8$ per cent.

Condensing Steam Performance :—

Same steam pressure as above
Condenser pressure .... 0·1 kg./cm.^{2} abs. (27 in. vacuum)
Heat drop .... 180 Cal./kg.
Thermo-dynamic efficiency of engine .... 0·60
Effective heat drop ....$0{\cdot}60 \times 180 =$ 108 Cal./kg.
Steam consumption per metric h.p., $632 \div 108 =$ 5·85 kg./hr.
Heat in superheated steam ...... 736 Cal./kg.
Heat in boiler feed-water ...... 40 Cal./kg.
Heat to be produced in boiler .. .. ———— 694 Cal./kg.
Heat required for 1 metric h.p., $5{\cdot}85 \times 694 =$ 4060 Cal.
Boiler efficiency .... 0·75
Heat value in fuel for 1 metric h.p., $4060 \div 0{\cdot}75 =$ 5280 Cal.
Total efficiency condensing performance
$632 \div 5280 =$ 12 per cent.

Diesel Performance :—

Diesel oil consumption .... 190 grm./metric h.p./hr. (0·419 lbs.)
Heat value of Diesel oil .... 10,000 Cal./kg.
Heat value of 190 grm. Diesel oil .... 1900 Cal.
Total efficiency of Diesel performance
$632 \div 1900 =$ 33 per cent.

Comparing now coal as the fuel for the steam performance, and considering the heat value of the latter as 7500 cal./kg., there can be produced from one kg. coal per hour :—

Non-condensing performance	1·05 h.p.
Condensing performance	1·42 h.p.

and from 1 kg. Diesel oil :—

Diesel performance	5·26 h.p.

The prices per kg. or lb. of coal should thus be proportioned to the kg. or lb. of Diesel oil, as follows :—

Non-condensing steam/Diesel performance	1 ÷ 5
Condensing performance/Diesel	1 ÷ 3·7

In the British colonies coal is now sold for about 30 shillings per ton of 2240 lbs. and Diesel oil costs 4½ pence per Imp. gal. or about 8·5 lbs. The price per lb. is thus respectively 0·16 and 0·53 pence, thus in the proportion of 1 to 3·3. In other countries other prices may prevail, but the efficiency account can be calculated the same way.

The lubricating oil consumption of Diesel engines, especially the high speed types, is between 3 and 5 grms. per h.p./hr. and from 1 to 0·5 gr./h.p./hr. or less for steam engines, and still lower for turbine performance.

Estimates for comparison of capital outlay and yield should be based upon 5 per cent. yearly interest as an average and 5 per cent. amortization for steam plant and 8 per cent. for Diesel engines. For spare parts a higher allowance has to be made for Diesel engines and the respective figures should be obtained from the manufacturers concerned. For maintenance charges the turbine will show a low figure, when the blading is of resistant material, and has not to be renewed at an early date.

The *Diesel principle* comprises either 2-stroke or 4-stroke performance and single acting cylinders are exclusively used for the requirements in cane sugar factories. The 2-stroke type is used for small plant and for very large engines, but the 4-stroke type calls for discussion here, as it is more economic in fuel consumption, as well as in lubricating oil, although the 2-stroke *Diesel* engine will be cheaper per h.p. output.

Moreover, only the *airless* or the Diesel engine without air compressor needs to be mentioned, as this now is the most efficient type (it is also called the *solid injection* type). Semi-Diesels, which do not have sufficient compression to give the necessary heat for spontaneous ignition of the fuel, are losing ground in favour of the true or full Diesel engine.

From *Fig.* 599, where a modern *High Speed Diesel Engine* is shown, the 4-stroke principle can be easily explained.

1st stroke (downwards) The air is aspirated through the air-filter *a* and the inlet valve not shown, which is in line with the outlet valve *c*.

2nd stroke (upwards) ..The air is compressed to about 30 atm. (about 440 lbs./sq. in.) and when the piston reaches the top dead centre, the fuel is injected by the atomizer *b* into the antechamber c at about 80 atm. (about 1170 lbs./sq. in.) This atomizer is specially constructed to prevent after-dripping, which causes carbonization The compression temperature is over 500°C. (932°F.)

3rd stroke (downwards)..The fuel is ignited by the high compression temperature and gradually burnt in the cylinder thoroughly mixed with the air. Complete and smokeless combustion takes place. As this occurs gradually, there is no "explosion," but a rapid burning of the fuel. Through the heat of combustion, the gases are expanded and mechanical energy is delivered through the pressure of the gases upon the piston *d*.

4th stroke (upwards) ..The expanded combustion gases are released and forced out by the upgoing piston through the discharge or exhaust valve *e*, into the water-cooled exhaust main *f*. The cycle is then repeated.

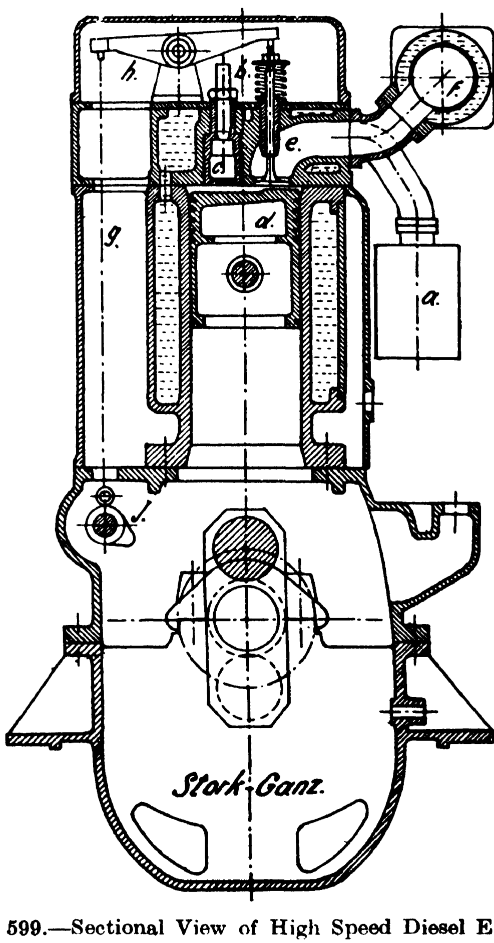


Fig. 599.—Sectional View of High Speed Diesel Engine.

The cylinder is jacketed for cooling by water supplied by a special pump, as is the cylinder head, to keep these engine parts at a reasonable temperature. The cooling water ratio is from 5·5 to 6·5 Imp. gals./h.p./hr. From the above-mentioned 1900 cal./h.p./hr. obtained from the combustion of the fuel, about 750 cal. have to be taken up by the cooling water and the remaining 518 cal. are discharged by the exhaust (632 cal. are the equivalent of one h.p./hr.).

The gas temperature at full load is about 380°C. (716°F.) and about 295°C. (563°F.) at three-quarters load. The exhaust gas temperature of all cylinders should be equal and high temperature at the cylinder exhaust indicates a good combustion, providing the cooling water pump is functioning.

The cooling water should preferably be soft and free from impurities. Hard water will cause incrustations in the cooling jackets of cylinders and valve heads, which have to be removed periodically by an acid solution.

The inlet and exhaust valves are operated by spring-loaded valve rods *g* and levers *h*, operated by the camshaft *j*, which is revolved at half the speed of the crankshaft by means of a set of gears. The crank case *k* is deep, so the circulating oil will not be splashed by the connecting rod heads; the oil level remains below their prescribed circles. An automatic displacement pump delivers the oil at about 3 lbs./sq. in. to the crank shaft and crank pin bearings ; from the latter it flows by a central bore through the connecting rod around the piston pin and then is splashed against the piston bottom on the inside, giving efficient cooling. The circulating oil is filtered by a duplex filter and cooled by an oil cooler. The engine is totally enclosed and dust cannot enter into the moving parts.

The engine shown has 150 mm. (about 6 in.) cylinder diameter and 185 mm. (about 7¼ in.) stroke. It develops at 1000 r.p.m. 18 h.p. per cylinder and the piston speed is about 20 ft. per sec. This is not an excessive figure, as there are special stationary engines running at 40 ft./sec. piston speed. The wear of the bearings is very small, when good lubricating oil is used; 0·01 mm. (0·0004 in.) after 2400 hours operating is a common figure.

The engine is started by hand (up to about 20 h.p.) and beyond that by compressed air or a small electric starting motor, which is fed by an accumulator, charged by a small dynamo during the run of the motor. It should be recollected that the engine can also be started by carbonic acid gas (carbon dioxide) in compressed state ; but starting by means of compressed oxygen is *very dangerous*, as the engine may burst, with disastrous consequences.

The minimum number of revolutions at which efficient operation is ensured lies around 600 r.p.m. The engine is started from the cold by a special patented valve lifting device, which causes the entering air to be heated by mechanical action.

The foundation block of concrete should comprise about 5 to 6 cub. ft. per h.p. developed and the soil bearing should be low. The 4-cylinder engine yields smaller reacting vibratory forces on the foundations than the 2- or 3-cylinder type.

CHAPTER XXXV.

ELECTRIFICATION.

LIGHTING — ELECTRICAL POWER DISTRIBUTION.

The principal advantages of electrification in cane sugar factories have been cited in the previous Chapter ; the lighting requirements during the night shift are necessarily of secondary importance compared with the demands for power.

Whether direct current or three-phase alternating current be selected, there are here a few points claiming our consideration. For lighting nowadays, incandescent lamps are arranged between one phase and the neutral of the alternating circuit as well as between the positive and negative conductors of the direct current system, so it is a matter of efficiency and cost which kind of current be used. For lighting alone direct current has been generally used with small transmission distances, but as soon as large power consumers (motors) enter the circuit, alternating current is mostly employed. For power transmission, three-phase alternating equipment has considerable advantages for easy transformation from high to low voltage and *vice versa*, as well as in the absence of commutators. Direct current power transmission, therefore, is only to be found where speed regulation is essential or a heavy starting torque is required of the motors. In cane sugar factories, carrier drives, centrifugal pumps, sugar centrifugals and kindred drives do not require speed regulation, and the starting torque requirements are as a rule not much above the normal full load torque. Alternating current for these drives can thus be used to advantage. For mill drives (see Chapter IX) a few designers have applied direct current, but most of the electrically-driven mills are coupled to three-phase alternating current motors and the constant speed has proved to be satisfactory or else the variable frequency system is applied.

1.—Lighting.

As the conductors for lighting as well as for power supply are calculated upon Ohm's resistance, the higher the voltage the less will be the area of the wires required. For economical reasons 220 volts is therefore the standard current. Good insulation of conductors and wiring devices is essential as the higher the voltage the more danger exists with unprotected cables. This explains also why manufacturers make 3-phase A.C. transformers for 20 v. to 25 v., to be used in combination with hand lamps.

When ordering incandescent bulbs, the maximum voltage should be given, as these do not support a higher voltage than that for which they are rated.

For the prime movers coupled to the electric generators the degree of uniformity should be between 150 and 300, and for turbines and Diesel engines with from 3600 to 600 r.p.m. a high degree of uniformity is easily obtained.

The author has known cases where the bulbs in the factory had been removed by the factory operators, to be used in their living houses, so the factory lighting system was arranged for 220 and the living houses for 110 volts, to make this purloining of bulbs impossible. Locking devices may be a sufficient protection in some instances.

The unit of lighting is the *candle power*, an arbitrary figure from a pentane or amyl-acetate (pear oil) lamp. The flux of light is measured in *lumens* and in the metric system one c.p. produces $4\pi l$ at 1 m. radius, One l per sq. metre is denominated a *lux* in continental Europe. Full moon gives a lighting intensity of about 0·25 *lux* and a few average values are given for factory lighting :—

	Lux
Yard around the cane unloader and weighing scales	1·5—3
General yard lighting	0·5—1·5
Mill house	25—35
Rest of factory	15—20
Offices and laboratory	35—50
Drafting room	60—80
Porch of dwellings	6—10
Living rooms	15—25

Thus : *lux* × sq. m. = *lumens*. Incandescent lamps nowadays are also rated in *lumens* (*deka-lumens*).

The height of the lights above ground level, when not required for special purposes, should be about 0·7 of the diameter of the area to be lighted.

The incandescent bulb has replaced in nearly every instance the arc lamp and it can now be had gas-filled up to over 1500 watts. With gas-filled bulbs the average current consumption is about 0·5 watt per c.p./hr. or about 0·4 watt per *deca-lumen*/hr. The average life of a good bulb burning at the rated voltage is over 1000 lighting hours.

The cables carrying the current are generally measured in *mils*. A wire having a diameter of 0·040 in. or 40 mils, has an area of 40 × 40 = 1600 *circular mils*. Insulated wires will carry 1 ampere per 500—2000 circular mils, the lowest value for the thinnest wire. In continental Europe the current-carrying area is given in mm² and 1·5 to 6 amp./mm² is a normal figure.

Lighting circuits have to be protected by *fuse plugs*. Enclosed fuses in paper or porcelain tubes can be had up to any required current intensity. The fuse wires or strips in the porcelain tube are renewable.

Electric wires for lighting should of course be insulated and be supported by porcelain insulators. Switches are of the push button, the rotary or the lever type. They are nowadays made of bakelite, the modern insulating material.

2.—Electrical Power Distribution.

Electrical generators and motors can be had from manufacturers who produce machines well designed for the purpose required, and reliable and continuous operation during crop time is well assured with electrical power. Sometimes, especially during the dead season, outside current is bought from public service companies.

The use of electrical generating units in sugar factories is increasing, and the larger factories will require larger units. A list of alternating current generators in Cuba of a number of known factories has been compiled by the author. They comprise :—

14	generators of	300 kw.	40	generators of	1000 kw.
24	,,	500 kw.	6	,,	1250 kw.
6	,,	600 kw.	14	,,	1500 kw.
12	,,	750 kw.	6	,,	2000 kw.

and it will be noted that the 1000-kw. generator has found a wide application. But it should not be overlooked that the factories which are included in the above list are for the largest part above 2500 tons cane grinding capacity per 24 hours.

Equal importance should be given to the power distribution system as to the generators and motors, since reliable operation depends greatly on the former.

The *Switchboard* in the power plant should be of an open design and have sufficient panels for efficient location of meters and switches, so as not to crowd these—a matter of no little importance for good supervision. The switchboard proper does not carry conductors or apparatus for heavy current, as the switch handles, etc., are connected by insulated mechanical transmitters to the circuit breakers located behind the switchboard or else in the basement of the power plant. The switchboard proper has to be connected to earth, as have all motor housings and the ironclad switch boxes inside the factory. The zero conductor of the star-connected generator is attached to a large copper *earth plate,* which has to be laid below ground water level. The different motor housings, etc., are earthed by one or several earth plates inside or close to the factory. In doubtful cases as to where the ground water level may recede, the author has driven in copper tubes, 20 ft. long, at the bottom of the excavation for the earth plates and attached them firmly by soldered copper wire to the latter (old heater tubes can be used).

Volt, ampere and kilowatt meters are to be found on the switchboard. Kilowatt-hour counters or totalizators are seldom used, but kilowatt recording meters are very useful for indicating the fluctuations in power supply and demand of the different sub-stations in the factory.

There is considerable advantage in automatic regulation of the voltage. For large circuit breakers of above 500 amp. the oil-immersed type, which prevents arc formation, is to be preferred.

Adequate protection for short circuit and overload should be provided to protect motors and generators. The *automatic overload circuit breaker,* having overload relays on two or better three phases of the alternating current system, is essential. Further protection has to be arranged by means of fuse bars.

The usual material of the switchboard nowadays is specially coated sheet steel. Marble has become obsolete in most cases, as fittings carrying current are no longer located on the panels proper. Marble has the inconvenience of absorbing dirt and easily breaks during shipment.

In *Fig.* 600 the *General Arrangement of Power Distribution* for a cane sugar factory (installed and furnished through the author) is shown. It is a partial electrification, and is built for future extension. The flow of the current is easy to follow :—

From the generator *a* the current passes the automatic circuit breaker *b,* having indicator lamps at *c*. The transformer *d* supplies low voltage current to the ampere, kilowatt and volt meters. A phase switch *e* is for measuring the current in each of the phases. The main bars *f* deliver the current to the different circuit breakers *h* of the sub-stations, each provided with a set of fuses *g*. The threefold conductors in the scheme are indicated by single lines.

Circuit 1 is connected by a subterranean cable to the ironclad switch box III inside the factory, which has a circuit breaker and a set of fuses, one for each connected current consumer. The connecting cables are of ample size and the ironclad boxes can be easily extended by connecting one or more fuse boxes. Circuit 2 connects to box II for five motors connected and one spare, whereas circuit 3 connects to box I for one motor. It will be seen that provision has been made for future extensions by spare connexions. Circuits 4 and 5 connect to the transformers for the 220-volt yard and factory lighting, each of 25 K.V.A., whereas Circuit 6 is connected to a 15 K.V.A. transformer for 110 volts for the staff and operators' dwellings.

Circuit 7 connects to box VI for supplying one large motor, Circuit 8 to box V for two outside motors, and Circuit 9 to box IV with two motors connected. Circuit 10 goes to a 4 kw. rotary transformer for supplying direct current of 110 volts to a magnetic tramp iron separator *k*, with a starting rheostat *j*. The direct current circuit is also laid on the switchboard.

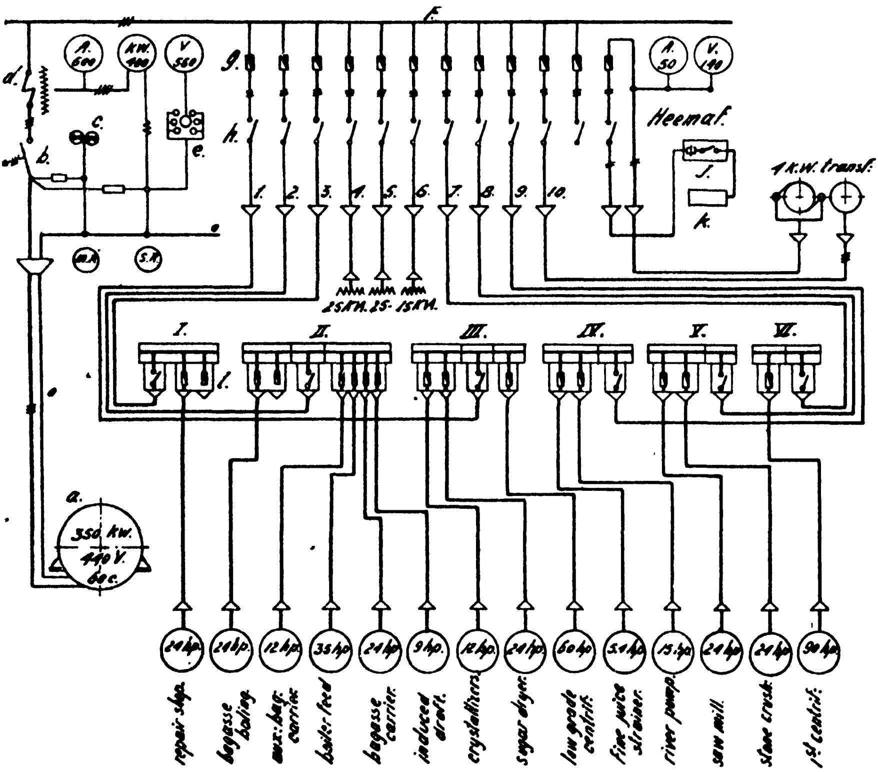


Fig. 600.—General Arrangement of Electrical Power Distribution.

The dead season Diesel generator supplies the same alternating current of 440 volts as does the crop generator, and is also connected to the same bars *f*. Each circuit now has a fourfold protection, viz. :—

(*a*) The automatic circuit breaker *b*.
(*b*) The switchboard fuses *g*.
(*c*) The fuses in the ironclad switch box *l*.
(*d*) The overload relays at the motor controller.

This installation has worked very satisfactorily under adverse conditions of humidity of the surrounding atmosphere and low skill of the operators. All cable connexions have been made in special trifurcating boxes, filled with a hot asphalt mixture to avoid entrance of moisture into the cable insulation proper.

When more than one alternating current generator has to work on the same circuit, the generators have to be *connected in parallel,* but this can be done without danger only when there is *synchronism* in phase, frequency and voltage. Bulb indicators are used to show when this synchronism exists. Automatic parallel switching can also be had.

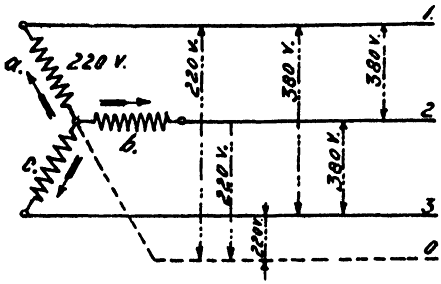


Fig. 601.—Star Switching.

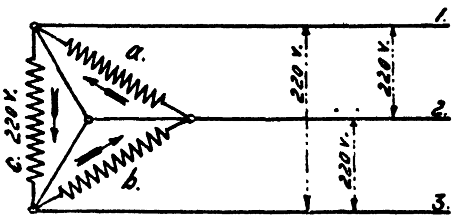


Fig. 602.—Delta Switching.

It is very important in an alternating current circuit that the motors are not of an over-size, as this will lower the power factor (see Chapter IX) and load the coils of generators and motors beyond what is required for the power supply proper. To correct the unfavourable power factor, *static condensers* or *synchronous A.C. motors* have been applied. The latter can be used for driving the injection and air pumps of the factory condensers.

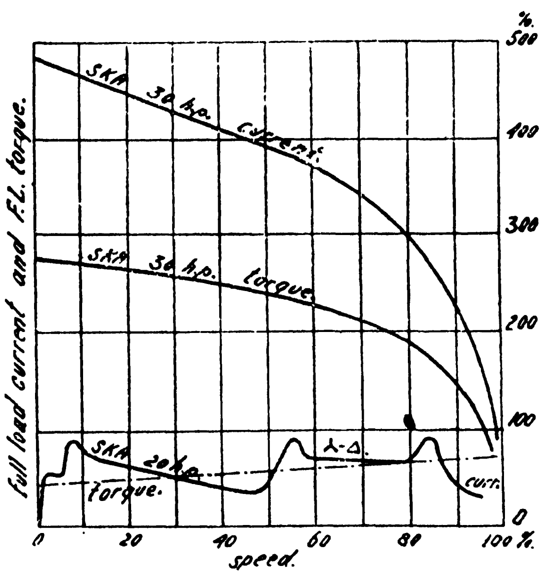


Fig. 603.—Torque-Current Diagram.

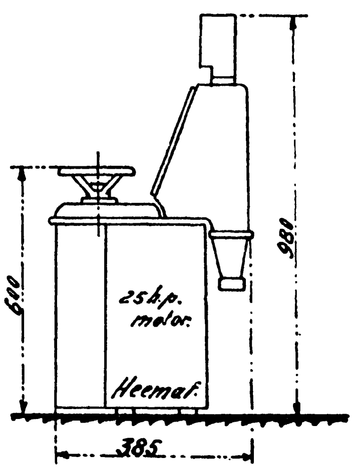


Fig. 604.—Oil-immersed Star Delta Controller

Direct current motors are started by resistances and alternating current motors by special star-delta controllers. To explain the *star and delta switching, Figs.* 601 and 602 are given. With star switching (*Fig.* 601), the three coil groups *a*, *b* and *c* of the three phases are interconnected centrally and when all three phases carry an equal share of the load, there will be no current at this zero-point and in the zero conductor, which generally is earthed. As soon as one phase does not carry the same current load as the others, the zero-conductor carries current. If the coil groups have a tension of 220 volts, then the tension between two phases will be $220 \times \sqrt{3} = 380$ volts and between a phase and the zero conductor 220 volts. With delta-switching, as shown in *Fig.* 602, the three coil groups *a*, *b* and *c* of the phases are interconnected in triangle. Each coil group is self-sustaining or balanced, and any difference in phase load does not require a zero-conductor.

In general terms, with star-switching the voltage is higher and the intensity in amperes lower, whereas with delta switching the voltage is normal and the current intensity $\sqrt{3}$ times higher. Use of the former is made with A.C. generators, and also for starting A.C. motors in star with a low initial current intensity, whereas delta connexion is for the running performance.

Characteristics for starting current and starting torque should be in agreement with the duty the motor has to perform, and in *Fig.* 603 is given the starting current and starting torque for a special double squirrel-cage motor directly switched on the line and a similar one switched by means of a star-delta controller. The dotted line gives the required starting torque of the latter. The current and torque are given in per cent. of full load current (F.L.C.) and full load torque (F.L.T.).

In *Fig.* 604 is shown the outside view of an *Oil-immersed Star Delta Controller* for starting A.C. motors. On the superstructure an ammeter is arranged, as well as an overload relay on at least two phases, a non-current relay and a timing relay for starting the motor. These controllers have given very good operating performance. For small A.C. motors air-cooled starters are used, up to about 10 h.p. Each case, nevertheless, should be discussed with the manufacturers to ascertain which kind of starting gear is to be provided.

In an electrically-driven factory, with the exception of the mills, the average power distribution has proved to be as follows :—

	Per cent.
Mill house auxiliaries	7
Boiling house auxiliaries	25
Injection and vacuum pumps	34
Crystallizers and molasses	4
Centrifugals (belt-driven)	18
Water service	1
Lighting	5
Repair shops	4
Power house auxiliaries	2
	100

CHAPTER XXXVI.

REPAIR SHOPS.

CARPENTER'S SHOP — MECHANICAL REPAIR SHOP — WELDING.

Cane sugar factories as a rule are located far from industrial centres and the continuous operation of the factory requires a fairly well equipped shop for both carpenter's and mechanic's work.

Nevertheless, this workshop should not grow into a machine shop, as there must be a limit, for economic reasons of capital outlay and efficiency in the first instance ; and, secondly, a proper machine shop will turn out a better piece of workmanship, when it comes to certain sizes. But in times of depression this rule will not invariably be adhered to ; in such cases the designing and drafting departments generally lag far behind the potentialities of the shop. A true cost price calculation, also, is not very practicable, as several employees are on the pay roll during the dead season, whose salaries are charged under administration expenses, although they actually perform their duty mostly on behalf of the shop.

In case of newly built factories it is important to start with the repair shop and the auxiliary power unit for driving it and/or lighting the premises later on.

An attempt is made here to give the overall requirements of these shops, although the author does not profess to be able to satisfy the wishes of all operating engineers. Space, therefore, should be allowed for individual requirements of each particular factory.

1.—Carpenter's Shop.

The work to be done under this sub-heading may comprise :—

(*a*) The building of factory buildings of wood.
(*b*) Wooden floors, stairs and railings.
(*c*) Dwellings of wood and brickwork.
(*d*) Moulding and pouring of concrete foundations.
(*f*) Wooden railway and passenger bridges for general traffic.
(*g*) Sawing and preparing of raw lumber.
(*h*) Railway tie cutting and drilling.
(*j*) Cane car repairs and railway sundries.
(*k*) Cane hoists in the fields.
(*l*) Weighing platforms for cane in the fields.
(*m*) Wooden hoisting gear for heavy machinery, when a crane is not at hand.

Where a sufficient supply of hard wood is available close to the factory a *saw mill* may turn out to be a paying proposition ; it has to be equipped with a large circular saw having a diameter at least more than twice the size of the logs to be sawn. The customary draw table for the logs has also to be provided with an automatic feeding device. The saw should be equipped with renewable teeth and it should be well guarded by railings, etc. The logs are laid in the open, but the planks should be stored under a roof.

The saw mill building occupies about 60 × 15 ft. and up to 25 h.p. is required for driving it.

If railway cars have to be repaired, jacks or hoists should be provided, so as to lift the cage or car body from the trucks.

The following list of equipment is not obligatory for the carpenter's shop, but most of it may prove useful :—

1.—*Swing Circular Saw* for cutting planks and beams crosswise, which is hung from the ceiling. Armlength up to 8 ft. Saw 18 in. dia. revolving at 2000 r.p.m. Driving pulley 500 r.p.m. 5 to 10 h.p.

2.—*Band Saw* with wheels from 36 in. to 27 in. diameter, revolving at from 425 to 500 r.p.m. resp. Power consumption, 3 to 2 h.p.

3.—*Circular Saw Bench.* Saw 16 in. dia. 2700 r.p.m., driving at 525 r.p.m. 5 h.p.

4.—*Universal Wood Worker.* This is a combination of band saw, saw table, jointer, shaper and boring machine. 800 r.p.m. 5 to 7 h.p.

5.—*Jointer* for planing one side by hand. Width of revolving knife 16 in. Countershaft 900 r.p.m. Cylinder knife head 4000 r.p.m. 3 h.p.

6.—*Planer* for 24 in. boards up to 8 in. thickness. Countershaft 900 r.p.m. Cylinder 4200 r.p.m. Feed 20 to 30 ft./min. 5 h.p.

7.—*Borer and Chisel Mortizer.* ½ in. to ¾ in. mortizing capacity. 4 in. stroke. 500 r.p.m. 1 to 2 h.p.

8.—*Post Borer.* Vertical travel 10 in. Spindle 1000 r.p.m. Driving pulleys 500 r.p.m. 1 h.p.

9.—*Turning Lathe.* When patterns have to be made for one's own or a nearby foundry. To be mounted on wooden bench. 20 in. swing. Countershaft 800 r.p.m. 1½ to 3 h.p.

10.—*Stone Crusher* of the jaw type, 8 in. × 16 in., capacity approximately 10 tons crushed stone per hour. Requires about 12 h.p. Broken stone for concrete and railroad ballasting.

11.—*Concrete Mixer* with lifting device. 10 cub. yd. capacity per hour. 30 batches/hr. Power 8 h.p.

Each carpenter should have his own bench and be well provided with tools. In some countries the carpenters have to provide their own, in which case this expense does not fall on the factory. The author has known foremen carpenters who made use in a very efficient way of equipment thrown out from the mechanical repair shop.

For field railroad work, some large sugar factories employ a steam or motor shovel, and for road work a road roller, steam or motor-driven.

2.—Mechanical Repair Shop.

In these shops for both the factory proper and the transport department a number of operations are required :—

(*a*) Cutting heavy beams and plates.
(*b*) Cutting pipes, squares, rounds, angles and tees.
(*c*) Turning shafts and bushings.
(*d*) Shaping and planing.
(*e*) Drilling.
(*f*) Mechanics' bench work.
(*g*) Mill roller grooving and journal turning.
(*h*) Thread cutting on pipes and bolts.
(*j*) Forging different pieces.
(*k*) Pressing wheels on axles, bushings, etc.
(*l*) Possible casting of bronze and sometimes iron.

The casting of metals is generally limited to the filling of bearings with white metal. For the other operations, except electric welding, a few tools are cited below :—

1.—*Acetylene Cutting Torch.* For cutting heavy beams and plates. Good design of the acetylene (C_2H_2) generator is essential, as the gas mixture is endothermic and becomes dangerous when mixed with air. It is produced by slaking calcium carbide with water (1 lb. carbide requires 0·562 lbs. water) and produces 0·406 lbs. acetylene gas, occupying about 5·5 cub. ft. For one foot single weld there is required about 0·14 to 33 cub. ft. of acetylene according to the thickness of the material ($\frac{1}{16}$ in. to $1\frac{1}{4}$ in.) and respectively about 0·16 to 36 cub. ft. of oxygen. The torch temperature is about 3270°F. (1800°C.). Electric cutting by means of a graphite torch is done to a minor extent for light materials. Moreover, the oxy-acetylene torch is also required for soldering bronze. Acetylene can also be bought in steel flasks (cylinders).

2.—*Handworked Pipe Cutters* from $\frac{1}{8}$ in. to 3 in.

3.—*Shearing and Punching Machines.* Normal capacities required for plates up to $\frac{5}{8}$ in., flats 3 in. × $\frac{3}{4}$ in., rounds and squares 1 in., angles 4 in. × $\frac{3}{8}$ in., tees $2\frac{1}{2}$ in. × $\frac{1}{4}$ in. The combined type for all these tasks should be considered. The knife, about 8 in. long, makes a stroke of $\frac{7}{8}$ in. and 22 strokes/min. Driving shaft 300 r.p.m. Power required, 3 h.p. The machine body is generally made of cast steel or from riveted steel plate.

4.—*Shaping Machine.* Stroke 22 in. Driving 265 r.p.m. $1\frac{1}{2}$ h.p.

5.—*Planer.* 40 in. stroke, 8 to 40 strokes/min. 30 ft./min. cutting speed. Driving 370 r.p.m. 10 to 12 h.p.

6.—*Hack Saw.* For cutting material up to 6 in. square or round. 50 strokes/min. $\frac{1}{2}$ h.p. Horizontal band saws for sawing metal are now also on the market.

7.—*Column Drilling Machine.* 12 in. vertical travel. 14 to 336 r.p.m. Automatic feed from 0·006 in. to 0·015 in. Motor driven, 1150 r.p.m. 2 h.p.

8.—*Radial Drilling Machine.* Up to 2 in. holes and tapping. 25 in. travel. 38 to 356 r.p.m. Feed 0·007 in. to 0·02 in. Driving 380 r.p.m. 5 h.p.

9.—*Screwing Machines* with tangential or radial self-opening die-heads for thread cutting on pipes up to 6 in. and for bolts up to $1\frac{1}{4}$ in. Power about 3 h.p.

10.—*Lathes* for large and small work. 7·5 and 5 h.p.

11.—*Mill Roller Lathe.* 15 h.p. In many instances the mill rollers are turned in the mill head stocks with a special turning gear attached to the latter.

12.—*Hydraulic Ram Forcer* or wheel press for pressing car wheels, bushings, etc. Hand-operated.

13.—*A Complete Smithy* for the blacksmith and another for the coppersmith, with blower, anvils, hammers and tongs.

14.—*A Mechanics' Bench* with four parallel vices, 8 in. wide, 12 in. mouth opening, with files, scrapers, oil cans, measuring tapes, chisels, etc.

15.—A few sets of precision measuring instruments like verniers and micrometers. The latter with attachment bars for inside diameters up to 40 in. and outside diameters up to 20 in.

16.—A small sheet roller for rain gutters, etc., from 6 in. to 12 in. diam.

17.—A few pneumatic or electric drills, 175 to 500 r.p.m. Air drills 100 to 325 r.p.m. 90 lbs./sq. in. air pressure, 55 cub. ft. aspirated air per min. About 2 h.p.

18.—An electric or pneumatic grinder.

19.—A set of grinding stones.

20.—A set of tube expanders up to 6 in. for boiler, evaporator, heater and vacuum pan tubes, and for general pipe work.

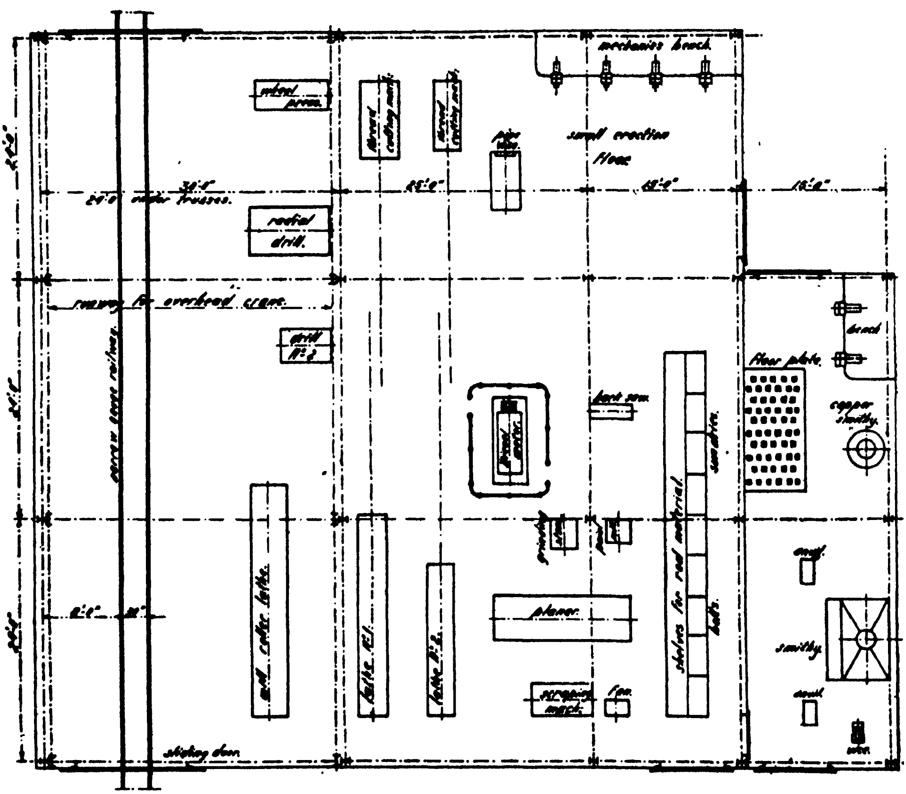


Fig. 605.—Layout of a Mechanics' Repair Shop.

In *Fig.* 605 is given the *Lay-out of a Mechanics' Repair Shop.* The heavy machinery is arranged in a bay with an overhead crane. Lathes and screwing machines should have sufficient room for long rods or tubes. The shelves or stands for rod material should allow for easy handling. The blacksmith and coppersmith shops should be in a separate building to avoid smoke. For good working conditions a spacious arrangement is to be preferred. The machines can be driven either by steam, Diesel or electric power, as may be required in each case.

3.—Welding.

Welding has become of such importance that each cane sugar factory should have a *portable welding outfit.* Oxy-acetylene welding and cutting is widely used, but electrical welding is of equal importance. The units are self-contained, and the generators are either driven by an A.C. motor, a D.C. motor and belt, or a gasoline engine. The *Generator Circuit* of a welding machine is shown in *Fig.* 606. On the same shaft of the generator G a separate exciter E is mounted, having field coils f. The current for the main generator field f_1 is supplied by this exciter and its intensity can be graduated by a rheostat R.

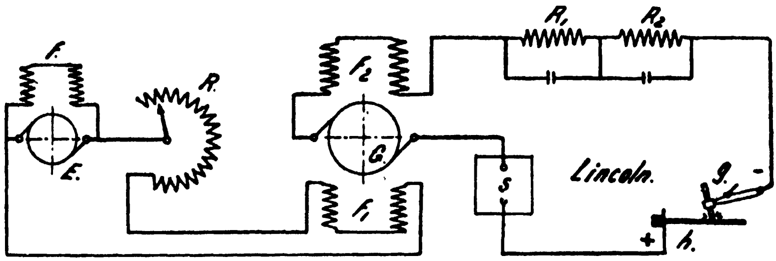


Fig. 606.—Welding Generator Circuit.

In series with the main current circuit is arranged the field f_2 which causes an opposed induction. Regulation now of the arc current is possible by regulation of the field current in f_1 by the resistance R of the separate exciter circuit, or by the resistances R_1 and R_2 of the main circuit in series. The electrode holder g is connected to the negative pole, whereas the work piece h is on the positive side. A stabilizer s is for stabilizing the arc current.

The scheme shown in *Fig.* 606 is for direct current welding, although A.C. welding is also done. The efficiency of the D.C. welding will be about 60 per cent., whereas A.C. welding has 80 per cent. efficiency. D.C. current welding requires only bare electrodes, whereas A.C. welding requires coated ones, which are considerably more expensive. But for overhead welding, coated electrodes are required, and in Great Britain they have found a wider application than, e.g., in the U.S.A. On the selection of a welding outfit, efficiency and expenses for electrodes should be compared, as well as the ease of operation.

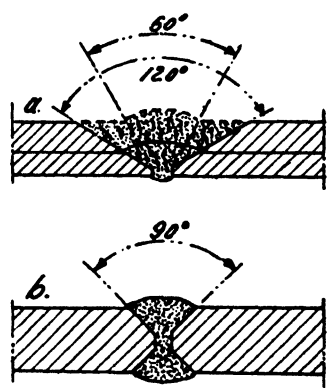


Fig. 607.
Cross Sections of Welds.

The power consumption is normally between 2 and 4 kw. or even less, and the arc current runs up, according to the welding conditions ruling, between 50 and 300 amp. The thickness of the electrodes, which are of very pure iron (about 99·85 per cent. pure), ranges between $\frac{1}{8}$ in. and $\frac{3}{8}$ in., the latter requiring the heavier current intensity.

Proper welding suits ana masks should be provided, as not only unprotected eyes, but also an unprotected skin will be affected, and masks and gloves must be worn while welding. Equal protection has to be given against dripping metal, which may cause wounds (cicatrix) difficult to heal.

The welding seam should be built up in rotary or zig-zag fashion. The cable connexion for electrically-operated welders should never exceed 50 ft., as long cables lying on the floor might be easily damaged by carts or falling pieces of material or equipment.

Electrical welding can be done for nearly all kinds of repair jobs, like burst tubes, tube welding for boilers and the factory pipe lines, cracks in boilers or tanks, structural work, filling worn shafts, etc. It is chiefly suited for light work, as heat stresses may result if proper annealing cannot be done afterwards, as with mill roller shafts and kindred heavy equipment. Not only can mild steel be welded, but cast steel as well, and even cast iron, when special electrode material is selected and a few pins driven in at the point of the weld. Broken teeth can be welded up, as the author has seen done several times with crown wheels and gears.

The welds are not overlapped as a rule, but preferably should be as shown in *Fig.* 607 for better stress distribution. A chamfering of 120° is suitable for thin plates, whereas for thicker material 60° will be better. The part *a* of the figure on top is for shear resistance, whereas *b* is for welds subject to lengthwise or tensile stresses.

CHAPTER XXXVII.

MISCELLANEA.

SUGAR REFINING — ALCOHOL DISTILLING — BOARD MANUFACTURE.

Those industries which make from the raw sugar the final product for the consumer, as well as those which use the by-products of the sugar manufacturing process as their raw material, are briefly described under the following sub-headings.

1.—Sugar Refining.

Direct consumption sugar is not always produced right at the cane sugar factory; more frequently it is raw sugar, which has to be refined in huge factories in the countries where the sugar is consumed. The consumption sugar made at the plantation is generally a "plantation white sugar" or "granulated," but market exigencies in many countries are such that only refined sugar of pure white colour and even grain can be sold.

Economically the question arises whether it would not be more efficient to produce consumption sugar, also refined, direct at the plantation. Although it must be fully admitted that our present-day refineries, situated at the mainland water-side for easy shipment of raw material and the finished product, have attained a very high degree of efficiency and a very low refining cost, there is nevertheless a growing tendency to produce refined sugar at the plantation and in each case investigation should be made whether it is a profitable proposition.

Moreover, existing tendencies have to be considered, and cane sugar with its natural flavour is not usually sold as such, but is bought at par with beet sugar, which has to be refined to become a consumable product. The question how this state of affairs came into being does not, however, come within the scope of this book.

A more technical consideration is whether white sugar should be produced by first making raw sugar and then remelting, or whether a single process for making refined sugar direct from the cane should be preferred. The remelt refining process has the great advantage that the bulk of the melassigenic impurities has been removed in the raw sugar centrifugals, which makes clarification or decolorizing much more practicable.

In beet sugar factories a "direct consumption" sugar is made directly from the beet juices by double carbonatation, and although this sugar is sold in some countries a trifle below the price for refined sugar, the quality of the product can be considered equal. To the Dutch sugar technologists in Java recognition should be given for having introduced this double carbonatation process for cane sugar juices and with very gratifying results; the difference between refined sugar and this "plantation white" has become in practice almost unnoticeable.

Refining (when not otherwise mentioned, remelt refining is meant here) is accomplished by filtering the remelt liquors in packed filters over animal charcoal, as is done in most large refineries. WYNBERG in Java invented in

1911 a vegetable carbon, the product of dry distillation of vegetable matter, which has a greatly increased decolorizing capacity, and is of so fine a structure that it can be mixed with juices or liquors and filtered in filter-presses or leaf-filters. This carbon refining is done nowadays at many cane sugar factories and some data as to its use are given below.

For an economic refining process it is of the utmost value to have the "raw" sugar of as high a polarization as possible, which requires less carbon, and have the remelt of a high density up to 65° Brix to reduce the fuel consumption. The carbon is mixed with the remelt at about 195°F. (90°C.) and pre-coating of the filter-cloth is required. The exhausted carbon can be treated with acid and alkali, as it is composed of nearly pure carbon, which is not the case with animal char. Moreover, it can be revivified or regenerated in special kilns at the producing temperature. The chemical treatment serves to eliminate inorganic matter, whereas re-burning under exclusion of air will carbonize organic matter at the temperature mentioned above.

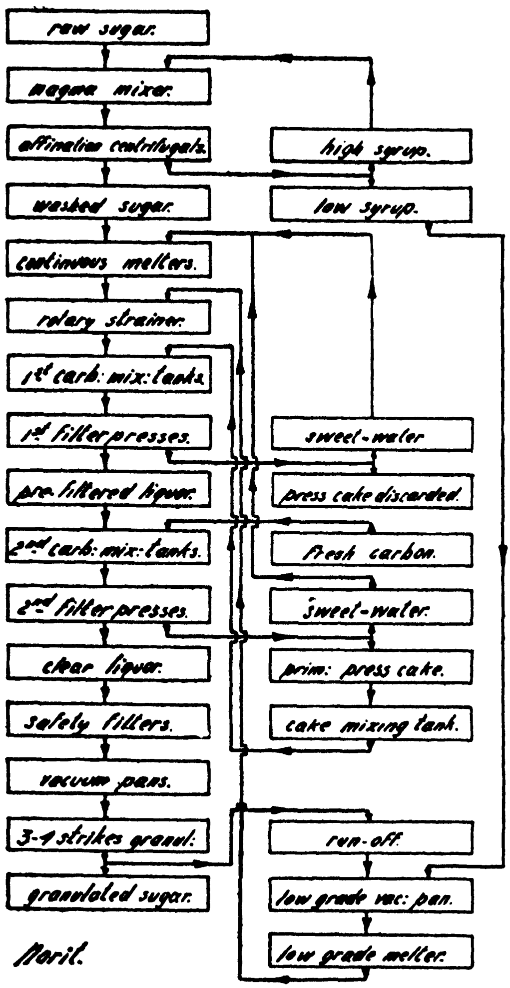


Fig. 608.—Flow Sheet of a Refining Process without Revification.

In *Fig.* 608 a *Flow Sheet of a Refining Process without Revivification* is shown. The raw sugar is mixed to a magma with a liquor of high purity and the affined sugar separated in centrifugals. The purity or polarization of the affined sugar is thus increased, as the film of low purity molasses, surrounding the crystals, has been washed partially away. The low syrup from the centrifugals is boiled into low grades, whereas the high syrup or wash can be used for mixing with the raw sugar. The affined sugar is melted in a continuous melter or a set of two intermittent stirrers; the remelt liquor is strained in a mechanical rotary strainer for removing coarse impurities, like bag fibre, etc. For removing a part of the impurities, a pre-filtration with already used carbon and diatomaceous filter-aids is carried out. The filter-presses have transverse washing and the clear run-off is again mixed, but now with fresh carbon for the second filtration.

After the latter a safety filter of the bag type is normally provided to remove the last traces of carbon or as an emergency filter, lest a cloth in the filter-presses should have ruptured.

The sweet-water is used for the remelt, and the boiling of the water-clear liquors is achieved in three or more strikes, as the purity drop necessarily has to be low (see Chapter XXII).

It should be recollected that cane sugar syrup contains such a quantity of impurities that it cannot be filtered without consuming very large amounts of carbon, so that revivification may even become impracticable. Special treatment is therefore essential. The juices or syrups of carbonatation factories can be filtered with carbon and the low grades of these factories can be efficiently affined and remelted before filtration over vegetable carbon. These clear liquors then can be used as *pied-de-cuite* for the first boilings.

Revivification becomes of importance for factories turning out above 200 to 250 tons refined per day, and in those countries with high freight rates and high import duties on carbon.

Liquors containing carbon have an abrasive effect on pumps, etc., and open impeller centrifugal pumps (phosphor-bronze) or special cast iron ones ("Duriron," etc.) are used, the packing glands being water-sealed to prevent the char wearing the shaft. Triplex plunger pumps are also used for the filter-press performance and automatic montejus of about 900 gals. capacity (the mixing tanks have the same capacity) have given very reliable operation.

Different data about vegetable carbon refining are given below :—

Filtration rate : about 3 to 6 Imp. gals./sq. ft. F.S./hr. (150-300 l./sq. m./hr.).

Sweet-water : about 10 times weight of cake.

Carbon in process (revivification) : up to 3 per cent. on solids in solution (revivification loss : 1·5–5 per cent. on carbon).

Carbon in process (without revivification) : from 0·35 to 0·75 per cent. on solids in solution.

Decolorizing power compared with bonechar 30 : 1.

Revivification of 1 lb. veg. carbon requires about 3200 B.Th.U. at 1000°F.

Lbs. sugar per lb. carbon revivified : 26 (approx.).

Lbs. sugar per lb. fresh carbon added : 240 (approx.).

K.W.H. required per lb. carbon for electrical revivification : 0·26.

Acid required per lb. carbon to be regenerated : 0·06 lb. (1 Imp. gal. of 30 per cent. strength per 40 lbs. carbon).

Lbs. carbon required for pre-coating : 0·4 lbs./sq. ft. F.S.

Pre-coating filtration rate : max. 2·2 Imp. gals./sq. ft. F.S./hr.

2.—Alcohol Distilling.

In some instances, for certain high classes of liquors or beverages, like rum, cane juice is fermented, but in general it may be said that industrial and consumption alcohol is produced by the fermentation of the final or black-strap molasses. Molasses are sold as raw material to large distilleries abroad, or are distilled at the plantation. Economic conditions govern the selection of what is best to do.

Fermentation is caused by a yeast, and the molasses is generally diluted to 15–20° Brix at a degree of acidity of about 5 *pH* for optimum conditions of fermentation. For good performance the yeast is cultivated at the distillery. Hydro-carbons, like sucrose $C_{12}H_{22}O_{11}$, glucose $C_6H_{12}O_6$ and cellulose (starch)

$(C_6H_{10}O_5)_n$, are converted into ethyl and methyl alcohol, C_2H_5OH and CH_3OH, with release of CO_2. This CO_2 generally is not recovered, but can serve for making so-called "dry ice."

The distilling process is complicated and the design of the apparatus has been guided to a great extent by empirical data[1]. In dehydrating the alcohol, i.e., producing a water-free product, such as is now required for motor fuel, the azeotropic process is most generally used.[2]

A *Diagrammatic Arrangement of an Alcohol Distillery* is to be seen in *Fig.* 609. The final molasses from the sugar factory is pumped into a mixing tank *a* and there mixed with acid and diluted with water to a density from 10° to 30° Brix. The diluted molasses is now mixed with pure yeast from the cultivating tanks *y* in the pre-fermentation tank *b*, from where it is lowered into the fermenting tubs *T*, which are generally made of wood; but nowadays cylindrical

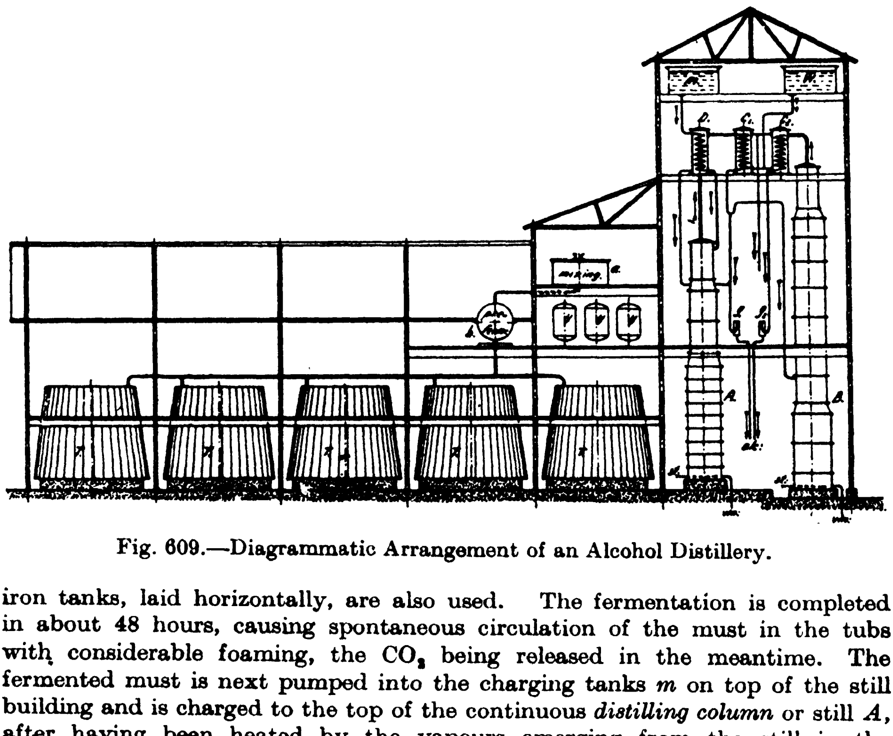


Fig. 609.—Diagrammatic Arrangement of an Alcohol Distillery.

iron tanks, laid horizontally, are also used. The fermentation is completed in about 48 hours, causing spontaneous circulation of the must in the tubs with considerable foaming, the CO_2 being released in the meantime. The fermented must is next pumped into the charging tanks *m* on top of the still building and is charged to the top of the continuous *distilling column* or still *A*, after having been heated by the vapours emerging from the still in the condenser *D* (also called dephlegmator or cooler). The still is heated by a steam coil at the bottom and as the vapours are deemed to be mixtures of water and alcohol vapour, emerging from liquids having different boiling points, it will be obvious that by passing these vapours through liquid-sealed openings in the still bottoms, the heavier vapours will be first condensed and the lighter ones, which contain more alcohol, will rise to the top of the still. The pressure in the columns is about 2 to 3 lbs./sq. in. and the still should be protected by heat-insulating material (this unfortunately is not the general practice). The condensate from the first condenser *D* is returned to the still and a second condenser or cooler C_1 causes further condensation, by means of cooling water

[1] For a complete study, see, E. HAUSBRAND, Die Wirkungsweise der Rektifizier-und-Destillier-Apparate
[2] *Int. Sugar Jl.*, 1933, pp. 29 and 71; 1934, p. 24.

supplied by the tank *w*. The alcohol of lower degree (generally from 40 to 50° Gay-Lussac), when required, is drawn off at the gauge g_1. In case a higher degree alcohol is required, containing less water, the vapours are led to a second still or *rectifier B*, having also a consecutive number of intermediate liquid-sealed bottoms. The final alcohol vapours are cooled in the condenser C_2 and drawn off at the gauge g_2.

The remaining slop, or vinasse, which is exhausted, accumulates at the bottom of the column and the rectifier and is continuously released to the gutter. Some aromatic oils, like fusel oil and the like, are also withdrawn in some instances from the columns.

A few data are given below for alcohol distillation.

Steam required for one gallon alcohol of 99·8° G.L. : about 15 lbs. under optimum conditions (the degree Gay-Lussac indicates the per cent. alcohol).

Ditto, 65 to 75° G.L. from must containing 6 per cent. alcohol : about 28 lbs.

Boiling points :—

Per cent. alcohol by volume in liquid	Per cent alcohol by weight in liquid	Boiling point °C.	Per cent. of Alcohol Vapours Dönitz.	Sorel.
12	7·9	91·5	53·0	55·8
36	29·9	84·7	74·6	70·9
48	40·7	84·6	79·0	74·4
96	94·0	78·7	—	96·2
100	—	78·4	—	—

Specific weights :		
	50 per cent. alcohol	0·93430
	90 ,, ,,	0·83339
	100 ,, ,,	0·79460

Air required for combustion of 1 lb. alcohol : 9·08 lbs.
Temperature of spontaneous combustion of alcohol : 395°C.
Elementary components of alcohol :
52 per cent. carbon, 13 per cent. hydrogen, 35 per cent. oxygen.
Yield in 100 per cent. alcohol from 100 lbs. molasses of 55 per cent. total sugars : 3·2 to 3·5 Imp. gals.
100 lbs. sugar produce as by-product in molasses : 1·4 to 2 gals. or more.
The percentage alcohol according to Gay-Lussac is measured at 15°C.
1° Gay-Lussac = 2° U.S. proof.

3.—Board Manufacture.

Bagasse has become a valuable raw material since synthetic boarding has replaced wooden boards for making partitions in living houses, for ceilings and for other purposes in the building industry. The standard sizes for wall board are 32 in. and 48 in. width, having a length from 6 to 14 ft.

Wallboards are made from vegetable fibres or wood pulp and the bagasse from cane has exceptional qualities for this kind of material, as may be gleaned from the reception bagasse boards (like Celotex in Louisiana, Canec in Hawaii and Vazcane in Cuba) have had in the building market. The boards are gently pressed for interior decoration and wall covering, as well as for insulating, whereas high pressure is applied in those instances where great compactness is required, as for furniture manufacturing, etc.

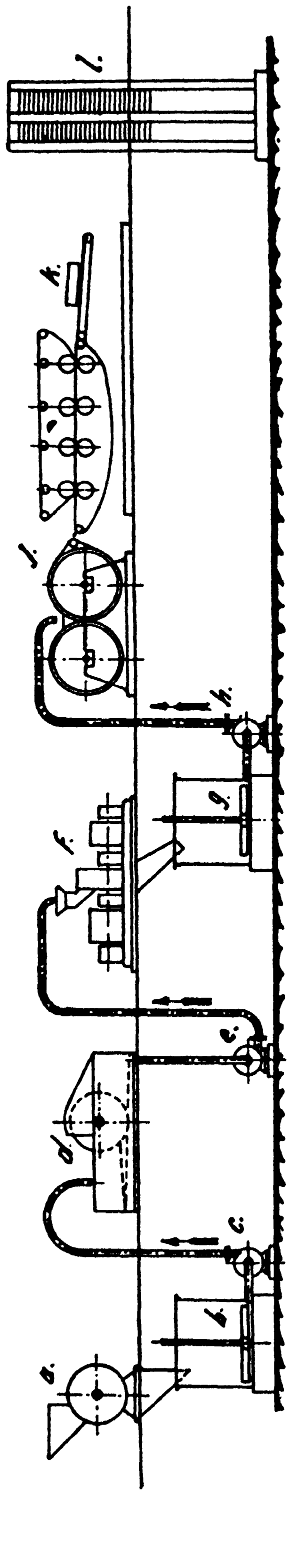


Fig. 610.—A General Board Manufacturing Scheme.

The bagasse fibres are impregnated with chemical solutions, so that the boards produced therefrom will not be attacked by insects and moreover will be rendered fireproof.

A *General Board Manufacturing Scheme* is shown in *Fig.* 610, and it should be mentioned here that two processes are applied, the one being mechanical throughout as shown in the figure, and the other partly chemical action by cooking the bagasse fibres in large globular cookers or digesters with lime and other chemicals, as is done in the straw-board industry. The purpose of the cooking is the disintegration and the preparation of the fibre. Sugar should be destroyed or eliminated when present in the bagasse.

The bales of bagasse are broken up by a bale breaker *a*, the unbaled bagasse being delivered into a stirring tank *b*, where it is mixed with water and pumped by an unchokeable pump *c* into the beater or breaker *d* for first disintegration. The pump *e* delivers the pulp to the rotary refiner *f*, from which it is discharged into the agitating tank *g*, where some binding material, like resin, is added. The pulp is sometimes passed under a magnet for removal of any pieces of tramp iron and is then pumped by pump *h* towards the board-making machine *j*, built on the principle of the drum type vacuum filter, as explained in Chapter XX.

The wet sheets are taken off the rolls and led between several sets of drying rollers, so they have already a certain consistency when emerging from the machine. At *k* a trimming and cutting machine is arranged for continuous operation of the board-forming machine. The sheets are next transferred to a dryer or a steam-heated platen press *l*. The final cutting and trimming is done after the boards have been dried.

The machinery comprises standard equipment, as used in the board industry for vegetable fibres.

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BUYERS' GUIDE.

This List must not be taken as exhaustive or conclusive. It should be noted that Items marked with a star can usually be obtained from any of the firms listed under "Sugar Machinery, General."

A

Agitators (*See* **Stirrers**).

Agricultural Machinery and Implements.

RANSOMES, SIMS & JEFFERIES, LTD.,	Ipswich, England.

Air Heaters.

DAVIDSON & CO., LTD.,	Sirocco Engineering Works, Belfast.
THOMPSON WATER TUBE BOILERS, LTD.,	Wolverhampton, England.

Asbestos Cement Roofing.

JOHNS-MANVILLE INTERNATIONAL CORPORATION	22, East 40th Street, New York, U.S.A.

B

Bagasse Briquetting Presses.

BUCKAU R. WOLF A.-G.,	Magdeburg, Germany.

Bagasse Furnaces.

STEWART & CO., LTD., DUNCAN,	London Road Iron Works, Glasgow, S.E.
THOMPSON WATER TUBE BOILERS, LTD.,	Wolverhampton, England.

Bagging Machines for Sugar.

RICHARDSON SCALE CO.,	Clifton, New Jersey, U.S.A.
SACK FILLING & SEWING MACHINE SYNDICATE, LTD.,	Kenmure Yard, Kenmure Road, Hackney, London, E.8.

Boilers, Water Tube.

ARMSTRONG WHITWORTH & CO. (ENGINEERS) LTD.,	Scotswood Works, Newcastle-upon-Tyne, England.
THOMPSON WATER TUBE BOILERS, LTD.,	Wolverhampton, England.

Boiler Feed Water Heaters.

ROYLES, LTD.,	Irlam, near Manchester.

Boiler Feed Water Pumps.*

REINEVELD N.V., MACHINEFABRIK	Delft, Holland.
WORTHINGTON-SIMPSON, LTD.,	Newark-on-Trent, England.

Boiler Feed Water Regulators.

TRIST & CO., LTD., RONALD,	1-3, Brixton Road, London, S.W.9.

Brake Linings.

JOHNS-MANVILLE INTERNATIONAL CORPORATION	22, East 40th Street, New York, U.S.A.

Buckets (for Elevators).

EWART CHAINBELT CO., LTD.,	Derby, England.

C

Cableways.
HENDERSON & CO., LTD., JOHN M., King's Works, Aberdeen.

Calorifiers.
ROYLES, LTD., Irlam, near Manchester.

Cane Knives (for Harvesting).
SUGAR MANUFACTURERS' SUPPLY CO., LTD., 7-8, Idol Lane, London, E.C.3.

Capstans, Electric (for Cane Car Haulage).
BROADBENT & SONS, LTD., THOS., Huddersfield, Yorkshire, England.
HENDERSON & CO., LTD., JOHN M., King's Works, Aberdeen.

Carbon, Activated and Decolorizing.
NORIT CO., AMERICAN, 551, Fifth Avenue, New York.
NORIT-VEREENIGING, N.V., Verkoop Centrale, Amsterdam-C., Holland.
SUCHAR PROCESS CORPORATION, 120, Wall Street, New York City.
SUGAR MANUFACTURERS' SUPPLY CO., LTD., 7-8, Idol Lane, London, E.C.3.

Carriers, Cane and Intermediate.*
FLETCHER & CO., LTD., GEORGE, Masson Works, Derby.
FULTON IRON WORKS CO., St. Louis, Mo., U.S.A.
SMITH & CO., LTD., A. & W., Eglinton Engine Works, Glasgow, C.5.
SQUIER MANUFACTURING CO., GEO. L., Buffalo, New York, U.S.A.

Carrier Chains.
EWART CHAINBELT CO., LTD., Derby, England.
SMITH & CO., LTD., A. & W., Eglinton Engine Works, Glasgow, C.5.

Centrifugal Machines and Accessories.
BROADBENT & SONS, LTD., THOS., Huddersfield, Yorkshire, England.
BUCKAU R. WOLF A.-G., Magdeburg, Germany.
FLETCHER & CO., LTD., GEORGE, Masson Works, Derby.
POTT, CASSELS & WILLIAMSON, Motherwell, near Glasgow.
REINEVELD N.V., MACHINEFABRIK Delft, Holland.
SANGERHAUSEN, MASCHINENFABRIK, Sangerhausen, Germany.
SQUIER MANUFACTURING CO., GEO. L., Buffalo, New York, U.S.A.
WATSON, LAIDLAW & CO., LTD., 98, Laidlaw Street, Glasgow, C.5.

Centrifugal Motors, Electric.
HEEMAF N.V., Hengelo, Holland.

Chains (Conveyor & Driving).
EWART CHAINBELT CO., LTD., Derby, England.

Chemicals (General).
SUGAR MANUFACTURERS' SUPPLY CO., LTD., 7-8, Idol Lane, London, E.C.3.

Chemicals (for Water Softening).
KENNICOTT WATER SOFTENER CO., LTD., Wolverhampton, England.

Chimneys, Steel.*
BRABY & CO., LTD., FREDK., Ida Works, Deptford, London, S.E.8.
HARVEY & CO. (LONDON), LTD., G. A., Greenwich Metal Works, London, S.E.7.

Clarification, Compound.
PETREE & DORR ENGINEERS, INC., 67, Wall Street, New York.

Clarification Plant.*
FLETCHER & CO., LTD., GEORGE, Masson Works, Derby.

Clarifiers, Continuous.
PETREE & DORR, ENGINEERS, INC., 67, Wall Street, New York.

Clarifiers (Lime, Phosphoric Acid, etc.).	
SUGAR MANUFACTURERS' SUPPLY CO., LTD.,	7-8, Idol Lane, London, E.C.
Clutches.	
BROADBENT & SONS, LTD., THOS.,	Huddersfield, Yorkshire, England.
Coils, Vacuum Pan Heating.	
YORKSHIRE COPPER WORKS, LTD.,	Leeds, England.
Compressors, Air and CO_2.*	
INGERSOLL-RAND CO., LTD.,	165, Queen Victoria Street, London, E.C.4.
WORTHINGTON-SIMPSON, LTD.,	Newark-on-Trent, England.
Condensers, Ejector.	
KORTING BROS. (1917) LTD.,	Sheepbridge Lane, Mansfield, England.
PETREE & DORR ENGINEERS, INC.,	67, Wall Street, New York.
Condensing Plant, Barometric.*	
FLETCHER & CO., LTD., GEORGE,	Masson Works, Derby.
INGERSOLL-RAND CO., LTD.,	165, Queen Victoria Street, London, E.C.4.
MIRRLEES WATSON CO., LTD.,	Scotland Street, Glasgow.
SMITH & CO., LTD., A. & W.,	Eglinton Engine Works, Glasgow, C.5.
STEWART & CO., LTD., DUNCAN,	London Road Ironworks, Glasgow, S.E.
Conveying Plant.	
DAVIDSON & CO., LTD.,	Sirocco Engineering Works, Belfast.
Conveyors, Sugar.	
BROADBENT & SONS, LTD., THOS.,	Huddersfield, Yorkshire, England.
EWART CHAINBELT CO., LTD.,	Derby, England.
POTT, CASSELS & WILLIAMSON,	Motherwell, near Glasgow.
RICHARDSON SCALE CO.,	Clifton, New Jersey, U.S.A.
SQUIER MANUFACTURING CO., GEO. L.,	Buffalo, New York, U.S.A.
WATSON, LAIDLAW & CO., LTD.,	98, Laidlaw Street, Glasgow, C.5.
Cooling Plants, Spray.	
KORTING BROS. (1917) LTD.,	Sheepbridge Lane, Mansfield, England.
Copper Work and Tubes.	
YORKSHIRE COPPER WORKS LTD.,	Leeds, England.
Cranes, Cane Storage.	
HENDERSON & CO., LTD., JOHN M.,	King's Works, Aberdeen.
Crushers, Splitting.*	
MCNEIL & CO., LTD., JOHN,	Colonial Iron Works, Govan, Glasgow.
SMITH & CO., LTD., A. & W.,	Eglinton Engine Works, Glasgow, C.5.
SQUIER MANUFACTURING CO., GEO. L.,	Buffalo, New York, U.S.A.
Crystallizers.*	
BUCKAU R. WOLF A.-G.,	Magdeburg, Germany.
FLETCHER & CO., LTD., GEORGE,	Masson Works, Derby.
POTT, CASSELS & WILLIAMSON,	Motherwell, near Glasgow.
SQUIER MANUFACTURING CO., GEO. L.,	Buffalo, New York, U.S.A.
WATSON, LAIDLAW & CO., LTD.,	98, Laidlaw Street, Glasgow, C.5.
Crystallizers (Counterflow Disc Type).	
SMITH & CO., LTD., A. & W.,	Eglinton Engine Works, Glasgow, C.5.
Crystallizers (Rotary).	
PETREE & DORR ENGINEERS, INC.,	67, Wall Street, New York.
Cultivating Machinery.	
RANSOMES, SIMS & JEFFERIES, LTD.,	Ipswich, England.

D

Decolorizing Carbons.

NORIT CO., AMERICAN,	551, Fifth Avenue, New York.
NORIT-VEREENIGING, N.V.,	Verkoop Centrale, Amsterdam-C., Holland.
SUCHAR PROCESS CORPORATION	120, Wall Street, New York City.
SUGAR MANUFACTURERS' SUPPLY CO., LTD.,	7-8, Idol Lane, London, E.C.3.

Derricks, Field.

HENDERSON & CO., LTD., JOHN M.,	King's Works, Aberdeen.

Diatomaceous Earth (Kieselguhr).

DICALITE COMPANY,	120, Wall Street, New York, U.S.A.
JOHNS-MANVILLE INTERNATIONAL CORPORATION	22, East 40th Street, New York, U.S.A.
SUGAR MANUFACTURERS' SUPPLY CO., LTD.,	7-8, Idol Lane, London, E.C.3.

Dryers (Vertical Tray Type).

JENKINS & CO., LTD., ROBERT,	Ivanhoe Works, Rotherham, England.

Drying Plant for Sugar.*

DAVIDSON & CO., LTD.,	Sirocco Engineering Works, Belfast.
FLETCHER & CO., LTD., GEORGE,	Masson Works, Derby.
JENKINS & CO., LTD., ROBERT,	Ivanhoe Works, Rotherham, England.
MIRRLEES WATSON CO., LTD.,	Scotland Street, Glasgow.
POTT, CASSELS & WILLIAMSON,	Motherwell, near Glasgow.
SQUIER MANUFACTURING CO., GEO. L.,	Buffalo, New York, U.S.A.
STORK & CO., GEBR.,	Hengelo, Holland.

Dust Collectors.

DAVIDSON & CO., LTD.,	Sirocco Engineering Works, Belfast.

E

Electric Equipment for Sugar Factories.

HEEMAF N.V.,	Hengelo, Holland.
U.C.M.A.S.	18, Chaussée de Charleroi, Brussels.

Electric Generators (A.C. and D.C.)

HEEMAF N.V.,	Hengelo, Holland.

Electric Locomotives.

ARMSTRONG WHITWORTH & CO. (ENGINEERS) LTD.,	Scotswood Works, Newcastle-upon-Tyne, England.

Electric Motors, General.

HEEMAF N.V.,	Hengelo, Holland.

Elevators, Sugar.*

BROADBENT & SONS, LTD., THOS.,	Huddersfield, Yorkshire, England.
EWART CHAINBELT CO., LTD.,	Derby, England.
POTT, CASSELS & WILLIAMSON,	Motherwell, near Glasgow.
SQUIER MANUFACTURING CO., GEO. L.,	Buffalo, New York, U.S.A.
WATSON, LAIDLAW & CO., LTD.,	98, Laidlaw Street, Glasgow, C.5.

Engines, Diesel.

ARMSTRONG WHITWORTH & CO. (ENGINEERS) LTD.,	Scotswood Works, Newcastle-upon-Tyne, England.
MIRRLEES WATSON CO., LTD.,	Scotland Street, Glasgow.
STORK & CO., GEBR.,	Hengelo, Holland.

Engines, Steam Mill.*	
FLETCHER & CO., LTD., GEORGE,	Masson Works, Derby.
FULTON IRON WORKS CO.,	St. Louis, Mo., U.S.A.
SANGERHAUSEN, MASCHINENFABRIK.	Sangerhausen, Germany.
SMITH & CO., LTD., A. & W.,	Eglinton Engine Works, Glasgow, C.5.
U.C.M.A.S.,	18, Chaussée de Charleroi, Brussels.
Engines, Vacuum Pumping.*	
FAWCETT, PRESTON & CO., LTD.,	Bromborough, Cheshire, England.
SMITH & CO., LTD., A. & W.,	Eglinton Engine Works, Glasgow, C.5.
Evaporators.*	
AITKEN CO., LTD., H. W.,	Caledonia Engineering Works, Paisley, Scotland.
BLAIRS, LTD.	Glasgow Engineering Works, Glasgow, S.W.1.
BUCKAU R. WOLF A.-G.,	Magdeburg, Germany.
FAWCETT, PRESTON & CO., LTD.,	Bromborough, Cheshire, England.
FLETCHER & CO., LTD., GEORGE,	Masson Works, Derby.
MCNEIL & CO., LTD., JOHN,	Colonial Iron Works, Govan, Glasgow.
MIRRLEES WATSON CO., LTD.,	Scotland Street, Glasgow.
SANGERHAUSEN, MASCHINENFABRIK,	Sangerhausen, Germany.
SKODA WORKS LTD.,	Praha, Czechoslovakia.
SMITH & CO., LTD., A. & W.,	Eglinton Engine Works, Glasgow, C.5.
SQUIER MANUFACTURING CO., GEO. L.,	Buffalo, New York, U.S.A.
STEWART & CO., LTD., DUNCAN,	London Road Ironworks, Glasgow, S.E.
STORK & CO., GEBR.,	Hengelo, Holland.
U.C.M.A.S.,	18, Chaussée de Charleroi, Brussels.
Evaporator Tubes.	
YORKSHIRE COPPER WORKS, LTD.,	Leeds, England.
Evaporator Tube Cleaners.	
SUGAR MANUFACTURERS' SUPPLY CO., LTD.,	7-8, Idol Lane, London, E.C.3.

F

Fans (Forced or Induced Draught).	
DAVIDSON & CO., LTD.,	Sirocco Engineering Works, Belfast.
Feed-Water Heaters.	
ROYLES, LTD.,	Irlam, near Manchester.
Filters (Auto).	
SMITH & CO., LTD., A. & W.,	Eglinton Engine Works, Glasgow, C.5.
SUCHAR PROCESS CORPORATION,	120, Wall Street, New York City.
Filters (Drum Type String Discharge).	
FILTRATION ENGINEERS, INC.,	858, Summer Avenue, Newark, New Jersey, U.S.A.
Filters (Drum Type Vacuum).	
FILTRATION ENGINEERS, INC.,	858, Summer Avenue, Newark, New Jersey, U.S.A.
OLIVER-UNITED FILTERS, INC.,	33, West Forty-Second Street, New York.
Filters (General Juice).	
JOHNSON & CO., LTD., S. H.,	Stratford, London, E. 15.
MIRRLEES WATSON CO., LTD.,	Scotland Street, Glasgow.
OLIVER-UNITED FILTERS, INC.,	33, West Forty-Second Street, New York.
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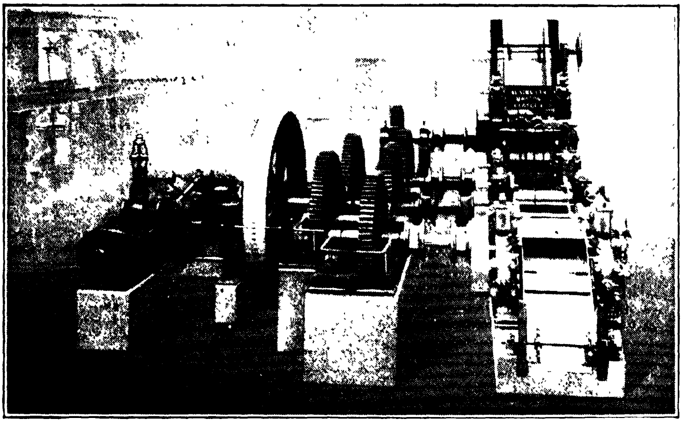


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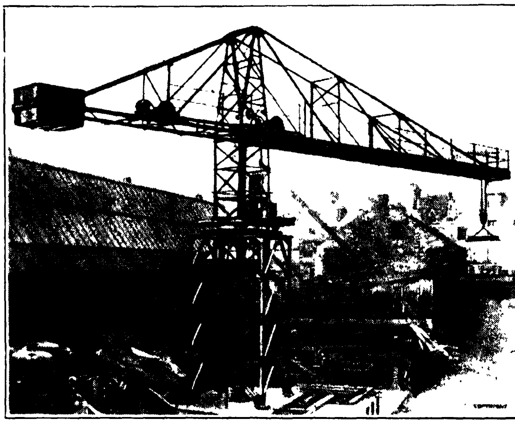

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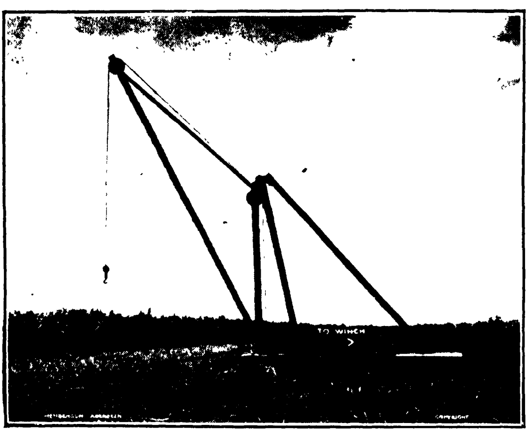

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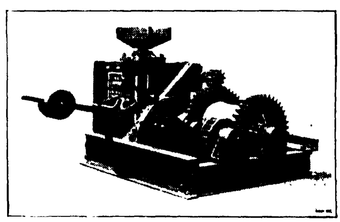

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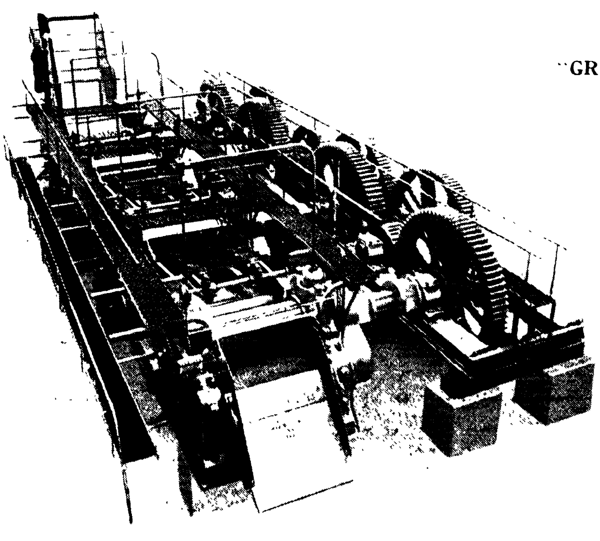

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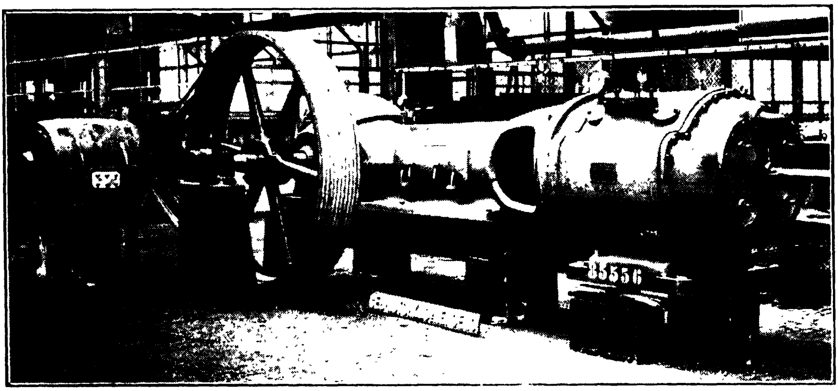

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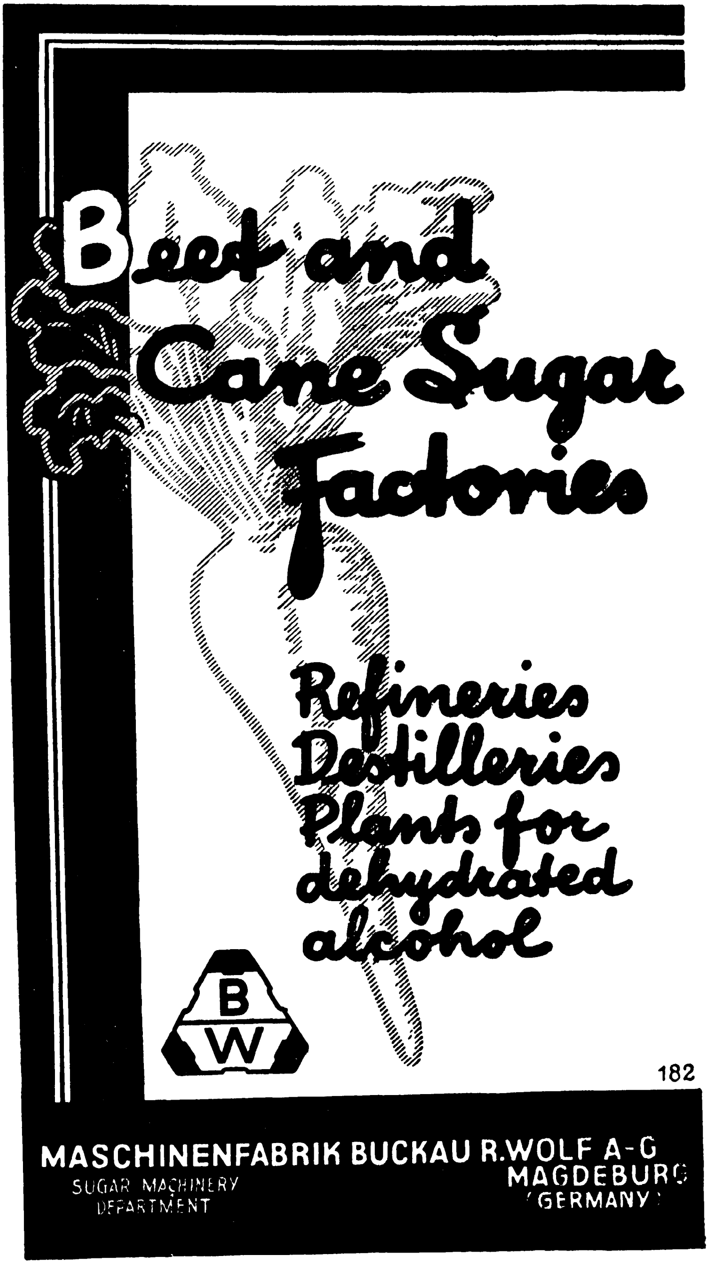
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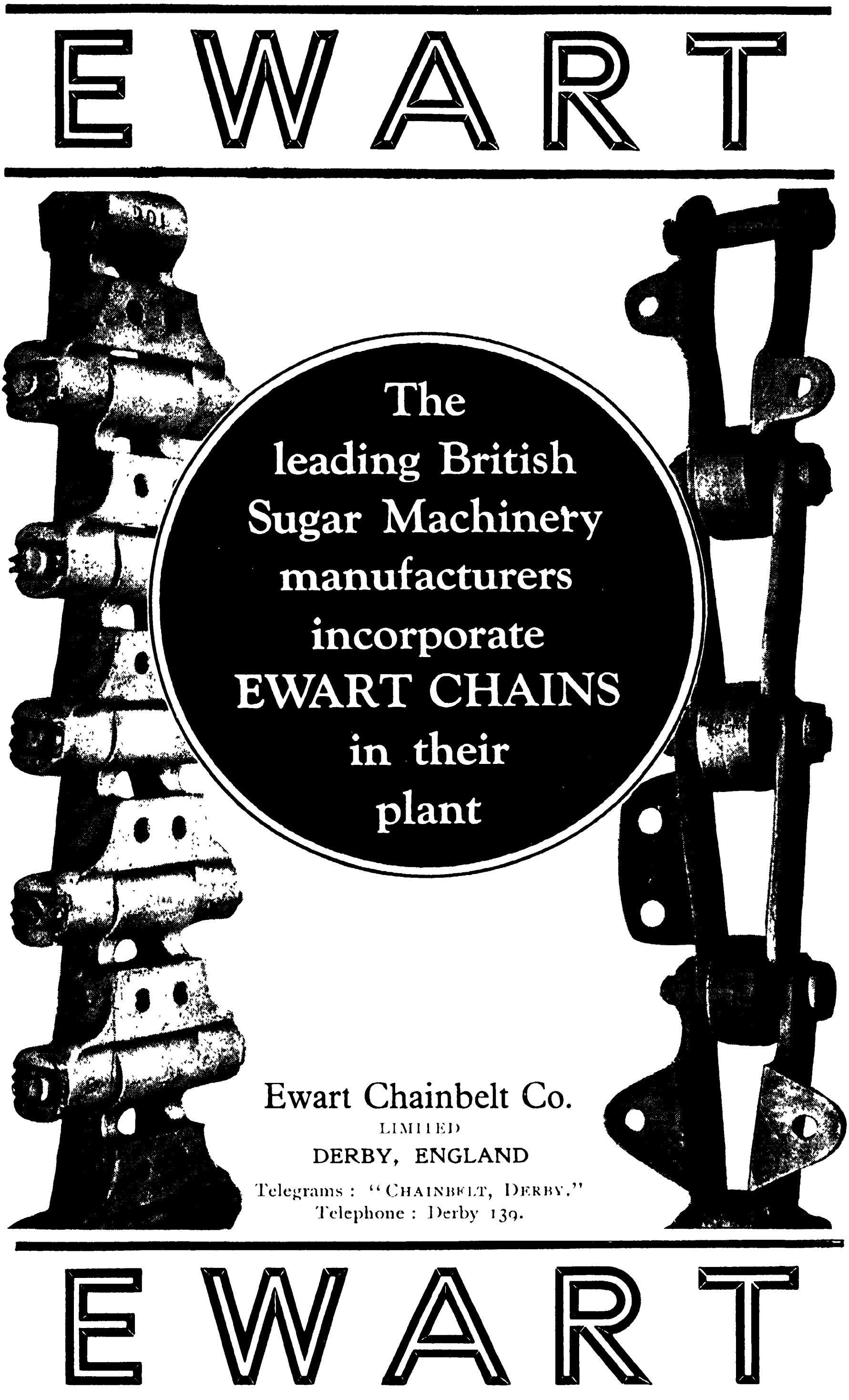
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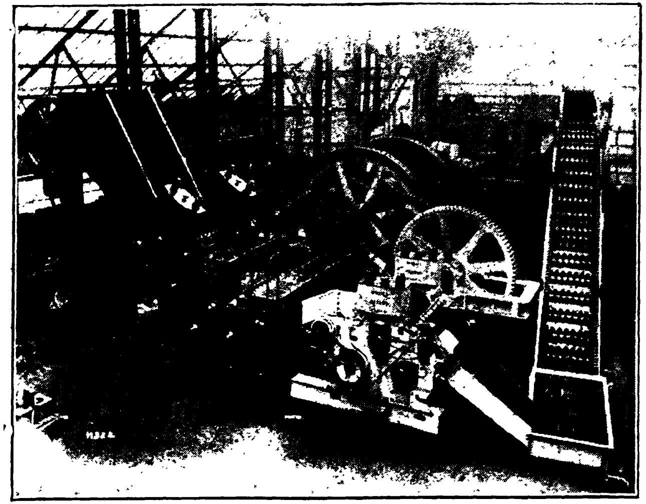

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