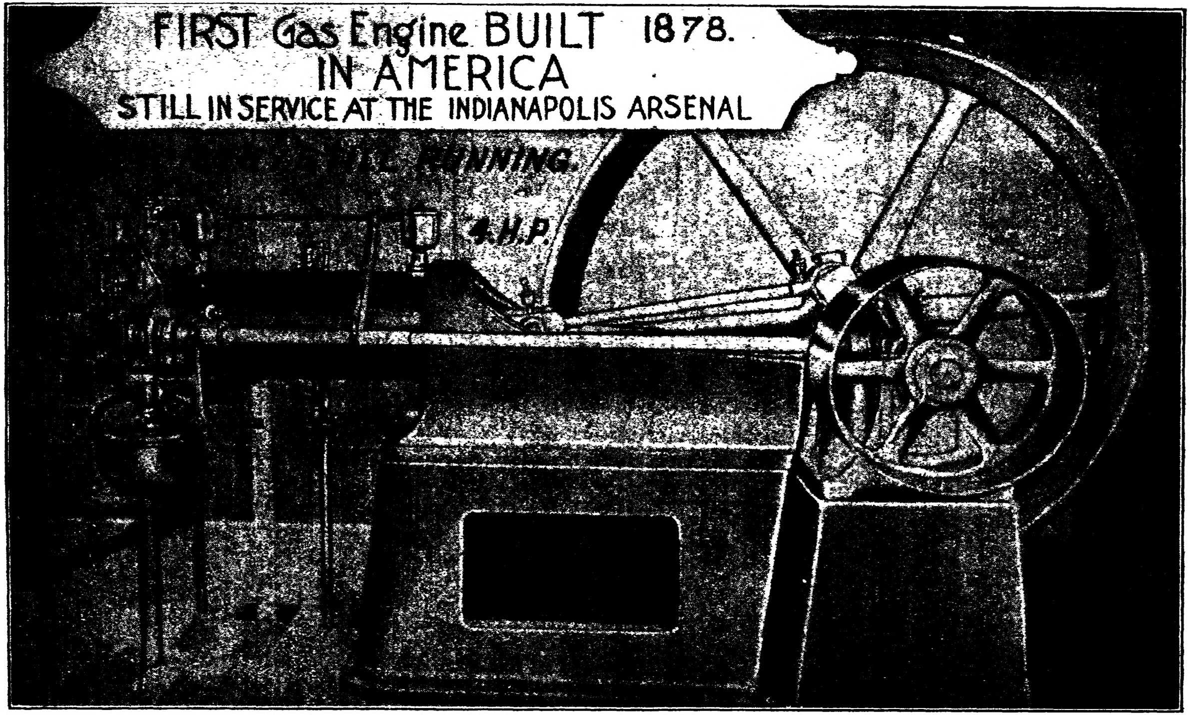
HEAT POWER



Frontispiece.—The first gas engine built in America. Built by the Otto Engine Works of Philadelphia. This engine used a slide valve on the cylinder head for inlet and exhaust of the gases, the valve being operated by the lay shaft. The governor and lubricators were also operated by the lay shaft. Ignition was by means of an open flame which was re-lighted by a second flame after each explosion. (See p. 6) (*Courtesy Otto Engine Works.*)

HEAT POWER

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PREFACE TO SECOND EDITION

It is said that the most up-to-date large steam turbine is from two to three years behind the most advanced engineering in this field by the time it is placed in service. An elementary textbook on the subject of heat power must, in the nature of things, lag behind the most modern equipment, but at the same time should be revised frequently enough to discard those items which do not represent standard practice and to take up the pertinent new developments.

The revised edition of this book has abandoned many illustrations of equipment of fading importance and has added new ones in about equal number so that the size of the book is not materially increased.

The following changes have been made in subject matter: (1) The discussion of boiler types has been much simplified; (2) the most recent available data on specific heats of gases and on properties of steam have been used, the latter through the courtesy of John Wiley & Sons, Inc.; (3) the generally accepted nomenclature, abbreviations, and symbols have been used as far as possible.

Additions to the book include (1) a chapter on refrigeration, prepared largely by Prof. Frank S. Roop, Virginia Polytechnic Institute; (2) answers to problems; (3) a combustion calculation chart (Appendix A), which, it is hoped, will be useful to student and instructor.

The success of the arrangement of the first edition, in which the discussion of internal-combustion engines appears ahead of that of the less familiar steam equipment, has induced the authors to retain the same order of treatment.

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August, 1939.

PREFACE TO FIRST EDITION

Within the past quarter of a century the development of primary power has undergone marked changes as to both magnitude and sources. Formerly the larger part of our power was produced by comparatively small steam-power and waterpower plants. Each industry had its own power plant: a waterpower plant if the industry was so fortunately situated, but more generally a steam-power plant. The internal-combustion engine using liquid fuel was favored for small industries, however, and here and there was to be found a larger installation using producer gas. At the present time, power generation is being rapidly concentrated in large steam-electric and hydro-electric central stations with a vast network of interconnecting transmission lines supplying electric power to industry, to the home, and even to the rural farmstead. There has been developed also a variety of internal-combustion engines, driving motor cars, trucks, and tractors.

These changes in emphasis, the authors believe, justify a new presentation of the subject of heat-power engineering. Every American youth can operate an automobile and knows something of the principles of operation of its engine. This knowledge offers a logical starting point for the development of the fundamentals of heat engines. The authors have therefore departed from the traditional arrangement and have deferred the presentation of steam-power machinery until the more familiar internal-combustion engine has been covered. Furthermore, the approach to thermodynamics appears to be facilitated by this arrangement, the student finding it easier at the outset to understand the "permanent" gas cycles of internal-combustion engines.

This volume is intended for an introductory course in the principles of heat power for students in engineering. Throughout the text an effort has been made to emphasize fundamental principles rather than current practice or mechanical details. It is believed, however, that the student must have some conception of the mechanism used if he is to understand the effects which that mechanism is expected to accomplish. Empirical formulas have been avoided, and, in so far as possible, rational formulas have been developed from basic facts. Although the treatment of the subject is necessarily mathematical, mathematics has been used as a tool and not for the purpose of developing mathematical agility.

In the belief that the student's mastery of a subject is both aided and measured by solving problems; examples have been worked in the text, and a series of study problems has been placed at the end of each chapter.

The authors desire to acknowledge the assistance of numerous manufacturers in supplying illustrations of the various types of equipment. Grateful acknowledgment is also made to the Research Committee on the Properties of Steam of the American Society of Mechanical Engineers for permission to use the new A.S.M.E. steam tables, especially to Prof. J. H. Keenan of the Stevens Institute of Technology for his assistance in supplying advance copies of these steam tables, and to Dean W. M. Cobleigh and Dr. Arnold H. Johnson for valuable suggestions.

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CHAPTER I

INTRODUCTION

1. The Growing Use of Power.—The prosperity of the United States has been attributed in large part to the generous and ever growing use of mechanical power by American industry. This abundant use of power has resulted in increased production per wage earner, reduced costs of manufactured goods, higher wage levels, higher standards of living, release of workers for newly created industries, increased demands for manufactured goods, and, in turn, a further increase in the use of power in the production of goods.

It was estimated[1] that in 1924 the installed power capacity per worker in the United States was 4.3 hp which could be used 8 hr per day by the worker and could do 68,000,000 ft-lb of work. If we assume that the average worker can do 2,000,000 ft-lb of work in his working day of 8 hr, it appears that the average worker in 1924 had the equivalent of 34 helpers. Another estimate[2] for 1924 showed that each man, woman, and child in the United States had the equivalent of 34.6 "mechanical slaves." Similar estimates for other nations gave the following interesting comparisons:

United States	34.6	Japan	2.2
Great Britain	22.9	Italy	2.0
Canada	21.0	Russia	0.64
Germany	20.8	China	0.25
France	8.9		

The total energy supplied by United States central stations to ultimate consumers has increased significantly since the World

[1] *Power*, Jan. 3, 1928, p. 2.
[2] *Mech. Eng.*, May, 1926, p. 531.

War as is indicated by the following tabulation.[1] Energy is in billions of kilowatt-hours:

1920	32.6	1926	56.1	1932	63.7
1921	30.7	1927	61.3	1933	65.9
1922	35.8	1928	67.0	1934	71.1
1923	42.2	1929	75.3	1935	77.6
1924	45.0	1930	74.9	1936	90 0
1925	50.2	1931	71.9	1937	99 3

The use of power for domestic and recreational purposes is having an even more phenomenal growth.

2. Sources of Power.—The relative magnitude of our sources of power is shown in Table I, which is based on rated horsepower

TABLE I.—INSTALLED PRIME-MOVER CAPACITY IN THE UNITED STATES Millions of Horsepower[1]

Year	Steam engines and turbines	Internal-comb. engs.		Hydro-power	Wind-mills	Work animals	Totals
		Other than automobiles	Automobiles and trucks				
1849	1.228	......	....	0.662	0.429	7.747	10.066
1859	3.263	......	....	0.930	0.639	10.961	15.793
1869	6.215	......	....	1.205	0.452	11.275	19.147
1879	11.636	......	....	1.353	0.507	15.324	28.820
1889	24.281	......	0.017	1.522	0.566	21.311	47.697
1899	38.445	0.924	0.032	1.860	0.658	22.274	64.193
1909	77.055	5.712	7.458	4.022	0.822	25.262	120.331
1919	116.880	27.563	219.468	7.650	0.836	24.221	296.118
1923	125.773	72.792	453.530	9.598	0.851	21.500	684.044
1929	....	......	1,960.000[2]	13.571[3]			
1937	....	......	2,960.000[2]	17.120[3]			

[1] *U. S. Geol. Survey, Water Supply Paper* 579.
[2] *Automotive Ind.*, Feb. 26, 1937.
[3] *U. S. Geol. Survey, Annual Rept.*, Jan. 1, 1937.

capacity. This table should be considered as illustrative only because of certain obvious approximations which it contains. For example, animals possess very high overload capacities for short periods. The average horsepower of automobiles, estimated at 30, is probably low for all cars.[2]

[1] *Elec. World*, Jan. 15, 1938, p. 83.
[2] *U. S. Geol. Survey, Water Supply Paper* 59, pp. 74, 75, 1928.

3. Importance of Heat Power.—Table I shows that by far the greatest proportion of our installed capacity in prime movers derives its energy from fuel, the sum of the water, wind, and animal power being only 4.67 per cent of the total for 1923 and a considerably smaller proportion for 1927.

The estimates of the U. S. Geological Survey place the potential water-power capacity in the United States as 38,000,000 hp available 90 per cent of the time, or 59,000,000 hp available 50 per cent of the time. Even if we take the larger figure of 59,000,000 hp. it must be evident from Fig. 1 that we cannot rely on water power for a very large proportion of our power capacity.

But there are obstacles in the way of completely developing these 59,000,000 hp: (1) The most economically feasible water-power sites have already been developed, and those which remain are, generally those which require a heavy capital investment per horsepower. (2) Many of the remaining sites are of small capacity in which operating costs would be high. (3) Many sites are remote and would require expensive transmission lines. (3) Many sites have small water-storage possibilities for seasons of water shortage. In the face of these obstacles, much steam-power capacity is being installed in competition with hydro-power.[1] (5) Among the undeveloped projects are several of very large capacity, for example, those on the St. Lawrence, the Colorado, and the Columbia rivers. These projects will require enormous investments for the initial dam and power-house structures, even though only a small portion of the ultimate generating machinery capacity is installed from time to time, as the demand for power grows.[2] In the meantime, steam and Diesel plants are being built at a much lower total cost and at a lower cost per kilowatt. Hydro power is at its best in the generation of electric power, but even in this field it is being outstripped by heat power. Figure 1, prepared by the Federal Power Commission,[3] shows that installed capacity of electric

[1] An evidence of this is the installation, in 1929, of the first 50,000-hp unit of a 300,000-hp steam plant at Seattle in that state which has the greatest potential water-power resources of all the states.

[2] The initial investment in the Rock Island plant on the Columbia River is reported to be $15,000,000 for a developed capacity of 80,000 hp. The ultimate capacity is to be about 240,000 hp.

[3] *Interim Rept., Power Series* 1, 1935, p. 8.

generating stations of public utilities in the United States is about 28 per cent hydro and 72 per cent fuel.

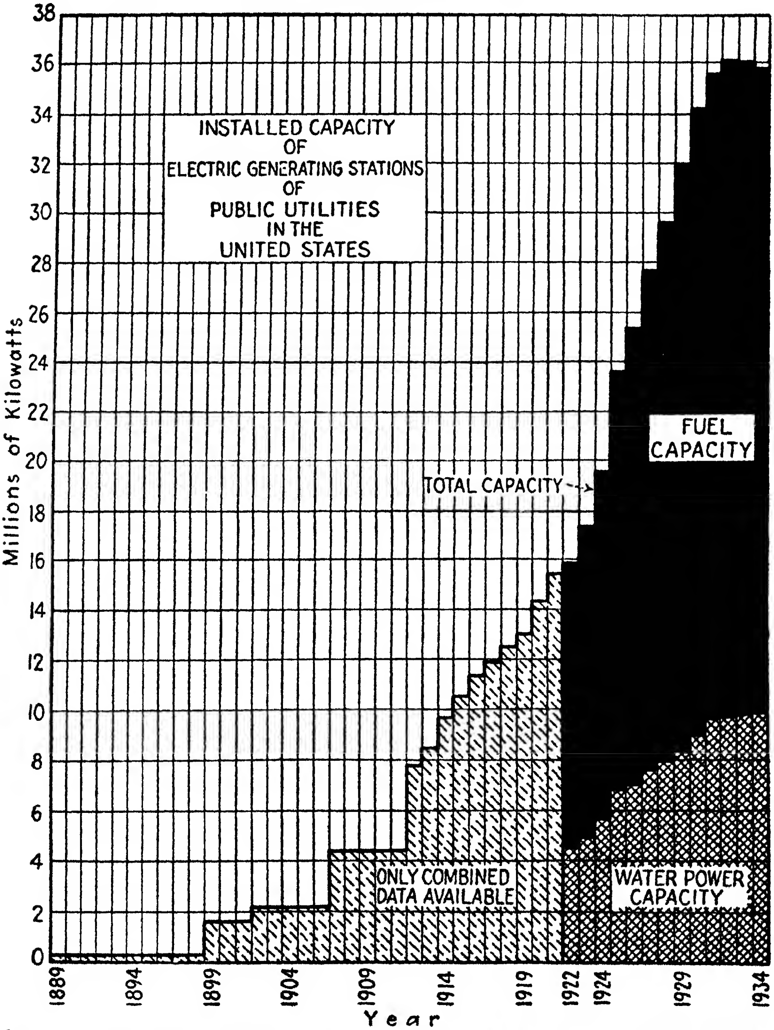


FIG. 1.—Installed capacity of electric generating stations of United States public utilities.

Strictly speaking, the source of all our mechanical power is heat, whether it be named steam, gas, hydro, tidal, wave, solar, terrestrial, or animal power. In some of these the transformation of heat into work is simple and direct, while in others it is involved and remote. For purposes of this book, we shall consider only those powers in which the energy is obtained from the combustion of fuel, and these will be called "heat engines." The magnitude of the quantities of fuel consumed each year justifies the most careful consideration of the problems involved in the efficient transformation of heat into work.

That the importance of high economy in the use of fuel for power generation has not been overlooked by the power industry is evidenced by the steady decrease in the pounds of fuel required to produce a kilowatt-hour. The following tabulation[1] shows the total "equivalent" coal consumed in power plants divided by the total kilowatt-hours generated by heat engines in public-utility plants:

1919	3.2	1925	2.1	1931	1.55
1920	3.0	1926	1.95	1932	1.50
1921	2.7	1927	1.84	1933	1.47
1922	2.5	1928	1.76	1934	1.47
1923	2.4	1929	1.69	1935	1.46
1924	2.2	1930	1 62	1936	1.44
				1937	1.43

[1] *Elec. World*, Jan. 15, 1938, p. 81.

CHAPTER II

THE INTERNAL-COMBUSTION ENGINE

4. Heat Engines.—Heat engines may be classified as "internal combustion" when the fuel is burned inside the working cylinder and "external combustion" when the fuel is burned outside the working cylinder, that is, under a boiler. Internal-combustion engines include gas, gasoline, oil, and even powdered-fuel engines. The gas turbine, which has good prospects of becoming a commercial success, is also of the internal-combustion type. External-combustion engines include steam, vapor, and hot-air engines and steam and mercury turbines.

In the internal-combustion engine the transformation of heat to work is the simplest and most direct. The fuel is taken into the cylinder as a part of the working medium or cylinder contents. The rapid combustion of the fuel produces heat in the resultant gas mixture, which thereby expands and does work. Many methods have been suggested for harnessing the expanding gases, and some of them have been tried out experimentally; only two have thus far proved commercially successful. These are the Otto cycle and the Diesel cycle.

5. The Four-stroke, Otto-cycle Engine.—Practically all automotive and airplane engines and a large part of all stationary and marine engines use the cycle invented in 1876 by Dr. Nicolaus A. Otto (1832–1891). In 1867, Otto and Langen had produced their atmospheric engine, which was both noisy and extravagant of fuel; in contrast, the new engine was called the "Otto silent gas engine." A working model of this engine was exhibited at the Philadelphia Centennial Exhibition in 1876 and the engine patented in the United States, Aug. 14, 1877, under the patent number 194047. A firm of engine builders, which later became the Otto Engine Works, was licensed to manufacture Otto engines, and the first gas engine built in America was produced in 1878. This engine is illustrated in the Frontispiece.

In the Otto cycle there are five principal operations, requiring the four strokes of the piston illustrated in Fig. 2, as follows:

1. The gas mixture containing the fuel is drawn into the cylinder by the outward stroke (Fig. 2*a*), after which the inlet valve is closed.

2. The mixture in the cylinder is compressed by the inward stroke (Fig. 2*b*), into the combustion chamber.

3. Ignition and combustion (explosion) of the fuel take place while the piston is at the inner end of the stroke, the temperature and pressure rising suddenly.

4. The piston now moves outward and the hot gases expand, doing work against the piston (Fig. 2*c*).

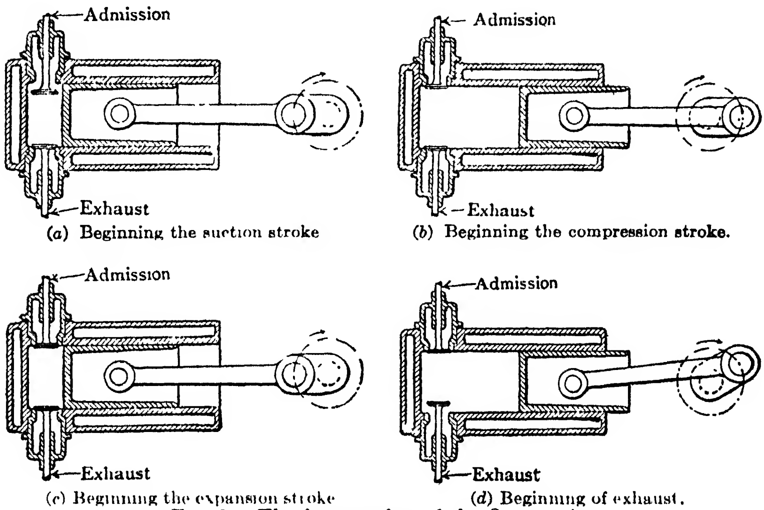


FIG. 2.—The four strokes of the Otto cycle.

5. The exhaust valve is opened, and the piston moving inward (Fig. 2*d*) expels the burned gas mixture to the atmosphere and thus completes the cycle.

In the meantime, the crank has made two complete revolutions to accomplish these five operations, and thus produce one power stroke.

In actual practice, it is necessary to make some modifications in the cyclic events, as illustrated in Fig. 3. It should be understood that the angles given in the figure are illustrative only and would be different for different rotative speeds.

1. It will be noted that the inlet valve opens at about 10 deg (in 1937 cars, varied from 19 deg before to 6 deg after) after the upper or inner dead center and remains open until the piston has started on its upward stroke and the crank has turned about 40 deg (in 1938 cars, 37½ to 71 deg) past the lower dead center. The purpose of this delay in closing the inlet valve is to allow the

incoming fresh charge to fill the cylinder with the greatest possible quantity of the charge. Even so the quantity of the charge, measured at atmospheric pressure is less than the displacement by some factor, say 80 per cent, called the "volumetric efficiency."

2. The compression stroke is now shortened by 40 deg of crank angle, compression continuing until upper dead center is reached, although ignition may occur before upper dead center.

3. Since it requires a measurable time for the flame to spread through the compressed charge after the spark occurs, the timer is set so as to make the spark a few degrees before upper dead center is reached, say about 10 deg for a certain engine speed.

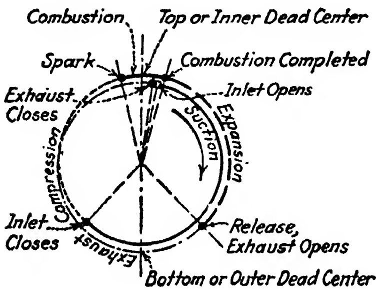


FIG. 3.—Crankpin positions for the events of the four-stroke Otto cycle.

4. Combustion requires an appreciable time and is not completed until 5 to 20 deg after upper dead center, and then expansion continues until the exhaust valve opens. Exhausting the burned gases against the friction of exhaust valves, ports, pipe, and muffler requires considerable force, and for this reason the exhaust valve begins to open at about 45 deg (in 1938 cars, 39 to 57½ deg) before dead center has been reached.

5. After the exhaust gases have got into motion, their inertia produces a slight vacuum in the clearance space, and, to get the full benefit of the suction thus produced, the exhaust valve is allowed to remain open until about 8 deg (in 1938 cars, 1 deg before to 28 deg after) after upper dead center. In some engines, the exhaust valve even remains open a few degrees after the inlet has opened.

6. The Two-stroke, Otto-cycle Engine.—It is possible and quite practicable to accomplish the events of the four strokes of the Otto cycle in two strokes instead of four if we make the piston double acting or, rather, perform certain operations on one side of the piston and other operations on the other side. In this way the underside of the piston is made to draw the fresh mixture into the crankcase and then force it into the cylinder. How this is done is illustrated in Fig. 4, which is a diagram of a two-port, two-stroke engine.

The piston *C* is shown at the lower end of its stroke with inlet port *D* and exhaust port *G* open. When the piston *C* travels upward, a partial vacuum is created in crankcase *B*, thus causing the air to pass into it. As this air passes through the mixing valve *A*, it carries with it a certain amount of gasoline, which is regulated by the needle valve.

When the piston *C* reaches the top of its stroke, the check valve in *A* closes.

On the downward stroke of piston *C*, the mixture in the crankcase is compressed to about 5 lb pressure. As the piston reaches

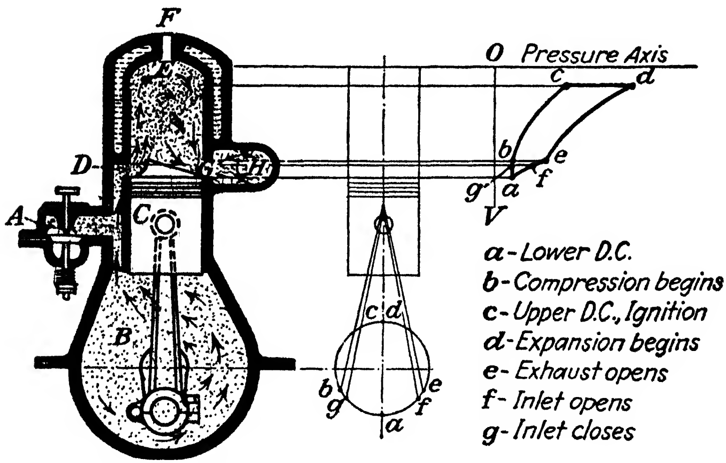


Fig. 4.—Two-port, two-stroke Otto engine.

the end of its stroke, it opens port *D*, allowing the compressed gases to pass from the crankcase into the cylinder. As the piston moves up, it closes port *D* and then compresses the mixture into the combustion chamber *E*. At about the top of the stroke of the piston the mixture is ignited by the spark plug, which fits into opening *F*. This causes combustion, or explosion, forcing the piston to travel downward until it opens port *G*, allowing the exhaust gases to pass into chamber *H*, which connects with the exhaust pipe. While *G* is still open, the transfer port *D* permits a fresh charge to enter from the crankcase, and this very largely displaces the burned gases. This operation is repeated at each revolution of the flywheel, producing an explosion at every revolution.

Figure 5 shows a three-port, two-stroke engine. The three-port engine needs no check valve in the carburetor because the skirt of the piston acts as the inlet valve on the third port which admits air and fuel into the crankcase.

The two-stroke Otto engine is best known in small sizes, although some very large engines have been built using this cycle. The tendency for the incoming charge to mix with the exhaust gases and escape is apt to make the engine less economical than the four-stroke engine.

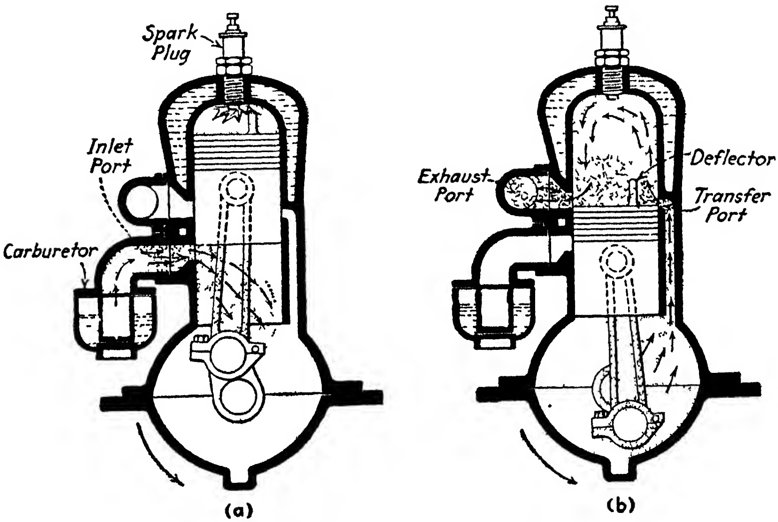


Fig 5.—Three-port, two-stroke engine

7. A Vertical Stationary Gas Engine.—Figure 6 shows a sectional view of a Hope vertical, four-stroke, stationary, gas engine. This engine has inlet and exhaust valves in the cylinder head operated by rocker arms and push rods actuated by a camshaft in the crankshaft. Since the cycle requires two revolutions of the main shaft for its completion, the camshaft must run at one-half engine speed. The camshaft also times the ignition of the charge.

The incoming cylinder charge enters through the mixing valve *K*, and the exhaust gases leave through the water-cooled exhaust manifold *N*.

This engine uses natural gas as fuel and develops 55 bhp per cylinder at 257 rpm with a $14\frac{3}{4}$-in. bore and 16-in. stroke. The clearance or volume of the combustion chamber is 21 per cent of the piston displacement. The weight of this engine is about 190 lb per hp.

8. A Horizontal Stationary Gas Engine.—Figure 7 shows a 170 hp, 180 rpm, Transit 17 by 24-in. twin, single-acting, four-

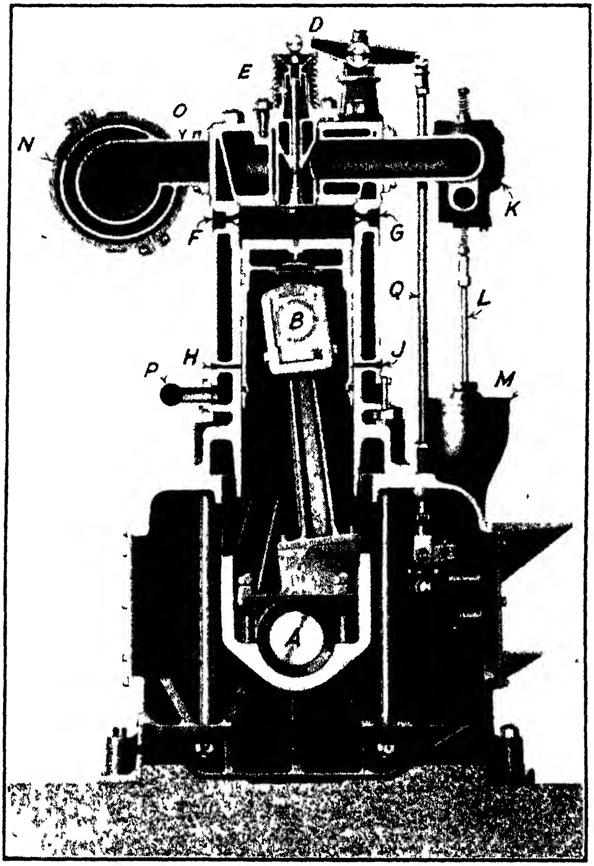


FIG. 6.—Hope heavy-duty gas engine.

stroke engine driving a two-stage natural gas compressor. The fuel used is natural gas.

The valves of the engine *I* and *E* are operated by cams on the lay shaft *L* in the cam housing *H*. The fuel gas is metered to suit the load by the control valve *V* which is, in turn, controlled

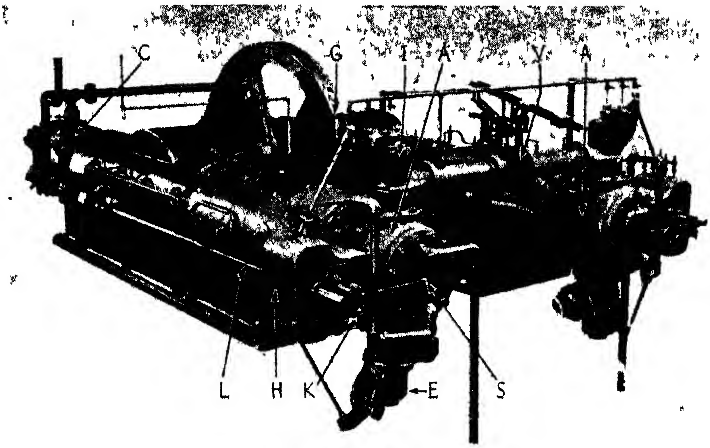


FIG 7.—National Transit 17 × 24-in gas engine driving gas compressors.

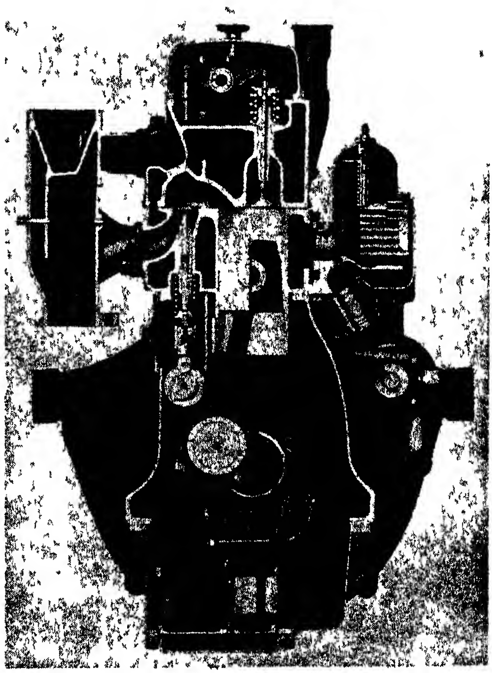


FIG. 8.—Cross section of Waukesha "Hi-Power" engine.

by the governor *G*. The fuel gas mixes with the air in the inlet valve *I* just before entering the cylinder.

The performance of this engine is shown in the table on page 136.

9. An Automotive Engine.—Figure 8 illustrates a modern F-head (see Art. 24), automotive engine of the heavy-duty type designed for truck service. This engine uses gasoline fuel and develops 21 peak hp per cylinder at 2,600 rpm with cylinder dimensions of 4⅜ by 5⅛ in. Its weight per horsepower in a

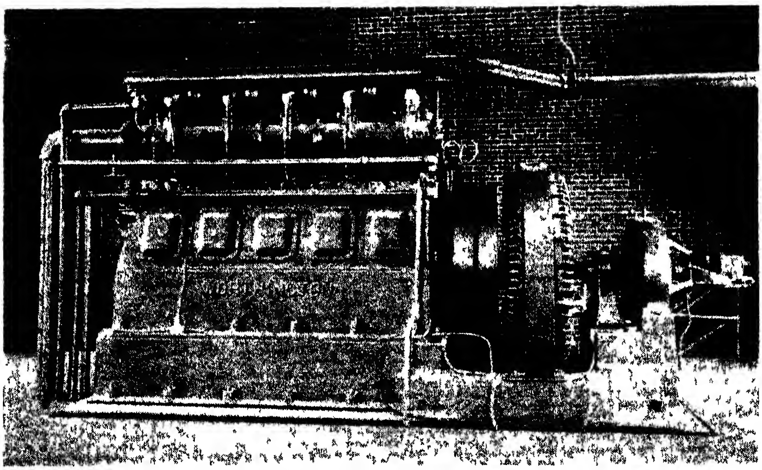


FIG. 9.—Worthington five-cylinder gas engine.

six-cylinder model is about 11.4 lb. Performance curves for this engine are shown in Fig. 79.

An interesting feature of this engine is the peculiar shape of the combustion chamber, which has a constricted opening into the cylinder to produce turbulence of the fuel charge at the end of the compression stroke. This turbulence tends to eliminate or reduce "ping," or detonation, during combustion of the fuel. Other heads accomplishing the same result have been developed by the General Motors Corporation and by other automobile manufacturers.

10. A Vertical Gas Engine.—Figure 9 illustrates a type of vertical, multicylinder engine that has found favor for electric power generation where low-cost gas fuel is available. This engine is of the four-stroke cycle type with valves in the cylinder head. The gas manifold is at the top of the engine with the water-cooled exhaust manifold just below it.

This engine is rated at 300 hp at 400 rpm using natural gas and 210 hp using sewage-sludge gas. It is capable of being converted into a Diesel engine with minor alterations.

11. A Commercial V-type Gas Engine.—Figure 10 illustrates in sectional view an Ingersoll-Rand multicylinder gas engine designed for compactness, lightness, and portability It is built

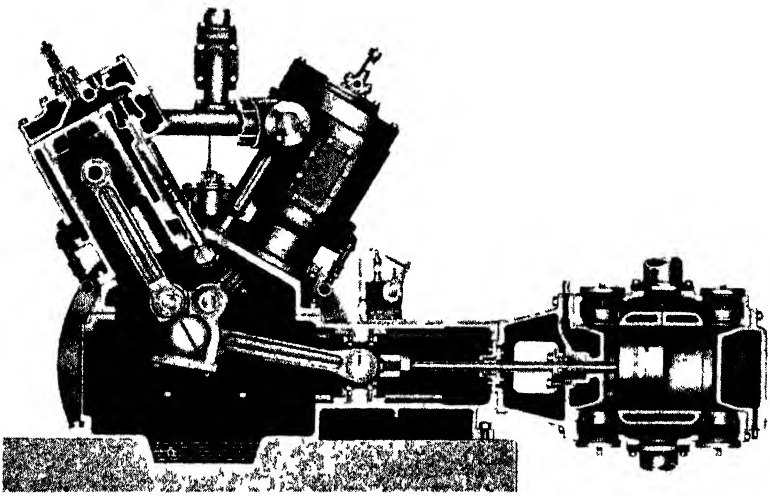


FIG. 10.—Ingersoll-Rand V-type multicylinder gas engine.

with as many as eight power cylinders which have L-heads. Starting is by means of compressed air which is admitted to the small valve in the cylinder head.

The eight-cylinder engine develops 300 bhp at 350 rpm, the bore being 11 in. and the stroke 12 in. The weight is 39,000 lb, including compressor cylinders.

12. An Aviation Engine.—Figure 11 illustrates a type of engine used in aviation service. This is the Wright Whirlwind J-5 engine installed in the Ryan monoplane *Spirit of St. Louis*, in which Colonel Lindbergh made his transatlantic flight on May 20, 1927.

This engine is of the nine-cylinder, air-cooled, static, radial type, using the four-stroke cycle. The rated horsepower is

200 at 1,800 rpm, while the average power developed at this speed at sea level is 223 hp. The bore is 4.5 in. and the stroke 5.5 in., which gives a displacement of 788 cu in. The compression ratio is 5.2:1.

The weight of this engine "dry" is about 508 lb, which gives a weight per horsepower of 2.54 lb at rated power and 2.28 lb per hp at average performance.

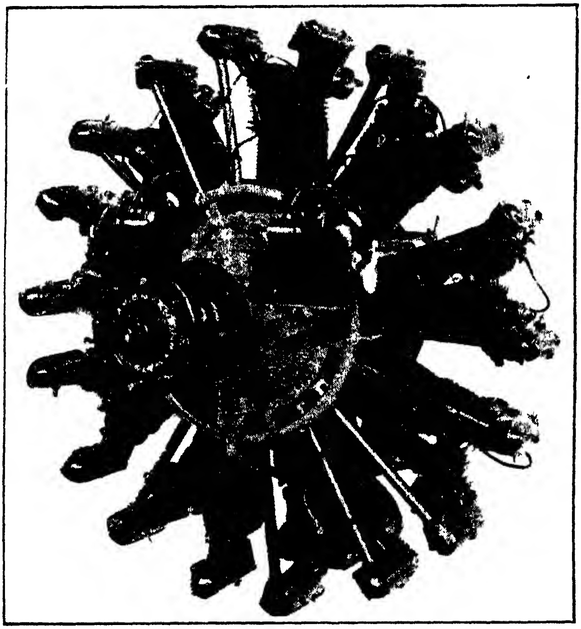


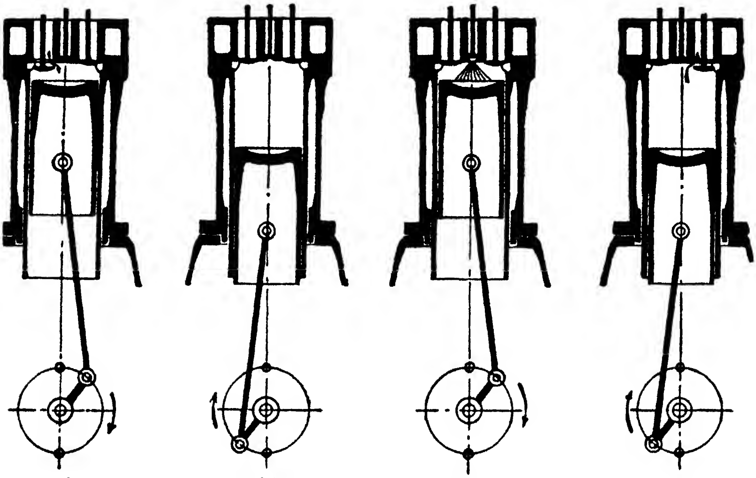
Fig. 11.—Wright Whirlwind airplane engine.

The J-5 engine has a single-throw, counterbalanced crankshaft, with one master connecting rod and eight link rods. The carburetor is at the bottom of the crankcase aft of the cylinders, and the exhaust pipes are just aft of each cylinder. Each cylinder is provided with two spark plugs fired simultaneously from two separate magnetos.

The performance of this engine is shown by the curves of Fig. 81.

13. The Diesel Engine.—The principle upon which the Diesel engine operates is that of progressive combustion of the fuel as it is fed into the cylinder instead of explosion as in the Otto cycle.

Fresh air is compressed in the cylinder to about 500 lb per sq in., which causes a rise in temperature of this air to well above that required for ignition of the fuel. Then, at upper dead center at the beginning of the power stroke, the fuel is injected into the cylinder at such a rate and in such quantity as to develop the power required, the cut-off of the fuel being controlled by the governor.



(a) (b) (c) (d)

FIG. 12.— Principle of the Diesel engine.

(a) Admission stroke. Piston travels downward; admission valve open; cylinder is being filled with pure air.
(b) Compression stroke. Piston travels upward; all valves closed; air in cylinder is being compressed.
(c) Power stroke. Piston travels downward; fuel valve open at top dead center, but closed at fraction of stroke; gases expand.
(d) Exhaust stroke. Piston travels upward; exhaust valve open; burnt gases are being expelled from cylinder.

The diagrams of Fig. 12 illustrate the operations of the four-stroke Diesel cycle, which are as follows:

1. Air without fuel is drawn into the cylinder by the downward stroke, Fig. 12*a*, after which the inlet valve is closed.
2. The air is compressed by the upward stroke, Fig. 12*b*, to a high pressure and temperature.
3. Fuel is injected into the cylinder while the piston moves downward to some point x, Fig. 12*c*, and is then cut off. The fuel burns as it enters the cylinder and comes into contact with the air of the charge.
4. The piston continues to move downward and the gases expand and continue to do work against the piston.
5. The exhaust valve is opened, and the piston, moving upward, Fig. 12*d*, expels the burned gas mixture to the atmosphere.

In the meantime the crank has made two complete revolutions to accomplish this series of operations and thus produce one power stroke.

The two-stroke Diesel cycle is achieved, as shown in Fig. 13, by having an auxiliary "scavenging" compressor on the engine which blows a new charge of fresh air into the cylinder and displaces the spent gases while the piston is still in the position shown in Fig 13(*b*). Each upward motion of the piston thus compresses a new charge of air, and each downward stroke is a power stroke

In general, the four-stroke cycle engine predominates below 1,000 hp, the two-stroke cycle above. The General Motors line of two-stroke cycle automotive engines seems to be an exception

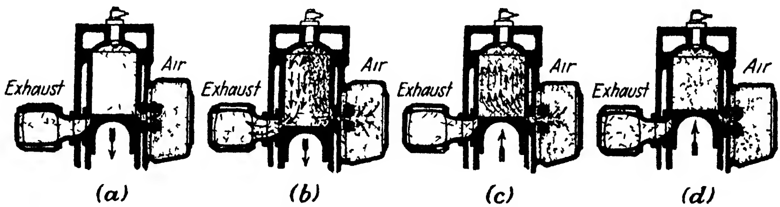


FIG. 13.—Principle of the two-stroke Diesel engine

(*a*) Piston near end of expansion stroke Gas pressure keeps check valves closed.
(*b*) Piston at bottom of stroke Exhaust ports are uncovered relieving cylinder pressure and allowing scavenging air to enter
(*c*) Piston has started up on compression stroke Exhaust ports closed fresh air still entering
(*d*) Piston moving upward compressing the entrapped air charge

Two methods are in use for injecting the fuel into the cylinder at the end of the compression stroke. The original method, devised by Dr. Diesel, is to provide a small auxiliary air compressor driven by the main shaft which produces a pressure of about 1,000 lb per sq in. This high-pressure air blows the fuel oil at high velocity into the cylinder against the 500 lb pressure and atomizes it. This finely divided fuel is then easily ignited by the high temperature of the air in the cylinder. Engines using this type of fuel injection are called *air-injection* Diesels.

The second method of fuel injection is by means of a small displacement pump which pumps the fuel oil directly into the cylinder at just the right instant in the cycle for the injection. The fuel enters the cylinder through a small spray valve which divides the fuel very finely. Engines using this type of fuel injection are called *solid-* or *direct-*injection Diesels.

The advantage of the Diesel engine is that it is capable of using the cheaper grades of low-volatile oils left as residues from the distillation of crude petroleum. Also the thermal efficiency of the Diesel cycle is higher than that of the Otto cycle, because of the greater ratios of expansion and compression.

Figure 14 shows a section through the power cylinder of a Nordberg, two-stroke, scavenging Diesel engine for stationary service. The interesting feature of this engine is its simplicity due to its having no inlet or exhaust valves. This engine uses

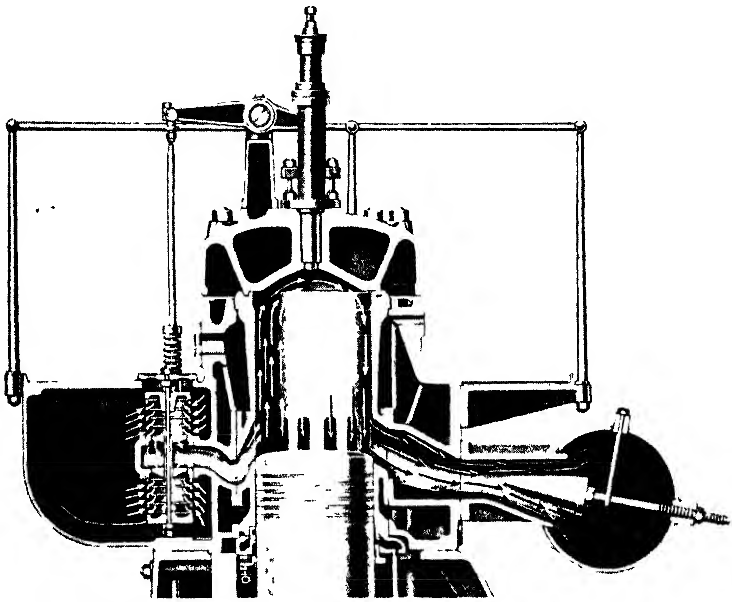


FIG. 14. Nordberg two-stroke, scavenging Diesel-engine cylinder.

air injection, the fuel valve being operated by an overhead rocker arm. The dome-shaped piston faces serve as deflectors for the purpose of causing the incoming scavenging air to be deflected to the ends of the cylinder. The multiple check valve at the left prevents contamination of the fresh air supply by the spent gases.

A Nordberg engine of this type having 21 by 29-in. cylinders and running at 225 rpm develops 2,250 shaft hp after deducting the power consumed by the scavenging cylinder and the air-injection compressor. Guarantee test performance of this engine is given in Fig. 78.

A recent development by Nordberg is a two-stroke Diesel engine, of the type just described, arranged to use either oil or gas fuel. Gas fuel is compressed in the air-injection compressor to about 1,100 lb and is admitted to the cylinder through the regular air-injection valve. A small amount of "pilot" oil fuel is used to insure ignition of the gas. Figure 34 shows the details of an air-injection valve such as is used on this engine, the gas being used to inject the oil.

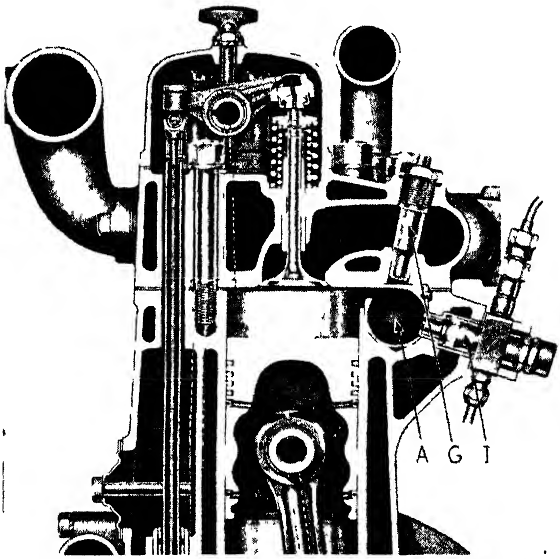


FIG. 15.—Hercules automotive Diesel cylinder in section.

14. Automotive Diesel Engines.—Since the Diesel was originally a high-pressure engine, it was long felt that it would not be feasible to build lightweight high-speed engines of this type for automotive purposes. Recently, however, several manufacturers have placed bus, truck, and tractor Diesels on the market, and automobile and airplane Diesels have been successfully produced. Figure 15 illustrates a four-stoke, solid-injection, Hercules Diesel engine designed for commercial vehicle service, the distinctive feature of which is the spherical precombustion chamber *A*. This chamber is so arranged that it sets the air charge in intense swirling motion on being compressed, which causes the fuel spray from the nozzle *I* to be quickly and thoroughly mixed with the air. The precombustion chamber, or disked cylinder or pistonheads, for producing turbulence in the charge is found in all high-speed Diesel engines. *G* is a "glow plug."

This engine in the six-cylinder, 5 by 6-in. size, develops 160 hp at 1,600 rpm and weighs 2,300 lb. Its compression ratio is 14.5 to 1. Performance curves for this engine are given in Fig. 79.

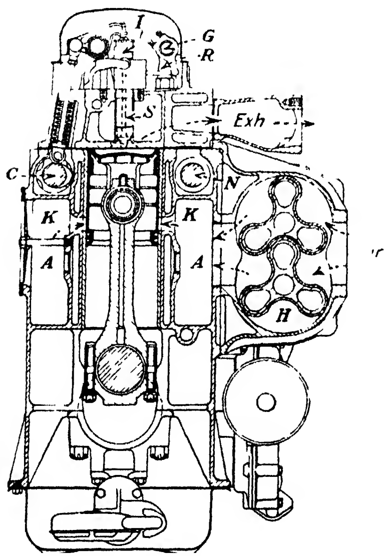


FIG. 16.—General Motors two-stroke, uniflow, Diesel engine.

General Motors has recently developed an automotive Diesel using the two-stroke principle for the purpose of reducing the weight per horsepower. Figure 16 illustrates a cross section of this engine. Air is forced by the kinetic blower *H* into the air boxes *A*, *A*. When the ports *K*, *K* are uncovered by the piston, the scavenging air blows the spent combustion gases out through two exhaust valves *S*, shown dotted, in the cylinder head. The overhead valves and the injection pump *I* are operated by the camshaft *C*. The camshaft and the opposite shaft *N* carry centrifugal balancing weights. The governor shaft *G* moves the rack *R*, which controls the fuel charge (see Art. 22).

This engine has $4\frac{1}{4}$ by 5-in. cylinders, and the six-cylinder model, weighing 1,635 lb, develops 160 hp, maximum, at 1,800 rpm and 82.7 bmep. The compression ratio is 16 to 1.

15. The Fuel-injection Spark-ignition Engine.—One of the disadvantages of the usual Diesel cycle is the high cylinder pressure at the beginning of the stroke, and several modifications have been proposed to obviate this. Also, as will be pointed out in a later paragraph, Art. 85, the efficiency of the Otto cycle for a given compression ratio is higher than that of the Diesel cycle.

The Swedish Hesselman type of engine is designed to take account of these two facts. It uses a fairly high compression

(125 lb) for good efficiency but not high enough for fuel ignition. About 50 deg before the end of the compression stroke, the fuel

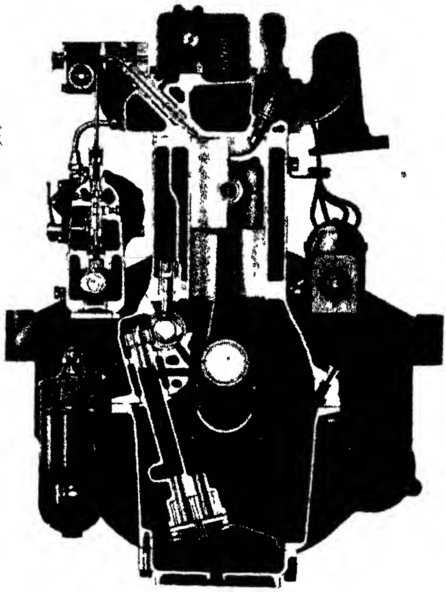


Fig. 17.—Waukesha Hesselman automotive-type oil engine.

oil, metered by the governor mechanism, is sprayed into the combustion space and then a few crank degrees later is ignited by an electric spark.

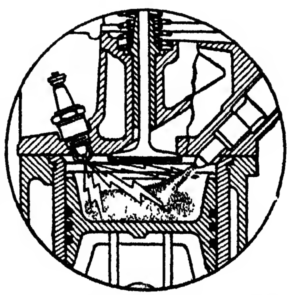


Fig. 18.—Head of Allis-Chalmers oil tractor engine.

Figure 17 illustrates the Waukesha Hesselman engine showing the cup-shaped pistonhead into which the fuel is injected, giving it high turbulence and thorough mixing with the air. Fuel is injected by means of the Bosch injection pump on the left side of the illustration, and the electric spark is provided by the Bosch high-tension magneto at the right (Art. 20). This engine, in the six-cylinder, 8.5 by 8.5-in. size develops 300 bhp at 900 rpm and weighs 7,950 lb. Performance curves are shown in Fig. 79.

Figure 18 illustrates the combustion chamber of the Allis-Chalmers oil tractor engine of the fuel-injection, spark-ignition type. Solid injection by means of the Deco equipment, Art. 26, is used and ignition is furnished by a Mallory high-tension spark coil using battery power, Art. 20. An indicator card and performance curves for a six-cylinder, 5.25 by 6.5-in. engine are shown in Fig. 80.

16. The Semi-Diesel Engine.—Another type of engine, which combines some of the features of the Otto and the Diesel engines, has found favor with some manufacturers. These engines are called by various names, depending upon the chief characteristic of the cycle, such as "heavy-oil engines," "semi-Diesel engines," "hot-bulb ignition engines," "dual-combustion engines," or "low-compression oil engines."

In the hot-bulb engine the compression is carried to from 150 to 250 lb per sq in., and then the fuel is injected and ignited by some "hot spot" in the cylinder which acquires a high temperature because it is not cooled by the cooling system. Such an engine must have some auxiliary means of heating the hot spot or hot bulb for starting.

A typical low-compression or dual-combustion Diesel engine built by Ingersoll-Rand compresses the air charge to about 250 lb per sq in. Fuel injection begins at about 10° of crank angle before top dead center, the fuel entering as two finely divided sprays into the combustion space at the top of the cylinder. The two sprays meet at the center of the combustion zone and ignition occurs almost instantaneously due to the heat of the compression. Fuel injection is so timed that combustion starts while the piston is at top dead center and is at such a rate as to keep the pressure about constant as the piston moves away from top dead center until the fuel is cut off. The combustion is called "dual combustion," because the first part is at constant volume and the second part at constant pressure. This engine develops 120 hp per cylinder at 240 rpm in cylinders having a bore of 17¾ in. and a stroke of 22 in. Further details of the performance of this engine are given in Table V, page 136.

From the foregoing paragraphs it will be seen that the sharp distinctions between the Otto and Diesel engines are becoming less sharp and that the internal-combustion engine of the future is likely to appear in many compromise designs, each of which will possess advantages for particular services.

17. The Powdered-fuel Engine.[1]—It was Dr. Diesel's original intention to operate his engine on coal dust, but he was forced to use oil in order to make his engine function. The idea persisted, however, in the mind of his associate, Rudolf Pawlikowski, who

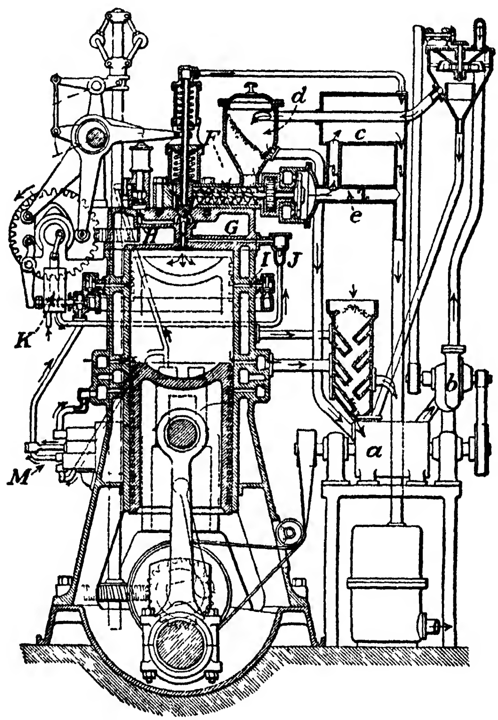


FIG. 19. Section of the Pawlikowski coal-dust engine.

has continued his experimentation and, in 1928, had two engines in Germany running successfully on coal dust.

Figure 19 shows a cross section of a three-cylinder, 180-hp engine with its belt-driven pulverizer plant at the right. The dust after passing the pulverizer *a* is elevated by the blower *b* to the separator where the coarse material is returned to the pulverizer and the fines are sent to the hopper *d*. Directly under the hopper *d* are two screw conveyers *F* side by side, which

[1] *Power*, Nov. 6, 1928.

cause a stream of coal-dust-air mixture to travel continuously around the valve *G*. In this way the coal dust is kept fluid. The screw conveyers are driven by a tiny turbine operated by the escaping exhaust gases. The coal, before being pulverized, is dried by the exhaust gases.

18. The Still Engine.[1]—A combination internal-combustion and steam engine that has been brought out in England since the World War by W. J. Still has attracted considerable attention. This engine uses the heat of the exhaust gases to generate steam, which is supplied to the underside of the piston in such a way as to make a double-acting cylinder with gas pressure above and steam pressure below. The heat from the waste gases is transferred to the cooling water of the combustion cylinder jackets in a "regenerator." The regenerator is a steam boiler which is heated by the exhaust gases when the engine is in normal operation but by a separate oil burner when starting up or maneuvering. The combustion cylinder operates on the two-stroke Diesel cycle and, thus, produces one gas-power stroke and one steam-power stroke per revolution.

The proportion of steam-horsepower to total shaft-horsepower output for a series of tests on this engine varies from 0 at 25 per cent engine load to about $\frac{1}{4}$ at full engine load.

The Still combination principle has, also, been applied to locomotive engines.

Problems

1. The gear on the camshaft of a gas engine has 60 teeth and the gear on the main shaft has 30 teeth If the two gears are meshed wrongly by one tooth, how much are the cyclic events in error?

2. A gas engine running at 1,800 rpm has a stroke of 6 in. (crank 3 in.) and a connecting rod 13 in. long. (*a*) How far does the piston move between ignition and upper dead center if the spark occurs 10 deg before dead center? (*b*) If the power stroke continues for 145 deg of crank angle, how far is the piston from lower dead center at release?

3. A gas engine runs at 1,200 rpm. If the average distance from the spark-plug points to the gas molecules is 2 in. and the rate of flame propagation is 50 ft per sec, how many degrees of crank angle should the spark occur before upper dead center to have combustion half completed at dead center?

4. An engine has a 24-in. connecting rod and a 6-in. crank. Through how many degrees does the crank turn while the piston moves 5 in. (*a*) from upper dead center; (*b*) from lower dead center?

[1] See *Engineering*, Mar. 3, 1922, p. 275; Apr. 15, 1927, p. 451.

5. The two-stroke engine in Fig. 4 has a bore of 4 in. and a stroke of 5 in. with an 11-in. connecting rod. Port G is $1\frac{5}{16}$ in. wide and port D is $\frac{3}{4}$ in. wide. Draw the cyclic events on the crankpin circle for this engine.

6. An early eight-cylinder car was fitted with a single-spark contactor for all eight cylinders. The tires were 33 in. and the engine-to-wheel speed ratio was 4.25 : 1. How many spark contacts were made per second at 40 mph?

7. The Wright J-5 200-hp engine has nine 4.5-in. bore cylinders with 5.5-in. stroke and runs at 1,800 rpm. What is (*a*) its displacement per revolution; (*b*) its working-stroke displacement per minute; (*c*) its working-stroke displacement per minute per horsepower?

8. The Hope engine of Fig. 6 develops 55 hp at 257 rpm per cylinder 14.75 by 16 in. What is its working-stroke displacement per minute per horsepower?

9. What is the working-stroke displacement per minute per horsepower of a two-stroke double-acting Diesel engine if each cylinder develops 725 hp at 95 rpm with a 28-in. bore and 40-in. stroke?

10. What is the working-stroke displacement per minute per horsepower of a 15,000-hp. Diesel-engine having nine 860 by 1,500-mm, double-acting, two-stroke cylinders and speed of 94 rpm?

11. A four-cylinder, four-stroke automobile engine has a $3\frac{3}{4}$-in. stroke and 4-in. bore and has a 1-in. carburetor opening. What is the average air velocity through the carburetor at an engine speed of 1,200 rpm? Assume a volumetric efficiency (ratio of free air drawn in to displacement) of 75 per cent.

CHAPTER III

INTERNAL-COMBUSTION ENGINE DETAILS

Practical Problems of the Internal-combustion Engine.—In working out the practical operation of the events of the Otto engine, several problems are encountered. The most important of these, which will be discussed here, are carburetion, ignition, lubrication, governing, and cooling.

19. Carburetion.—In order to make an explosive charge, it is necessary to "carburet" the air by mixing with it the proper proportion of the fuel, usually hydrocarbons. On engines using gaseous fuels, the carburetor is simply a mixing valve which opens the gas-supply and air-supply ports simultaneously in the correct ratio. The part K of Fig. 6 is such a valve.

For liquid fuels a vaporizer must be provided. Small two-stroke engines for marine use frequently have a device similar to that shown on the engine of Fig. 4 and called a *mixing* or *carburetor valve*. This consists of a spring-loaded check valve with a conical face in the seat of which is a small gasoline orifice controlled by a needle adjustment. As the suction of the piston draws the inlet air through the check valve, the gasoline is drawn along with the air into the engine. The liquid level in the supply chamber must be slightly below the orifice.

Constant-speed stationary engines may use a simple mixing valve with two orifices, one for the gasoline and one for the air, which are adjusted in one direction by hand and are controlled in another direction by the governor. Other engines of this type use a simple carburetor of the form shown in Fig. 20. The gasoline is admitted to the chamber k through the float valve c, which is closed by the float b when the liquid level reaches the dotted line. . The flange at the top is connected to the inlet of the engine and the suction of the pistons causes a strong rush of air in the direction of the arrows through the narrow throat or "venturi" j. The top of the nozzle e is slightly above the gasoline level in the float chamber, and the suction produced by the air flow through the venturi draws the gasoline through the orifice *a* and mixes it with the air. The opening of the orifice *a*, and consequently the mixture richness, is controlled by the needle *g*.

For automotive engines which operate under widely varying conditions of load and speed, a more complex device is needed. The elementary carburetor of Fig. 20 has the disadvantage that the flow of gasoline through the nozzle orifice is not proportional to the quantity of air flowing through the venturi but

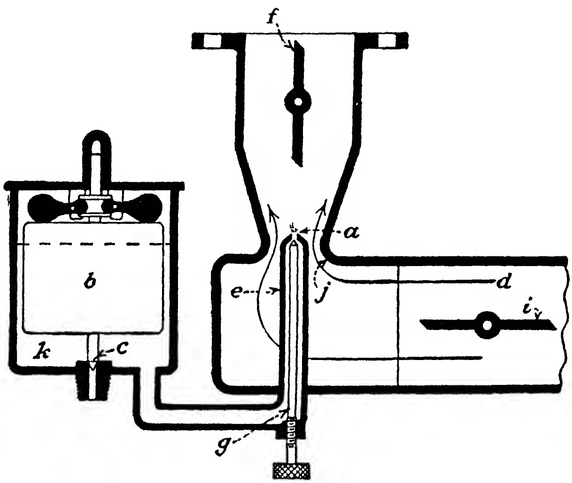


FIG. 20.—Elementary plain-tube carburetor.

produces a lean mixture at low air velocities and a rich mixture at high velocities.

The ideal carburetor for an automotive engine, according to the Purdue Engineering Experiment Station,[1] is one that produces (1) a rich mixture for idling, (2) a lean mixture of nearly constant strength for level road operation, and (3) a rich mixture for full-load operation. As a result of an elaborate series of tests, the Purdue Experiment Station arrived at the values of the air-fuel ratio for various air flows and have expressed these as a curve. This is shown as the dotted "ideal" curve of Fig. 21.

A great many variations of the elementary carburetor have been devised to give rich mixture for idling and full load and an economical one for average running conditions. One carburetor

[1] Commercial Carbureter Characteristics, *Bull.* 21, 1925.

which has made elaborate provisions for meeting these conditions is shown in Fig. 22. This is the Schebler model "S" and is of the type called an "air-fuel proportioning" carburetor having an air valve and two air inlets. The fixed-air inlet is through the venturi *w*, which is very small, furnishing barely enough air to enable an engine to idle slowly. The auxiliary air valve *c* is closed for idling and wide open for full power.

The gasoline enters the float chamber through the valve *r*. The quantity admitted to the mixing venturi *w* is governed by the needle *m*. The motion of the air valve *c* is transmitted to the needle-valve lift lever *b*, which rotates about a fulcrum point *a*. Thus the flow of gasoline varies in proportion to the

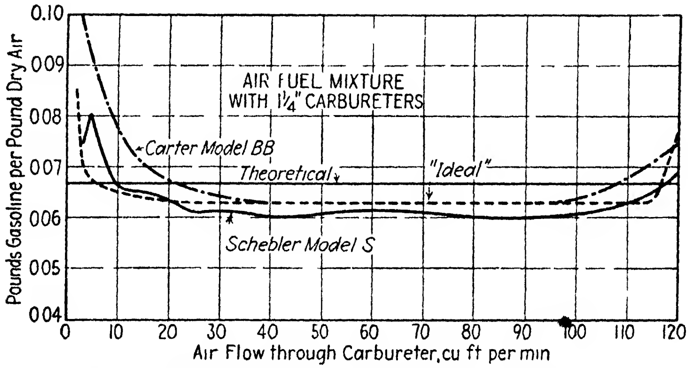


FIG. 21—Metering characteristics of two carburetors

air entering the engine When the throttle is wide open for full power and a rich mixture is wanted, the cam *A* depresses the lever *D* which slides the fulcrum point *a* along the lever *b* toward the air valve, thus lifting the metering pin *l*. This will produce a rich mixture at full throttle.

When the throttle is opened suddenly, the accelerating pump *q* lifts a quantity of gasoline, which enters the holes in the piston and discharges it into the venturi near the main gasoline nozzle, thus enriching the mixture for acceleration. If the throttle is opened slowly, the gasoline flows out of the holes in the piston.

The dashpot on the air-valve stem is to prevent fluttering. When the throttle is opened suddenly, the upper spring compresses and restrains the air valve until the dashpot piston has moved to its new position. The little valve in the dashpot piston allows the air valve to close quickly.

The metering characteristics of a carburetor of the air-fuel proportioning type are shown by the full curve of Fig. 21.

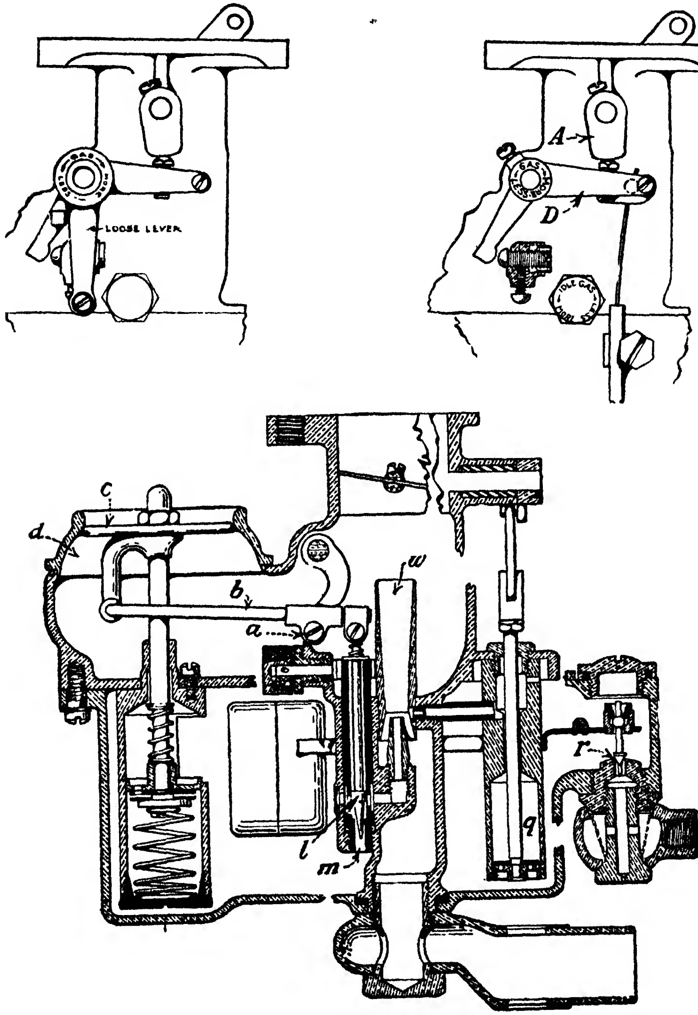


FIG. 22.—Schebler model "S" up-draft carburetor.

A carburetor of the down-draft type is the Carter *BB* shown in Fig. 23. The fuel enters the main metering jet *E*, from the float chamber, and flows up the sloping passageway to the two outlets *O* on the sides of the boat-shaped bridge across the venturi throat, the fuel flow increasing as the air flow through the venturi increases, approximately in proportion to the air flow.

When the throttle T is opened wide for acceleration or for a hard pull with low engine speed, a rich mixture is needed. At such times the manifold vacuum, which is transmitted through passage F, is reduced to such a point that it is insufficient to hold down the step-up piston A against its spring. As A moves up

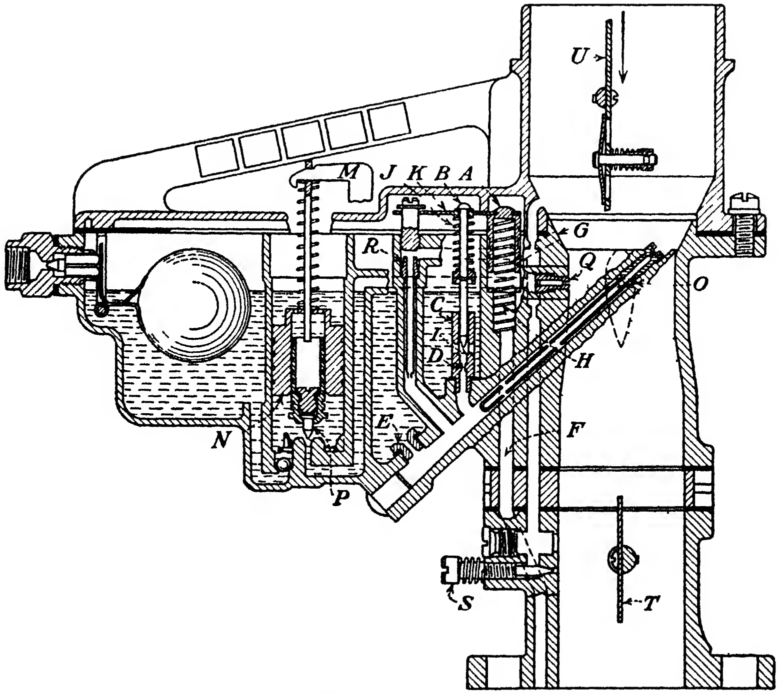


FIG 23—Carter Model *BB* down-draft carburetor.[1]

it allows the step-up needle valve B to rise, permitting additional fuel to flow through orifice D.

When the throttle is opened suddenly, the accelerating pump N is pulled down by hook M pressing against the coil spring. This lifts needle valve P from its seat and forces fuel through the passages and out of accelerating jet Q. As the pressure continues, P settles back to its seat, and, when M is again lifted, fuel flows past the ball check valve.

[1] Copyright by Carter Carburetor Corp. Used by permission.

On idling, the throttle is nearly closed, and the suction through O is very small. The fuel then rises in the sloping passage and up to the orifice in idling jet R. Fuel is then lifted, by the vacuum under the throttle, through R and discharged past idling adjustment screw S.

The characteristics of this carburetor are shown in Fig. 21.

For carburetion of gaseous fuels it is necessary only to thoroughly mix the fuel and air in proportions for perfect combustion.

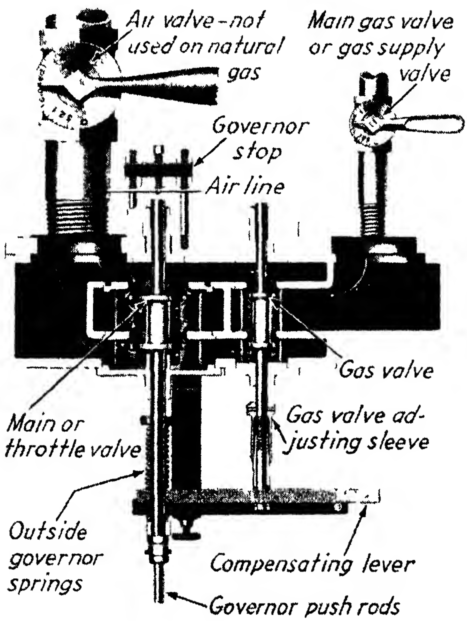


Fig. 24.—Ingersoll-Rand gas-fuel mixing valve.

In the Transit engine of Fig. 7 the fuel is introduced through 16 small jets and thus thoroughly mixed with the air immediately before the charge enters the inlet valve. Figure 24 illustrates the Ingersoll-Rand gas-fuel mixing valve which is provided with hand-adjustable throttle valves to care for change in heat value of the fuel. It will be noted that the balanced fuel and mixture valves are controlled separately by the governor levers at the bottom and that the fuel is mixed with the air *before* passing the main throttle valve. Figure 25 shows the Worthington mixing valve used on the sludge-gas engine of Fig. 9. In this valve the two valves are on the same stem and mixing occurs *after* throttling.

20. Ignition.—Otto's early engines were equipped with an open gas flame for igniting the charge through a small valve-

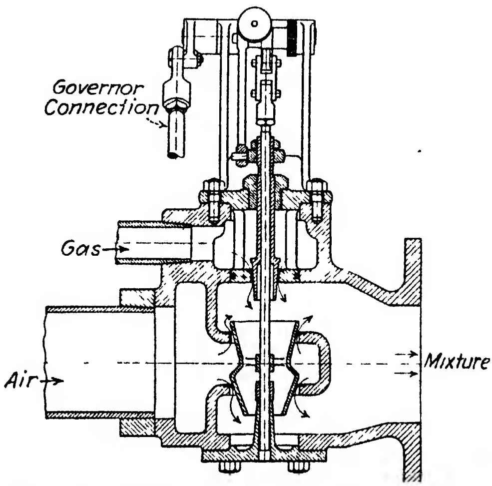


Fig. 25.—Worthington air-and-gas mixing valve.

controlled port in the cylinder head. The port was closed promptly, but the flame was extinguished by the explosion and had to be relighted by a second flame.

Electric ignition is now almost universally used for Otto engines. Two systems are in use, the *jump spark* and the *make*

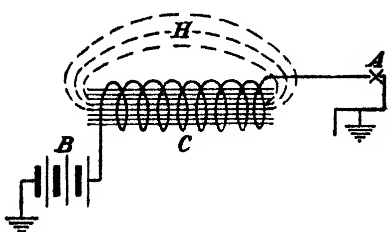


Fig. 26.—Self-induction, low-tension coil circuit.

and break. Figure 26 shows a diagram of a make-and-break system in which a battery *B* is connected through a self-induction coil *C* to a make-and-break device *A* inside the combustion space of the cylinder. The second terminal of the battery is grounded to the engine, as is also the contactor in the cylinder. Shortly before the time for ignition, the contact is made and, at the proper instant, broken again, and the arc set up between the electrodes ignites the charge.

The coil *C* has an iron core which is highly magnetized by the current flowing. On opening the circuit at the electrodes, the decaying magnetic flux *H* cuts the turns of the coil windings and generates in the coil an electromotive force in the same direction as the original battery potential; this has the effect of prolonging the arc. The voltage obtained from a coil energized by a 6-volt battery may be as much as 100 or 200 volts.

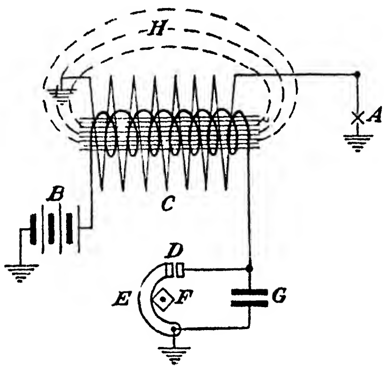


FIG. 27.—High-tension coil circuit.

Stationary engines using lean fuels which require high compression commonly use this system with 110 volts direct current for the power supply. The gas engine of Fig. 7 uses make-and-break ignition. The igniter *S* is operated from the lay shaft.

Figure 27 is a diagram of a "single-spark," jump-spark system in which a battery *B* is connected to the coarse winding of the coil *C*, through a timer-breaker *E* operated by the cam *F* on the timer shaft of the engine. One side of the battery and of the breaker is grounded. Well before time for ignition, the contact *D* is made and the iron core of the coil *C* is highly magnetized by the current flowing. When the cam *F* opens the contact *D*, the decaying magnetic flux *H* cuts the large number of turns of the secondary coil winding and generates in the coil a very high voltage which is sufficient to jump across the 0.020-in. air gap between the points of the spark plug *A*.

The sparking voltage necessary for spark plugs[1] depends upon length of gap, shape of electrodes, temperature of electrodes, and density of the gases in the cylinder. The effect of density has been shown to be expressed by the formula

$$E = 1.1 + 1.7\rho$$

in which E = sparking voltage in kilovolts.

ρ = density of the gas relative to air.

It has also been shown that electrode material, mixture ratio, and turbulence have very little effect on the breakdown voltage.

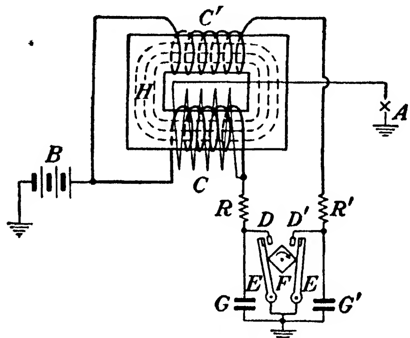


FIG. 28.—Mallory ignition circuit.

Modern high-compression engines require a hotter spark than engines of a few years ago.

The decaying flux also cuts the turns of the primary winding, which generates an emf in that winding which tends to prolong the primary current and thus maintain the flux. Since the secondary emf depends upon the rate of decay of the flux in the core, it is important to absorb the energy very rapidly, and to this end a condenser G is placed across the contact points. The condenser also serves to protect the points from burning due to arcing.

The Mallory ignition circuit is shown in Fig. 28, the operation of which is as follows. Contact D is closed by the cam F allowing a battery current to flow through primary coil C and resistance R, thus building up a strong magnetic field H in the closed magnetic circuit. An instant later contact D' is closed, thus building up a magnetic field which tends to neutralize that produced by

[1] *N.A.C.A. Rept.* 10 (1934), p. 357.

coil C. Contact D now opens, and the magnetic lines are forcibly reversed by coil C'. This very rapid decay of the magnetic field produces a very high potential in the secondary and a hot spark at A. This equipment is used on the Allis-Chalmers oil engine, described on page 21, which compresses to about 125 lb.

A high-tension magneto circuit is shown in Fig. 29. The armature B carries a double winding of coarse primary wire of few turns and a fine secondary wire of many turns. The primary winding is connected to the contact point D and the arm E which rotates with the armature B at camshaft speed. The cam F is stationary, and the contact points are closed except at

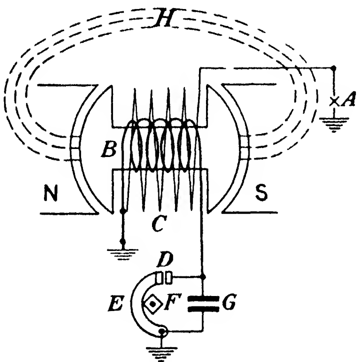


Fig. 29.—High-tension, shuttle-armature magneto circuit.

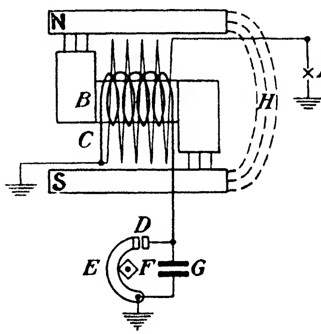


Fig. 30.—High-tension, inductor magneto circuit.

the instant of spark. The rotation of the armature causes the flux to flow first in one direction and then in another through the core, and this induces a current in the primary winding, which intensifies the flux through the armature core. When the primary circuit is broken, the decay of the flux generates the high voltage in the secondary winding in the same manner (Fig. 27).

Figure 30 shows the arrangement of an "inductor" type of high-tension magneto in which the inductor B rotates about a horizontal axis in the plane of the paper and causes a reversing flux to flow through the primary winding. Both the primary and the secondary coils are stationary. The action is similar to that of the rotating type.

21. Lubrication.—The lubrication of the pistons of internal-combustion engines presents two problems. (1) The oil must be of a kind that will retain its "body" or viscosity at high temperatures; and (2) the oil must not "crack," see Art. 27, under the action of the high temperature, thus giving a combustible vapor and leaving a carbon deposit in the cylinder.

Small gas engines use the splash system, in which the oil is splashed on the cylinder walls by the rotation of the crank from troughs in the crankcase. This splash carries oil to the wrist pins, crankpins, main bearings, and camshaft as well.

Large gas engines and the better makes of automotive engines use force systems in which a displacement pump or a gear pump takes oil from a sump in the crankcase and forces it to the bearings and other parts to be lubricated. Some engines use a combination of forced and splash feed.

The lubrication system of the automotive engine of Fig. 8 is by force feed to the main and crankpin bearings and splash for the wristpin and camshaft. The conventional oil pump is of the gear type and may be seen in Figs. 8 and 17.

The cylinder of the Hope engine of Fig. 6 is lubricated by cup feed or forced feed through the two small holes *H* and *J* about the middle of the barrel. The piston has a ring near the bottom with leak-offs drilled from the groove into the inside for the purpose of scraping off and removing the excess oil.

The Transit engine of Fig. 7 has a lay shaft along the cylinders which drives sight-feed force pumps. This lay shaft, also, operates the make-and-break igniters, which may be seen in the illustration.

22. Governing.—In order to maintain the desired speed, a gas engine must have some means of adjusting the power output to the load. For automotive or marine engines this is done manually by the operator, who adjusts the throttle to suit conditions. For stationary, constant-speed engines a governor is provided for automatically proportioning the fuel to the load.

Three methods are commonly used in governing gas engines: (1) hit and miss, (2) throttling, and (3) cut-off.

Hit-and-miss governors vary the number of power strokes made per minute by keeping the exhaust valve open and the inlet valve closed during some of the inspiration strokes, thus preventing a power impulse on the succeeding downward stroke. Such governors are common on small, single-cylinder engines. The advantage of this governor is that it gives constant quantity and quality of the air-fuel charge, while the disadvantage is that the regulation is not very close. "Regulation" means the variation in speed from no load to full load expressed in percentage of full-load speed.

Throttling governors control the quantity of the air-fuel mixture by means of a speed-operated throttling valve similar to the manual-control valve of an automobile. The disadvantage of this method of governing is that the compression varies with the quantity of the mixture charge drawn into the cylinder, and at light loads the engine becomes inefficient (see Art. 91).

Cut-off governors function by controlling the quantity of fuel mixed with the air, while keeping the quantity of the mixture constant. This is sometimes called "quality" governing. The cut-off method has the disadvantage, in the Otto cycle, that at light loads the mixture may become so lean that combustion is slow and the engine inefficient. Diesel engines are governed by cut-off governors, the disadvantage just stated not being applicable to the Diesel cycle.

Most engines using these methods of control depend for their sensitiveness to speed upon some form of centrifugal governor, which consists essentially of a pair of flyballs mounted on a spindle rotated in synchronism with the engine, through either mechanical or electrical drive. The centrifugal force tending to throw the balls apart is resisted by a spring that can be adjusted to adjust the normal speed, the motion of the balls being transmitted to a sleeve. The motion of this sleeve may be applied to the valve which controls the flow of fuel to the engine (or steam or water to a turbine) either by (1) direct mechanical connection or by (2) indirect hydraulic means.

The gas-engine valve of Fig. 24 is connected mechanically to the governor sleeve as are also the Diesel engine governors of Fig. 35 and Fig. 36.

Figure 31 is a diagrammatic representation of the Woodward hydraulic governor. At the top is the speed-sensitive, flyball element, the functions of which are to take account of (1) the instant of, (2) the magnitude of, and (3) the direction of a speed change due to load change. At the instant illustrated a part of the load on the engine has been dropped and the flyballs have moved outward from the vertical position due to an increase in speed. This causes the pilot-valve plunger to be lifted, thus uncovering the oil port and allowing the oil above the power piston to be discharged to the sump and the power spring to move the piston upward. The power piston rod, being connected to the energy-medium control mechanism of the prime mover, reduces the flow of energy and reduces the speed.

The upward motion of the actuating compensating plunger attached to the power piston causes a reduced pressure in the cylinder which allows the receiving compensating plunger to rise,

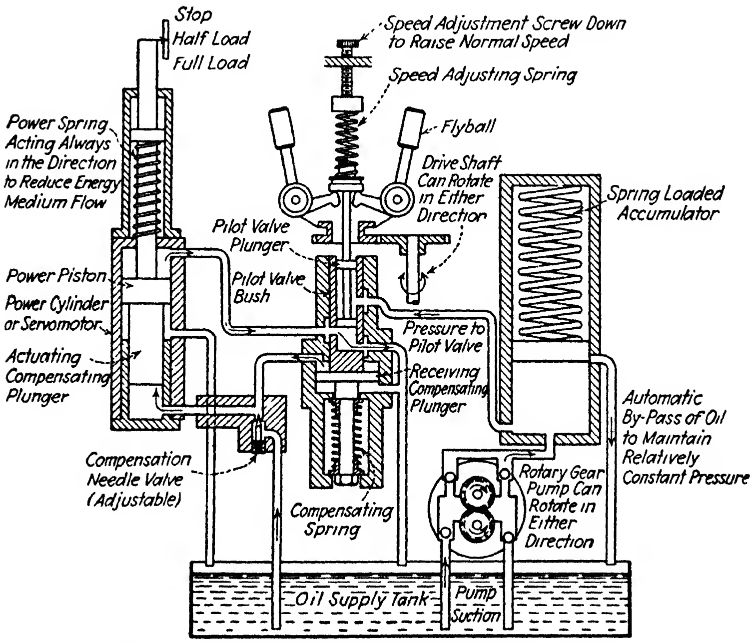


FIG. 31.—Principle of the Woodward hydraulic governor.

lifting with it the pilot-valve bush. This continues until the port in the pilot-valve bush is again covered by the pilot-valve plunger and oil flow from power cylinder, and consequently the upward movement of the power piston is stopped. Now if the movement of the power piston has been just right to bring the speed back to normal, the flyballs and the pilot piston will return to their normal positions. Thus the compensating needle valve now allows the oil from the sump to flow to the receiving compensating plunger which gradually returns the entire mechanism, except the power piston, to its normal position, and the prime mover speed will again be back to normal.

This hydraulic governor is *isochronous*, that is, does not allow the speed of the engine to droop with increase of load. Mechanically connected governors must permit a droop in order to function.

23. Cooling.—In all internal-combustion engines some provision must be made for cooling those parts in contact with the hot gases. Excessive cylinder temperatures endanger the materials of the cylinder, piston, and valves, and failures might result. Also, at high temperatures, lubrication becomes very difficult, and in Otto engines preignition will occur.

Air-cooling is used on some automotive and airplane engines in which weight is an important consideration. When air-cooling is used, the cylinders are provided with fins or ribs which increase the outside radiating surface of the cylinder. A current of air is forced over these fins by the motion of the vehicle or by means of a blower. Automobile and large air-cooled stationary engines have the fins running parallel to the cylinder axis; in motorcycle and airplane motors the fins are transverse. The Wright engine of Fig. 11 is air-cooled.

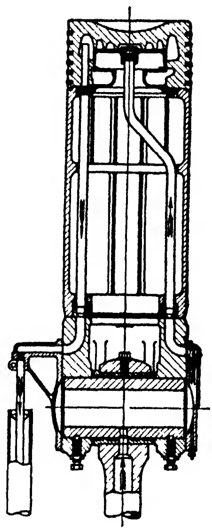


FIG. 32.—Nordberg piston cooling by lubricating oil.

Water-cooling is nearly universal for automobile, marine, and stationary engines. On large engines the exhaust valves and exhaust manifold are sometimes water-cooled. The engine of Fig. 6 has a jacketed manifold.

In double-acting engines the piston and piston rods have no external radiating surface, and these parts must be cooled by water or oil circulation. The piston of the Nordberg engine of Fig. 14 is cooled by the lubrication oil as illustrated in Fig. 32. The circulating pump delivers a large excess of oil to the crankshaft which is delivered through the hollow piston rod to the wrist pin, thence to the pistonhead and out through a telescoping pipe to an oil cooler.

24. Cylinder Arrangements.—There are several ways of locating the valves in the engine cylinder, six of which are illustrated in Fig. 33. The argument for the valve-in-head arrangement (sometimes called the "I-head") is that the shape of the cylinder head is simpler, and consequently the exposed wall area is smaller in proportion to the inclosed volume. This tends to decrease the heat loss from the working charge. On the other hand, the valve sizes are limited.

The L-head possesses the advantage of being able to use a very simple valve-operating mechanism. The T-head has a large cooling surface to the combustion chamber and is therefore less efficient than the other types.

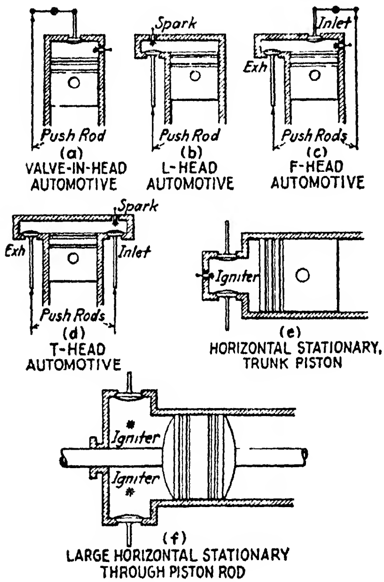


FIG. 33.—Combustion chamber and valve arrangements.

In horizontal engines having trunk pistons, the valves are frequently placed in a pocket extension on the cylinder head, as shown by *e*. In large double-acting engines having through piston rods, the valves are placed in pockets at the top and bottom or at the side of the cylinder at the ends.

25. Valves and Their Mechanisms.—Valves for small engines may be made of cast-iron heads cast or riveted on steel stems. The inlet valve, since it is cooled at each suction stroke by the incoming charge, can be made very satisfactorily of cast iron in most engines. The exhaust valves, not being so cooled, have a tendency to overheat, and in important engines it may be necessary to use alloy-steel exhaust valves or provide some means of cooling them, as explained in Art. 23. Valve heads are sometimes made laminated to insure proper seating.

For high-speed automobile and airplane engines the valve must be made as light as possible so that the springs may close them quickly. To this end, the valve heads are made very thin and of alloy steel. Modern automotive engines are fitted with inserted valve seats of wear- and heat-resisting alloys.

26. Diesel-injection Mechanisms.—The fuel-injection system of a Diesel engine must meet the following requirements:

1. The fuel must be introduced in such a manner and in such condition that ignition will begin promptly, proceed rapidly and continue to completion.

2. The fuel must be injected in an amount exactly suited to the load on the engine.

3. The fuel must be injected at the correct instant in the cycle and at the proper rate.

In large, slow-speed engines these requirements are relatively easily met, but in small, high-speed, multicylinder engines the accurate and reliable handling of the exceedingly small quantity of fuel required per cycle in a very short time interval presents a difficult problem.

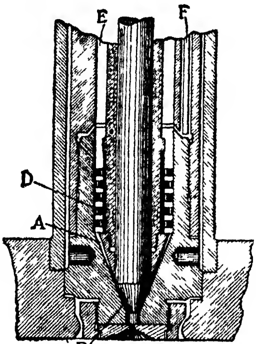


FIG. 34.—Closed-type (air injection) fuel spray valve

It is the function of the *injection nozzle* to meet the first requirement. Figure 34 shows the principle of the air-injection nozzle which was used by all the early Diesels but is now used only to meet special fuel requirements. The pump forces the fuel-oil charge into the space *E* through the passage *F* against the injection air pressure, which may be as high as 1,000 lb per sq in. At the proper instant the camshaft raises the conical valve *B*, and the fuel charge is forced through the holes in the atomizing disks *D*, through the fluted passages in the cone *A* and out of the spray nozzle *C* in a finely divided spray. The rate at which the fuel is introduced is controlled by the lift of the valve *B*, which, in turn, is controlled by the profile of the cam that raises it. The Nordberg engine of Fig. 14 uses this type of injection when running on low-grade oil or on natural gas.

Mechanical or *solid* injection nozzles are made in two types, those opened hydraulically by the oil pressure and those opened mechanically by a cam. The Bosch fuel-injection system, Fig. 35, employs a hydraulically operated nozzle *A*. Oil is delivered by a pump *B* (one pump for each cylinder) through the fuel duct *C* to the annular space *E*, above the needle valve *J* and below the valve plunger *D*. When the pressure reaches a predetermined valve, the plunger *D* is lifted against the pressure of spring *G* and the fuel charge sprayed into the cylinder through the nozzle orifice *F*. *H* is a leak-off drain.

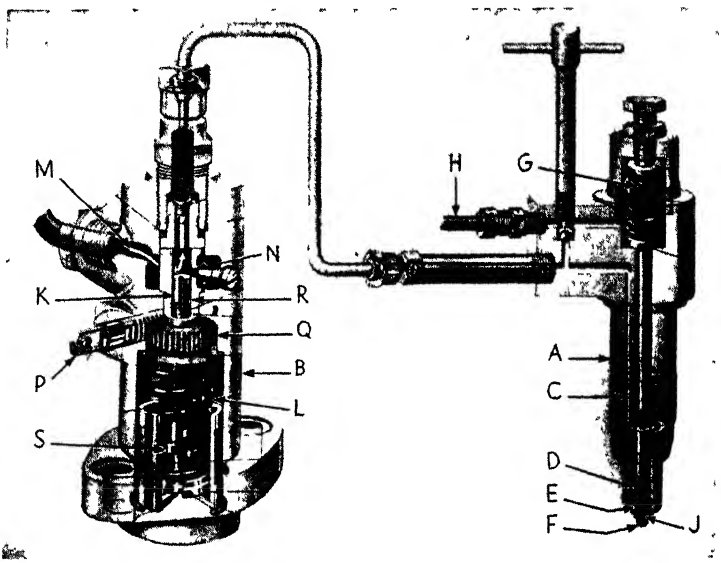


Fig 35 —Bosch Diesel fuel-injection system

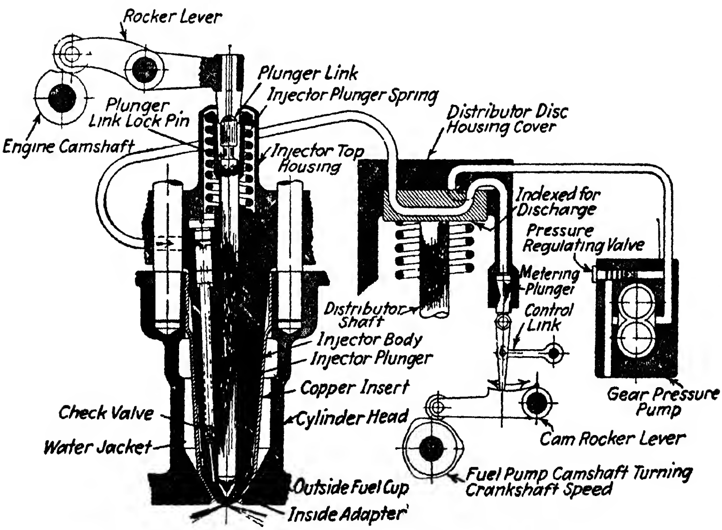


Fig. 36.—Cummins Diesel fuel-injection system.

The Cummins system, shown in Fig. 36, has a nozzle which is opened mechanically by the engine camshaft. Oil delivered by the gear pressure pump passes through a distributer, and a metered amount is sent by the metering plunger at about 100 lb pressure to the injector, forcing it past the check valve. The oil

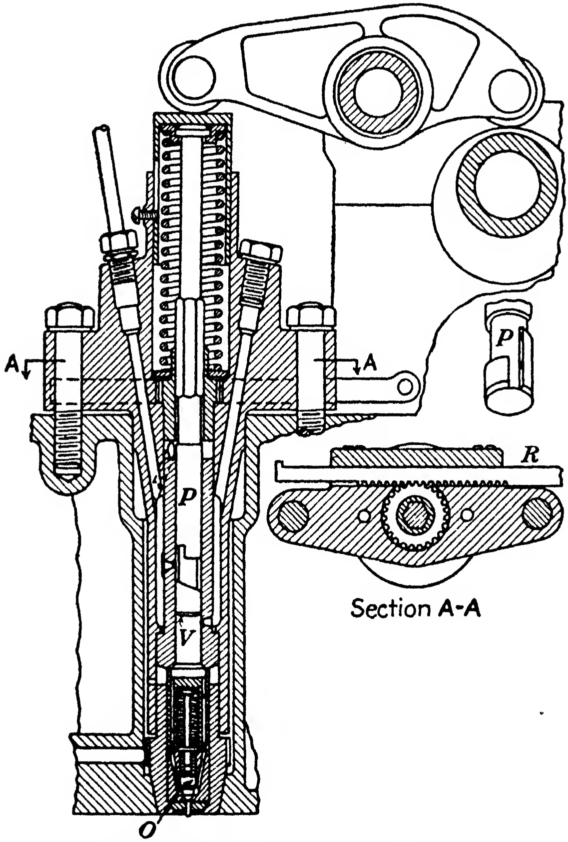


Fig. 37.—General Motors combined fuel-injection pump and valve.

flows into the conical cup beneath the injector plunger but, because of the cylinder pressure, does not flow into the cylinder through the open nozzle orifices. On the contrary, the hot, compressed air passes into the fuel cup thus vaporizing some of the charge, and a moment later the plunger is depressed thus forcing the charge into the cylinder.

Figure 37 shows a combined injection pump and nozzle of the hydraulically opened type used on the General Motors automotive Diesel.

Governing of Diesel engines is accomplished by metering the fuel charge at each cycle. The Bosch equipment of Fig. 35 has a pump B of the *constant-stroke* type operated by means of a camshaft. As the plunger K is raised against the pressure of spring L, fuel oil delivered by the supply pump to passage M is trapped above the plunger when it covers port N. As the plunger continues its upward movement, fuel is sent to the nozzle A until

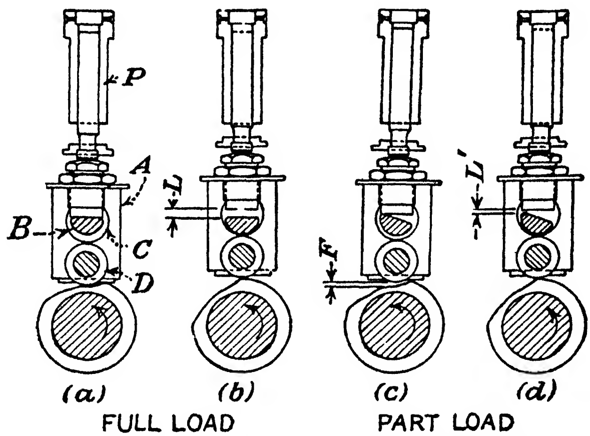


FIG. 38.—Deco variable stroke Diesel fuel-injection pump.

the spiral edge R of the plunger uncovers the port N. The governor link is connected to rack P which meshes with gear Q, which is in turn splined to the lower end of the plunger by the lug S. If part of the load is dropped, the governor pulls rack P to the left, turning edge R to the right, thus releasing the pressure and cutting off the fuel earlier.

The Cummins equipment, Fig. 36, is controlled by a *variable stroke* pump. As the cam rocker lever moves up and down through a regular angle, the governor moves the control link to the right or left, changing the lever arm applied to the metering plunger.

Figure 38 shows the variable stroke arrangement of the Deco pump used on the Allis-Chalmers oil engine of Fig. 18. The pump plunger tappet assembly A has a large hole C, through which a governor rod B of crescent cross section is passed and on which the tappet assembly rests as shown at (a) when roller D is at the low point of the cam. At (b) the roller is on the high point of the cam, and the pump plunger has a stroke L. At (c) for part load, the governor rod is turned on its axis so that the roller is held up from the cam by the amount F, and so the lift of the plunger is reduced to L' in (d).

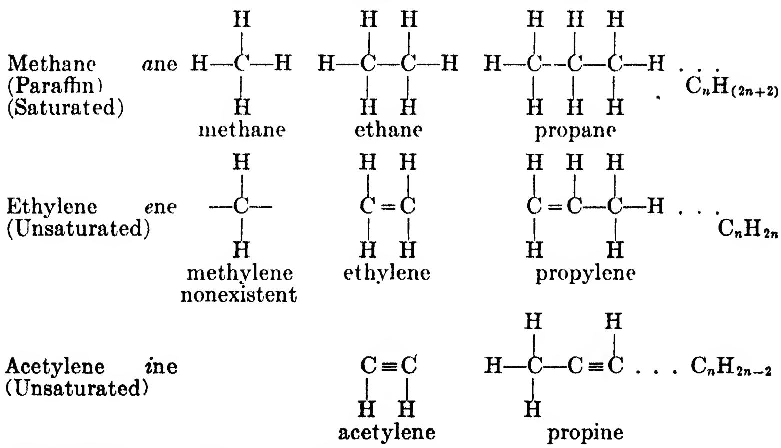
CHAPTER IV

FUELS FOR INTERNAL-COMBUSTION ENGINES

27. Petroleum.—Petroleum, or "rock oil," is a very complex mixture of hydrocarbons and in the crude state may contain some sulphur and other impurities. In American crudes these hydrocarbons are chiefly of the methane series along with some of the ethylene and naphthene series. Russian petroleum consists chiefly of naphthenes with small parts of methanes and ethylenes.[1]

The relative proportions of the members of these series vary greatly in the different fields, and these varying proportions account for the different characteristics of the oils. The American crudes are roughly divided into two classes, called "paraffin base" and "asphalt base," depending upon whether the residuum

[1] "ALIPHATIC" HYDROCARBON SERIES (Paraffins, acyclic compounds):



NAPHTHENES (Cyclic compounds C_nH_{2n}):
Cyclohexane C_6H_{12}-Hexamethylene

AROMATICS: Benzene C_6H_6 Toluene, methyl benzene $C_6H_5(CH_3)$ Xylene $C_6H_4(CH_3)_2$

For an excellent treatment and bibliography on this subject see Nelson "Petroleum Refinery Engineering," McGraw-Hill (1936) left after the oils are removed contains a larger proportion of paraffin or of asphaltic bitumen. All oils contain both of these materials, but, in general, Eastern oils are paraffin base, while Pacific and Gulf states oils are asphalt base, although all sections of the country contain deposits of both kinds.

Crude oil is "refined" by boiling in a closed still, the more volatile hydrocarbons passing off first, followed by the heavier distillates. Theoretically, we should have an infinite number of distillates, but, in practice, several groups of products are collected between certain overlapping temperature bands.

The first product to distill is the crude naphtha or benzine and may constitute 25 to 40 per cent of the volume of the crude oil. The distillation procedure is controlled by the density of the stream flowing from the still, the crude-naphtha cut being collected until the gravity drops to about 50° Bé.

The second cut is the kerosene distillate, which is collected until the gravity drops to about 40° Bé., and may constitute 15 to 20 per cent of the crude.

The next cut is the gas- or fuel-oil cut, which is collected until the gravity drops to about 36° Bé., and may constitute 10 to 15 per cent of the crude.

The remainder contains lubricating oils, waxes, and tars of higher molecular weight. By heating the heavier distillates under pressure to temperatures above their boiling points, "cracking" takes place. Cracking is a decomposition in which two lighter hydrocarbons are produced from a more complex one and is, in general, expressed by the formula

$$C_nH_{2n+2} \rightarrow C_{n-p}H_{2(n-p)} + C_pH_{2p+2}$$

28. Petroleum Products. *Gasoline.*—Gasolines on the American market are of three kinds:[1]

1. *Straight refinery gasoline* is produced by distillation from the crude petroleum in a fire still, the condensed product being collected until some predetermined density is reached. This "crude naphtha" is then acid refined and steam distilled. Several products of different volatility ranges may be produced, or the steam distillation may be for the purpose of separating the gasoline from the less volatile "bottoms." Straight refinery gasolines consist mainly of aliphatic hydrocarbons with, generally,

[1] DEAN, Motor Gasolines, *Bur. Mines, Tech. Paper* 166.

a low content of unsaturated and aromatic hydrocarbons. The distillation range is usually free from marked irregularities.

2. *Blended casing-head gasoline* is obtained by compression or absorption from natural gas and is too volatile for general use. It is blended with enough heavy naphtha to produce a mixture that can be handled safely. As a result of this blending, this gasoline contains considerable percentages of both high- and low-volatility constituents and a lack of the intermediate ones.

3. *"Cracked" and blended gasoline* differs from straight refinery and casing-head gasolines in having a larger amount of unsaturated and aromatic hydrocarbons.

Kerosene.—Kerosene (from *keros,* meaning "wax") is an American name for oils that were originally used for burning in lamps. It was at first produced from mineral wax, bituminous shale, and coal and is sometimes called "coal oil." At the present time, it is produced from petroleum distillation, and the processes of its production are exactly similar to those for gasoline. Kerosene is sold in various commercial grades from "water white" to "standard" with increasing slight color toward the "standard" grade.

Heavy Oils.—Crude oil is not used as a fuel for internal-combustion engines to any considerable extent, for the reason that it contains gasoline fractions which are valuable as automotive engine fuels and which do not greatly increase the value of the crude oil as an oil-engine fuel. The residue from the gasoline stills and lighter kerosene stills is, therefore, purified and marketed as fuel oil. Although this product may be called the "skim milk" of the petroleum industry, it has only about 5 per cent less heating value per pound than the more volatile fuels.

29. Natural Gas.—Natural gas, being generally high in heating value and low in inert-gas content, is an excellent fuel for internal-combustion engines. It has until recent years been available only in favored locations near the gas fields but the extension of the long pipe lines to industrial centers is increasing the importance of this fuel. Natural gas is chiefly methane with small percentages of ethane and usually some carbon dioxide and nitrogen. Propane, butane, and higher hydrocarbons may be removed at the well and marketed as liquid fuels.

30. Commercial Butane.—A gas-well product that has recently been placed on the market in the Western states is commercial "butane" which consists of about 80 per cent butane and 20 per cent propane by liquid volume. Owing to its high vapor pressure it must be handled in pressure tanks and delivered to the carburetor under pressure; for this reason it is considered too hazardous for general use.

31. Coal Gas.—The original method of making illuminating gas was by the distillation of the volatile matter from coal in retorts. Many of these volatile constituents condensed again to tars, but a considerable part of the volatile matter of the coal was altered into fixed gases which could be distributed for use. In some places this method of making coal gas or bench gas is still in use, the choice between this method and the "water-gas" process depending upon local conditions such as the coal available, etc. The composition of this coal gas varies with the coal available and with operating conditions. In general, it will contain from 40 to 50 per cent by volume H_2, 30 to 40 per cent CH_4, 4 to 10 per cent CO, with small percentages of CO_2, N_2, and heavier hydrocarbons such as C_2H_4, C_2H_6, etc. The heat value generally lies between 600 and 700 Btu per cu ft but may go outside that range.

32. Carbureted Water Gas.—The chief competitor of coal gas is carbureted water gas. The process of making this may be considered in two parts: (1) the manufacture of uncarbureted water gas, or the so-called "blue water gas," and (2) the enrichment or carbureting of this gas with oil vapors to give it the desired heating value and illuminating value. The blue-gas process consists in passing steam through a deep incandescent bed of fuel (usually, coke or anthracite coal). The steam is decomposed, freeing the hydrogen, while the oxygen combines with carbon to form carbon monoxide or carbon dioxide. If all the oxygen went into carbon monoxide, it would give a gas containing, by volume, one-half H_2 and one-half CO, according to the equation

$$H_2O + C = H_2 + CO$$

Both of these gases are combustible, and such a gas would have a heat value of 325 Btu per cu ft. It would burn with a blue, nonluminous flame; hence the name "blue gas." Because some carbon dioxide is unavoidably formed, the actual heat value is somewhat lower. After steam has been run through the fuel bed for a time, the latter will drop in temperature because of the heat absorbed in the reactions The generator must then be connected to the atmosphere and blown with air to restore the high temperature to the fuel bed. These two alternating parts of the process are called the "run" and the "blow."

33. Oil Gas.—Oil gas has been used for a number of years in those Pacific Coast states in which oil is a cheaper source of hydrocarbons than coal. It is made in an apparatus consisting of two vertical steel shells having a connection at the bottom which is partially filled with firebrick checkerwork similar to that used for carbureting water gas. The process is cyclic. For 10 to 15 min an oil burner at the top of one chamber heats the checkerwork to about 2000°F., the combustion products escaping from a stack on the second chamber. Then for 20 to 30 min oil is sprayed on the hot bricks and is thus gasified. At the end of the "run" the steam is introduced, thus generating a quantity of water gas in the product.

Since water gas came upon the market as a substitute for coal gas, it was necessary to give it a heat value and an illuminating value comparable with coal gas. This is done by adding to the blue gas a gas made by spraying oil upon a checkerwork of hot bricks in the carburetor. This gas adds CH_4, C_2H_4, and some heavier illuminants and reduces the proportions of the blue water gas in the resultant mixture. The carbureting is sufficient to give the resultant gas a heat value of at least 600 Btu per cu ft, so that for illuminating, heating, or power purposes it is equivalent to the coal gas made by the older process.

34. Producer Gas.—Producer gas has many desirable features as a gas for internal-combustion engines and is a very economical fuel for isolated power plants.

Usually, a gas producer and engine are installed as a unit, the suction of the engine drawing the air and steam into the producer. The gas is thus generated just as it is required by the engine, and no gas storage is necessary. The producer is supplied with fuel, coal, coke, or charcoal through a valve and hopper at the top. The green fuel drops into a distillation zone, where the volatile matter is distilled off, and the resulting coke gradually settles down into the incandescent fuel bed in the center of the producer. The gases from the volatile matter are drawn out of the producer while air is admitted at the bottom of the generator, whence it is drawn through the grate into the incandescent fuel bed. The oxygen of the air burns some of the carbon to carbon dioxide, but this carbon dioxide passes on in contact with more incandescent carbon and is largely reduced to carbon monoxide, which is a combustible gas. This carbon monoxide, with the nitrogen of the air, passes out and mingles with the gases from the distillation zone.

Under ideal, or theoretical, conditions, using only pure carbon for fuel and producing only CO, each cubic foot of air admitted would have 0.791 cu ft N_2 and 0.209 cu ft O_2. The 0.209 cu ft O_2 would, theoretically, be converted to 0.418 cu ft CO, thus giving a resultant gas of 0.791 cu ft N_2 and 0.418 cu ft CO. The resultant gas would have $0.791/(0.791 + 0.418) = 0.65$, or 65 per cent, N_2. The CO would constitute the remaining 35 per cent.

Actually, the gas contains some carbon dioxide which is not reduced in the producer. The gases from the volatile matter of the coal add materially to the combustible content of the gas. The richness of the gas is further improved by admitting with the air supply a limited amount of steam, which is decomposed by the heat of the fuel bed into hydrogen and carbon monoxide or carbon dioxide. This use of steam serves a number of purposes. It enriches the gas, as mentioned above; it serves as a control on the temperature of the fire; and it utilizes for its vaporization some of the sensible heat of the gases leaving the producer, the steam being generated in the water-jacketed head of the producer or in a separate waste-heat boiler.

A good average analysis of producer gas from hard coal would be: CO, 24 per cent; CO_2, 5 per cent; H_2, 17 per cent; N_2, 54 per cent. Heat value, 145 Btu per cu ft. Gas from soft coal will contain CH_4 and will have a higher heat value, perhaps 155 Btu.

35. Blast-furnace Gas.—The blast furnaces used to reduce iron ore in steel plants act as gas producers. The by-product, blast-furnace gas, contains less hydrogen than ordinary producer gas, because only a small amount of moisture is carried in with the charge and with the air and also because the fuel, coke, contains no volatile matter. The gas is very low in heating value because of its high nitrogen and carbon dioxide content.

The most serious problem in connection with blast-furnace gas as internal-combustion engine fuel is that of freeing it of the very large amount of dust carried along with it. At the present time the tendency is toward the utilization of this fuel gas under boilers for generating steam to operate turbogenerators. Blast-furnace gas is used as a source of heat and power only in the industry that produces it.

36. Coke-oven Gas.—Another by-product gaseous fuel of considerable importance is coke-oven gas. Coke is produced by heating coal in closed "ovens," the volatile gases that pass off being collected, cooled, washed, and freed of tar. About half of this gas is used for heating the ovens and the remainder is available for power purposes. Since this gas is rich in hydrogen and methane, it is high in heating value.

37. Sludge Gas.—In the process of digestion of sewage sludge, a gas is generated which has proved to be an excellent fuel for gas engines used to drive pumps and electric generators in the disposal plant. This gas, which consists of about two-thirds methane and one-third carbon dioxide has a heating value of 600 to 700 Btu per standard cubic foot. In some cases hydrogen sulphide may be present in quantities sufficient to cause corrosion due to formation of acid. Owing to its high inert-gas content, it is slow burning and may be used to advantage in high-compression engines.

38. Alcohols.—Methyl and ethyl alcohols have been extensively used in Europe as a motor fuel, but in America, owing to their relatively high cost compared with that of petroleum fuels, they have not been used to any appreciable extent. Alcohols have the advantage of being able to stand much higher compression pressures than petroleum products. Since the efficiency of the Otto cycle is in direct proportion to the compression ratio, Art. 77, the thermal efficiency of an engine using alcohol fuel is generally higher than when using petroleum fuels.

Alcohols have the disadvantages, however, of (1) high cost due to internal-revenue regulations, (2) the legal requirement that ethyl alcohol must be "denatured," (3) water content. Commercial alcohols contain 5 to 20 per cent or more of water, which not only reduces the heating value but also makes starting difficult. Carburetors for alcohols must have some provision for heating them in order to accomplish complete and rapid evaporation.

39. Comparison of Gas-engine Fuels.—The economic selection of an engine fuel depends upon its availability and relative cost.

Each fuel described in the foregoing paragraphs has advantages in a particular set of local condiions. Table II compares the physical properties of the several liquid and gaseous fuels in most general use.

Antiknock ratings of commercial gasolines are made on the basis of an "octane number" which denotes the percentage of iso-octane (2, 2, 4-trimethyl pentane) in a mixture of iso-octane and normal heptane which will give the same antiknock properties as the gasoline in question. The tests of the gasoline are made on a special knock-test engine, known as the C.F.R. Engine, developed for this purpose. Average grades of gasoline have octane ratings from 60 to 70 while premium grades have 78. It should be pointed out that in a particular engine, a high-octane gasoline is not necessarily more efficient than a low-octane gasoline, although it will make for smoother operation. High octane does, however, make the use of high compressions feasible which, in turn, improves efficiency. Detonation or "ping" in an Otto engine is now generally held to be due to a compression wave and its consequent high temperature wave which ignites the charge ahead of the ignition flame.

In Diesel engines detonation may occur due to ignition lag. The tiny droplets of the fuel spray entering the cylinder require a measurable time to reach the ignition temperature, and, when this temperature is reached, combustion may be so rapid as to produce detonation. The longer this delay, the more the accumulated fuel and the "rougher" the combustion. A Diesel fuel has been rated for ignition quality by "cetane," $C_{16}H_{34}$, number which denotes the cetane percentage in a blend of cetane and alphamethylnaphthalene, $C_{11}H_{10}$, having the same antiknock properties as the fuel oil in question.

TABLE II.—CHARACTERISTICS OF INTERNAL-COMBUSTION ENGINE FUELS

Fuel	Source	Typical composition: liquid fuels, per cent by weight; gas fuels, per cent by volume	Theoretical air required		Higher heating value of fuel,[1] Btu	Heat value of theoretical carbureted mixture[1]	Practical compression pressure	Inflammability limits, per cent in air-gas mixture
			Cu ft per cu ft fuel	Lb per lb fuel				
Commercial butane	Oil and gas wells	$20C_3H_8$, $80C_4H_{10}$ (liq. vol.)	31	.....	3,200	100	75–200	
Gasoline	Gas wells, petroleum	14.4H, 83.3C, $1.1O_2$, $1.1N_2$, 0.1S	.....	14.6	20,300	.....	90–120	
Kerosene	Petroleum	12.4H, 83.0C, $1.6O_2$, $2.8N_2$, 0.2S	.....	13.9	19,600	.....	70	
Fuel oil	Petroleum refineries	11.6H, 83.0C, $2.8O_2$, $1.4N_2$, 1.2S	.....	13.6	18,800	.....		
Methyl alcohol	Wood distillation	CH_3OH	7.23	6.46	9,770	100.5	120	6.0–36.5
Ethyl alcohol	Vegetable products	C_2H_5OH	14.32	8.99	12,760	101.4	200	3.5–19
Natural gas	Oil City, Pa.	$64.3CH_4$, $33.4C_2H_6$, $3.3N_2$	11.70	.....	1,240	97.7	100–115	4.2–13.6
Natural gas	Elk Hills, Calif.	$92.1CH_4$, $0.8C_2H_6$, $1.1N_2$, $6.0CO_2$	10.02	.....	937	85.0	100–115	
Natural gas	Dry Creek, Mont.	$81.4CH_4$, $15.6C_2H_6$, $3.0N_2$	10.35	.....	1,100	96.9	100–115	
Carbureted water gas	Coal enriched with oil gas	$35H_2$, 34CO, $15CH_4$, $13C_2H_4$, $2N_2$, $1CO_2$	19.69	.....	550	26.6	130	5.7–42.5
Oil gas	Low-grade petroleum oils	$54H_2$, 10CO, $27CH_4$, $3C_2H_4$, $3N_2$, $3CO_2$	19.19	.....	550	27.2	....	5.0–32.5
Producer gas	Bituminous coal	$12.8H_2$, 22.2CO, $3.2CH_4$, $0.2C_2H_4$, $55.6N_2$, $5.8CO_2$, $0.3O_2$	8.86	.....	140	14.2	100–165	0.5–2
Producer gas	Coke	$10H_2$, 30CO, $1.5CH_4$, $0.5O_2$, $54.5N_2$, $3.5CO_2$	10.73	.....	135	11.5	100–165	0.5–2
Blast-furnace gas	Iron making	$2H_2$, 28CO, $58N_2$, $12CO_2$	0.72	.....	95	55.0	150–200	0.5–2
Coke-oven gas	By-product coke ovens	$57.3H_2$, 5.9CO, $26.9CH_4$, $3.0C_2H_4$, $4.7N_2$, $1.4CO_2$, $0.2O_2$	4.48	.....	535	97.0	80–100	3–10
Sludge gas	Sewage disposal	$2H_2$, $65CH_4$, $3N_2$, $30CO_2$	6.7	.....	632	82.1	110–180	

[1] Per lb for liquid fuels; per cu ft for gas fuels and carbureted mixtures.

CHAPTER V

PRINCIPLES OF COMBUSTION

40. Chemistry of Combustion.—Combustion may be defined as a chemical reaction evolving heat. For engineering purposes the term is applied to the rapid combination of various substances with oxygen, accompanied by the production of heat. Combustion is complete when each of the combustible substances has united with all the oxygen with which it is capable of entering into combination.

The most important combustible elements for the commercial production of heat are carbon, hydrogen, and sulphur. Carbon and hydrogen may be in their free states or may be combined with each other to form hydrocarbons or may be in other combinations in more complex fuels.

According to the chemical law of combining weights, the elements combine in simple and constant proportions to form definite compounds, and these proportions are in simple ratio to the molecular weights of the substances uniting. Table III gives the atomic and molecular weights of the substances commonly encountered in combustion calculations:

Applying the law of combining weights to the combustion reactions and using the molecular weights of Table III, if we multiply each term in a combustion equation by its molecular weight, we shall have the formulas in terms of *weight* instead of numbers of molecules. For example:

$$\begin{array}{lll} C & + O_2 & = CO_2 \\ 12 & \quad 32 & \quad 44 \\ \hline 12 \text{ lb } C & + 32 \text{ lb } O_2 & = 44 \text{ lb } CO_2 \end{array}$$

Stated in words this means that 12 lb carbon will unite with 32 lb oxygen to form 44 lb carbon dioxide.

Likewise, the weight formula for the combustion of hydrogen is

$$\begin{array}{lll} 2H_2 & + O_2 & = 2H_2O \\ \quad 2 & \quad 32 & \quad 18 \\ \hline 4 \text{ lb } H_2 & + 32 \text{ lb } O_2 & = 36 \text{ lb } H_2O \end{array}$$

TABLE III — DATA ON ELEMENTARY FUEL SUBSTANCES

Substance	Molecular formula	Molecular weight	Volume at 60°F and 30 0 in Hg: Cu ft per lb	Volume at 60°F and 30 0 in Hg: Cu ft per mole	Boiling point at 1 atm °F	Heat of combustion[1] Btu: Per cu ft	Heat of combustion[1] Btu: Per lb	Air ratio for perfect combustion: By volume	Air ratio for perfect combustion: By weight	Inflammability limits per cent fuel in mixture
Carbon monoxide	CO	28 01	13 51	378 2	−310	322	4 345 *G*	2 38	2 47	12 5–75
Hydrogen	H_2	2 016	187 9	378 8	−422	325 8	61 010 *G*	2 403	34 60	4 1 75
Methane	CH_4	16 04	23 56	377 8	−263	1 013	23 890 *G*	9 56	17 24	5 3 14
Ethane	C_2H_6	30 07	12 45	379 9	−119	1 793	22 330 *G*	16 88	16 09	3 2 12 5
Propane	C_3H_8	44 09	8 36	368 3	− 13	2 591	21 670 *G*	24 50	15 67	2 3 9 5
Butane nor	C_4H_{10}	58 12	6 32	367 0	+ 33	3 373	21 310 *G*	31 97	15 46	1 9– 8 4
Butane iso	C_4H_{10}	58 12	6 32	367 0	+ 1	3 367	21 280 *G*	31 97	15 46	1 8– 8 4
Pentane	C_5H_{12}	72 15	5 25	378 1	+100	4 030	21 080 *G*	38 12	15 33	1 4– 8
Hexane	C_6H_{14}	86 17	4 40	378 5	+156	4 730	20 800 *L*	45 28	15 25	1 3–
Ethylene	C_2H_4	28 05	13 40	375 6	−155	1 615	21 640 *G*	14 40	14 78	3 2–34
Propylene	C_3H_6	42 08	9 00	378 5	− 53	2 227	20 040 *G*	21 44	14 78	2 2– 9 7
Acetylene	C_2H_2	26 03	14 40	374 7	−118	1 497	21 570 *G*	12 05	13 28	2 5–80
Hydrogen sulphide	H_2S	34 08	10 98	374 2	− 79	637	6 990 *G*	7 23	6 08	4 3–45 5
Heptane	C_7H_{16}	100 20			+210		20 680 *L*		15 18	1 1–
Octane nor	C_8H_{18}	114 22			+258		20 600 *L*		15 13	1 0–
Octane iso	C_8H_{18}	114 22					*L*		15 13	
Cetane	$C_{16}H_{34}$	226 33					20 600 *L*		14 05	
Benzene	C_6H_6	78 09	4 85	378 5	+177	3 710	17 980 *G*	35 78	13 28	1 4– 8
Toluene	$C_6H_5CH_3$	92 13	4 11	378 4	+230	4 454	18 300 *G*	42 88	13 50	1 3– 7
Methyl alcohol	CH_3OH	32 04	11 81	378 1	+131	827	9 770 *L*	7 15	6 47	6 0–36 5
Ethyl alcohol	C_2H_5OH	46 07	8 21	378 0	+173	1 554	12 760 *L*	14 32	9 00	3 5–19
Carbon	C	12 01					*S*		11 51	
Sulphur	S_2	64 13					4 100 *S*		4 31	
Oxygen	O_2	32 00	11 82	378 0	−297					
Nitrogen	N_2	28 02	13 50	378 2	−321					
Air		28 97	13 06	378 2						
Carbon dioxide	CO_2	44 01	8 54	375 7	−109					
Sulphur dioxide	SO_2	64 07	5 77	369 8	+ 12					

[1] "Higher" heating value Chiefly from *J Research Natl Bur Standards* vols 3 6 7 8 12 15 18 19 *G* signifies for fuel in gaseous state to gaseous CO_2 and liquid H_2O *L* signifies for fuel in liquid state to gaseous CO_2 and liquid H_2O

which indicates that 4 lb hydrogen will unite with 32 lb oxygen to form 36 lb water.

The Mole.—The two formulas in the preceding paragraph suggest a very useful unit of measure of the substances entering into a chemical reaction.

A reaction equation indicates by its coefficients the relative numbers of molecules entering into the reaction. Thus, in

$$2H_2 + O_2 = 2H_2O$$

we see that 2 molecules of hydrogen plus 1 molecule oxygen give 2 molecules H_2O. To get this into pounds, we must multiply each coefficient by the molecular weight of its molecule. This expression may be shortened to *pounds-molecular weight* or *pound-mole* or simply *mole*. Thus the reaction equation above may be read: "Two moles hydrogen plus 1 mole oxygen gives 2 moles water." A "pound-mole" or "mole," then, means a number of pounds equal to the molecular weight of the substance.

In the case of the combustion of carbon to carbon dioxide, the reaction equation

$$C + O_2 = CO_2$$

may be read: "One mole carbon plus 1 mole oxygen gives 1 mole carbon dioxide." In pounds, then, $1 \times 12 = 12$ lb carbon plus $1 \times 32 = 32$ lb oxygen $= 1 \times 44 = 44$ lb carbon dioxide.

At 60°F. and atmospheric pressure of 30 in. (standard conditions for engineering calculations), 1 lb oxygen occupies a volume of 11.82 cu ft; 32 lb, or 1 mole, oxygen will occupy a volume of $32 \times 11.82 = 378.2$ cu ft.

At standard conditions, 1 lb carbon dioxide occupies a volume of 8.54 cu. ft.; 44 lb, or 1 mole, carbon dioxide will occupy a volume of $44 \times 8.54 = 375.7$ cu ft.

Likewise, 1 lb hydrogen at standard conditions occupies a volume of 187.9 cu ft; 2 lb, or 1 mole, hydrogen will occupy a volume of $2 \times 187.9 = 375.8$. If we use the more nearly correct molecular weight of hydrogen of 2.016, we should have as the volume of 1 mole hydrogen $2.016 \times 187.9 = 378.8$ cu ft.

In the same manner, the volumes per mole of some of the more common gases were computed and tabulated in column 5 of Table III. The column reveals the very interesting and useful fact that *the volumes of a pound-mole of all gases are very nearly 378 cu ft.*—nearly enough, in fact, for practically all engineering uses. This fact is readily explained if we assume that Avogadro's hypothesis is correct, namely, that "equal volumes of all gases under the same conditions of temperature and pressure contain the same number of molecules."

41. Air Required for Combustion.—Oxygen is the sole supporter of combustion, and our practical supply of oxygen is in the air. Atmospheric air is a physical mixture of oxygen, nitrogen, carbon dioxide, water vapor, argon, and traces of other inert gases. In combustion work the rare gases and carbon dioxide are considered as included in the nitrogen, since they are inert so far as the reactions are concerned

The proportions of the gases in the atmospheric mixture vary with both latitude and altitude, but the generally accepted values for dry air for the United States are as follows.[1]

Element	Volume per cent	Molecular weight	Product	Weight per cent
Oxygen	20 94	32 00	670 5	23 14
Nitrogen	78 08	28 02	2187 8	75 54
Argon	0 93	39 88	37 2	1 28
Carbon dioxide	0 03	44 00	1 3	0 04
Hydrogen	0 01	2 01	0 0	0 00
Neon, krypton, helium, ozone, zenon, others	0 01			
Dry air	100 00	28 97[1]	2896 8	100 00

[1] Virtual molecular weight

Or, for engineering combustion calculations:

	Volume per cent	Weight per cent
Oxygen	20 94	23 14
Nitrogen and other inerts	79 06	76 86

In order to supply 1 lb oxygen for combustion, it is necessary to supply 1 ÷ 0.2314 = 4.32 lb air, 3.32 lb of which is nitrogen.

[1] Humphreys, "Physics of the Air," McGraw-Hill Book Company, Inc., (1929), p. 76. Also *J. Amer. Chem. Soc.*, 59: 358, February, 1937.

Referring again to the reactions of Art. 40, we have, for hydrogen,

$$4 \text{ lb. } H_2 + 32 \text{ lb } O_2 = 36 \text{ lb } H_2O$$

or, dividing through by 4,

$$1 \text{ lb } H_2 + 8 \text{ lb } O_2 = 9 \text{ lb } H_2O$$

To supply these 8 lb oxygen for each pound of hydrogen, it is necessary to furnish $4.32 \times 8 = 34.56$ lb air.

Likewise, for carbon we have

$$12 \text{ lb } C + 32 \text{ lb } O_2 = 44 \text{ lb } CO_2$$

or, dividing through by 12,

$$1 \text{ lb } C + {}^{32}\!/_{12} \text{ lb } O_2 = {}^{44}\!/_{12} \text{ lb } CO_2$$
$$1 \text{ lb } C + 2\tfrac{2}{3} \text{ lb } O_2 = 3\tfrac{2}{3} \text{ lb } CO_2$$

To supply these $2\frac{2}{3}$ lb oxygen for each pound of carbon, it is necessary to furnish $4.32 \times 2\frac{2}{3} = 11.52$ lb air.

For sulphur, we have the relation

$$32 \text{ lb } S + 32 \text{ lb } O_2 = 64 \text{ lb } SO_2$$

That is, 1 lb oxygen is required for each pound of sulphur, and to supply this oxygen there are required 4.32 lb air.

In like manner, the quantities of air for the hydrocarbon fuels may be determined. For example, for methane we have the reaction

$$CH_4 + 2O_2 = CO_2 + 2H_2O$$

in molecules or volume. By weight we have

$$(12 + 4) \text{ lb } CH_4 + (2 \times 32) \text{ lb } O_2 = 44 \text{ lb } CO_2 + 36 \text{ lb } H_2O$$

or, dividing through by 16,

$$1 \text{ lb } CH_4 + 4 \text{ lb } O_2 = {}^{44}\!/_{16} \text{ lb } CO_2 + {}^{36}\!/_{16} \text{ lb } H_2O$$

To supply these 4 lb oxygen for each pound of methane, it is necessary to furnish $4.32 \times 4 = 17.28$ lb. air.

The tenth column of Table III gives the calculated weight of air theoretically required for the elementary fuel substances, using the accurate molecular weights of the third column instead of the approximate values as above.

It is sometimes convenient to state the composition of a fuel in terms of the percentages of each of the elementary fuel substances which it contains, such as carbon, hydrogen, or sulphur. For example, a gasoline may be described as having C per cent carbon and H per cent hydrogen. In this case the air theoretically required for combustion would be determined by multiplying the weight of each element per pound of gasoline by the amount of air required for the combustion of 1 lb of that element as found above. This gasoline would require

$$11.52 \times \frac{C}{100} + 34.56 \times \frac{H}{100}$$

pounds air per pound of gasoline.

In the case of those fuels, such as alcohol or coal, which contain oxygen in combination, a correction must be made in the amount of air required to take account of this oxygen. It is customary to assume that the oxygen in fuels is in combination with the hydrogen. Then, since 1 lb hydrogen unites with 8 lb oxygen, 1 lb hydrogen is withdrawn from the reaction for every 8 lb oxygen already in the fuel. Stated in another way, one-eighth of the weight of the oxygen in the fuel is deducted from the weight of the hydrogen. The weight of air required for a fuel containing oxygen is, then

$$11.52 \times \frac{C}{100} + 34.56 \frac{H - \frac{O}{8}}{100}$$

in which C, H, and O represent percentages by weight of these elements in the fuel.

For fuels containing sulphur, it is necessary to add to the above equation the quantity $1/0.2314$ ($= 4.32$) times the decimal fraction of sulphur, or $4.32(S/100)$, in which S represents the percentage of sulphur in the fuel. Then

$$\text{Air required} = 11.52 \times \frac{C}{100} + 34.56 \frac{H - \frac{O}{8}}{100} + 4.32 \frac{S}{100} \qquad (1)$$

A good thumb rule is that about 1 cu ft air is required for each 100 Btu "lower" heating value.

42. Heat of Combustion.—The heat quantities evolved during the combustion of the elementary fuels of Table III are given in column 8. These heat quantities are determined by burning a small quantity of the substance in a *fuel calorimeter*, one form of

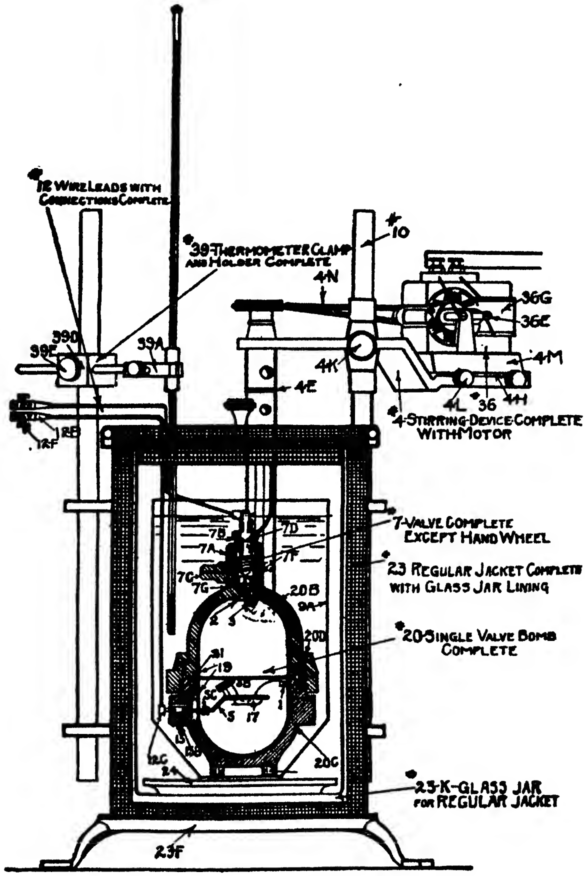


FIG. 39.—Emerson bomb calorimeter.

which is shown in Fig. 39. This instrument, which is adapted for solid and liquid fuels, is called a *bomb* calorimeter.

The bomb consists of two heavy steel cups joined by the steel nut 20*D*. The cups are lined with a nickel or platinum lining 19. The nickel or platinum fuel pan 17 is supported on an insulated-wire bracket 5 which is connected to the outside of the bomb by the electrode 12*C*. The fuse wire 5*B* is connected from the pan bracket across to the inside of the bomb at 1 and is looped down into the fuel in the pan.

The fuel (about 1 g) is weighed out very accurately, the bomb screwed together securely, and then the nipple *7C* is connected to a compressed-oxygen cylinder. Oxygen is carefully admitted through the valve *7D* until the pressure in the bomb is about 300 lb per sq in. A weighed quantity of water is placed in the calorimeter *9A* and the bomb set in the water. The thermometer and stirrer are immersed in the water.

Now, when the wire leads are connected in an electric circuit, the fuse wire *5B* will be heated and the fuel ignited. The combustion of the fuel in the bomb will raise the temperature of the bomb, water, and the calorimeter, the stirrer hastening the equalization of the temperature of the system.

The water equivalent of the apparatus being known and proper corrections being made in the temperature rise for radiation, the fusing of the ignition wire, etc., the heat absorbed by the calorimeter is readily computed. This heat quantity divided by the weight of the fuel burned will give the heating value of the fuel.

Liquid fuels may be tested in this calorimeter by the use of thin-walled glass bulbs as fuel containers.

For gaseous fuels a calorimeter of the continuous-flow type is used, both the fuel and the water flowing at a uniform rate. Such a gas calorimeter is shown in Fig. 40 in which *A* is the wet-gas meter, *B* the pressure governor, *C* the burner which, in use, is placed in the center of the water-jacketed chamber *D*. Cooling water from a gravity tank flows through the hose *E* maintaining a constant head by overflow *F*. Cooling water is weighed in the can *G* and its temperature rise measured by the thermometers T_1 and T_2.

The heating values of the elementary fuels of Table III have been very accurately determined by calorimetric measurements, and the generally accepted values are those given in column 8. The heating value of a commercial fuel may be determined directly by the calorimeter or may be determined approximately from the combined heating value of its elements if the analysis is known. Thus, if a fuel contains *C* per cent carbon, *H* per cent hydrogen, *O* per cent oxygen, and *S* per cent sulphur, its heating value would be approximately,

$$14{,}540\frac{C}{100} + 61{,}000\frac{H - \frac{O}{8}}{100} + 4{,}000\frac{S}{100} \qquad (2)$$

This is called Dulong's formula. It has been found that this formula does not agree exactly with the direct calorimetric test, because some of the elements are in states of combination that require heat to break down

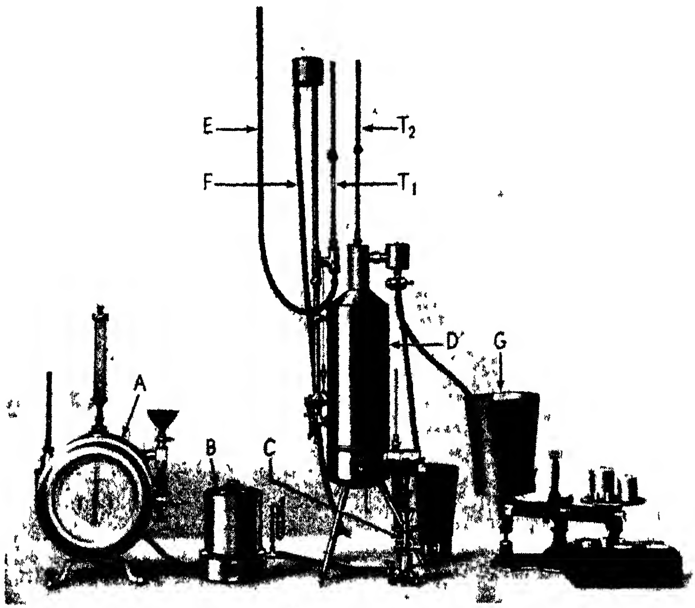


Fig. 40.—American Meter Company's gas calorimeter.

In a calorimeter the products of combustion are cooled to the original temperature of the fuel. Any H_2O formed from the combustion of hydrogen will thus be in the form of water, and the latent heat of this water will have been given up in the calorimeter. Values so determined are called the *higher heat values*, see note under Table III.

In many forms of commercial heat apparatus, such as gas engines or furnace chimneys, the water resulting from the combustion of hydrogen is ordinarily discharged in the form of superheated steam, and this reduces the effectiveness of the fuel by the amount of the latent heat of the steam plus the sensible heat of the steam and the water above the original temperature of the fuel. This heat loss is sometimes deducted from the calorimeter heat value, or the *higher heat value,* to obtain what is called the *lower heat value* of the fuel. Since this deduction varies greatly, depending upon the conditions under which the combustion take place, the "lower" heat value should not be used except for those particular cases in which the conditions are known. A blanket lower heat value should not be used for all cases.

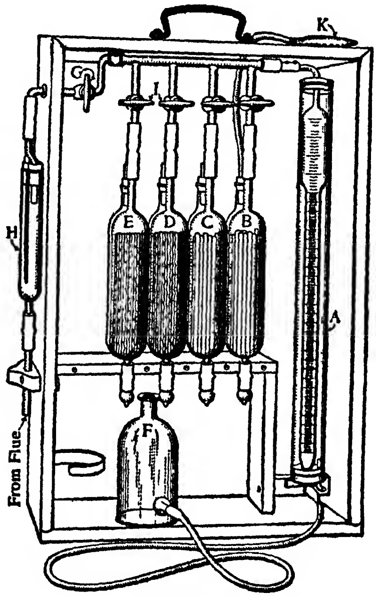


Fig. 41.—Orsat apparatus.

43. The Products of Combustion.—From Table III it will be observed that the products of combustion of the elementary fuel substances found in the engineering fuels are carbon dioxide, carbon monoxide, water, and sulphur dioxide. The quantities of these combustion products in terms of molecules per molecule of combustible, or in terms of moles per mole of combustible, were calculated in Art. 41. These calculations were made on the assumption that exactly the right amount of oxygen was present for the reaction, but in practice it has been found necessary to supply an excess of oxygen in order to make sure that each atom of fuel will be able to find its oxygen. This means that there will be some oxygen in addition to the combustion products mentioned above. Also, since air is always used in practical combustion work, we shall have nitrogen carried into the furnace with the oxygen and out with the combustion products.

The analysis of the combustion products is most commonly made by means of the Orsat apparatus shown in Fig. 41. The burette *A* is graduated in cubic centimeters up to 100 and is surrounded with a water jacket to prevent changes in temperature of the gas sample. Pipette *B* contains a solution of potassium hydroxide for the absorption of carbon dioxide, pipette *C* an alkaline solution of pyrogallol for the absorption of oxygen, and pipettes *D* and *E* an acid solution of cuprous chloride for the absorption of carbon monoxide. Each pipette contains a number of glass tubes which increase the liquid surface and facilitate the absorption of the gases. The solution in each pipette is drawn up to a point in the capillary tube near the cocks *I*. Before beginning a test, the water in the burette is brought to zero by opening cock *G* to atmosphere and adjusting the quantity of water in bottle *F*, taking care that the capillary tube at the top contains no obstructing plug of water or solution. The water in *F* should be kept slightly acid and an indicator added for evidence.

Gas is drawn from the flue through the U-tube *H* into the burette *A* by lowering the bottle *F*. Turning the three-way cock *G*, the gas may be discharged to the air by raising the bottle *F*. The gas sample to be analyzed is drawn in and the excess expelled through *G*, while the level of the water in *F* returns to the level of the zero mark in *A*. The cock *G* is then closed.

The gas is washed in the solution in pipette *B* for the removal of the carbon dioxide by opening the cock to *B* and raising and lowering the bottle *F* several times, finally returning the solution to the top of the pipette and closing the cock. The water level in bottle *F* is placed at the same height as that in the burette, and the volume of the remaining gas is read on the graduated scale. The shrinkage is the volume of carbon dioxide per 100 cc flue gas.

In like manner the gas is washed for the removal of oxygen in the pipette *C* and then for the removal of carbon monoxide in pipettes *D* and *E*. The process must be carried out in the order named, because the pyrogallol will also absorb carbon dioxide and the cuprous chloride will also absorb oxygen. After washing in pipettes *D* and *E*, the sample is returned to pipette *B* to remove traces of carbon dioxide liberated from the solutions, before final measurement in *A*. It is customary to assume that the gas now remaining in *A* is nitrogen, but this assumption may be far from correct when analyzing exhaust gases from Otto engines or flue gases from oil or gas-fired furnaces. Many investigators have found, especially in cases of insufficient air, that intermediate combustion products as well as free hydrogen, which are not absorbed by the Orsat reagents, may be present in these gases.[1]

The water vapor from combustion of hydrogen will be absorbed by the water in the burette, so that the Orsat analysis may be considered as the composition of "dry flue gases." Any sulphur dioxide in the gas will also be absorbed by the water, but this is usually negligible in quantity.

44. Calculations Based on Analysis of Combustion Products.—The calculations of Art 41 for the theoretical amount of air required for combustion may be reversed, and, knowing the composition of the flue gases, we may determine the quantity of air that was actually used for combustion. For this purpose a very useful relation is employed, namely, that *the volume percentages of a gas analysis give also the molal composition directly.* For example, the percentages by volume of nitrogen and oxygen in air are 79.1 and 20.9, respectively, and, since the mole is a unit of volume for gases (378 cu ft at 60°F. and 30 in.), 100 moles air would consist of 79.1 moles nitrogen and 20.9 moles oxygen.

The method of making the calculations for air actually used in combustion will be shown by the following examples:

Example A.—Coke (considered pure carbon) is burned in a furnace and the flue gas is analyzed and found to consist of CO_2, 17.2 per cent; O_2, 3.7 per cent; and N_2, 79.1 per cent by volume. What was the percentage of excess air used?

Solution.—The diagram of Fig. 42 shows how the quantities of the substances involved in the reaction are related before and after passing through the furnace. Consider three steps in the calculation:

Step 1.—Beginning with 100 moles flue gas as the basis for the calculations, we have the quantities of CO_2, O_2, and N_2, as shown at the right of

[1] Interpretation of Exhaust Gas Analyses, *Ore. Eng. Exp. Sta., Bull.* 4, 1934; also A. H. Senner, *Mech. Eng.*, November, 1936, p. 705.

the diagram. Now, remembering that all of the 79.1 moles N_2 came from the air supplied for combustion, we know at once that there were 100 moles air supplied because there are 79.1 moles N_2 per 100 moles atmospheric air.

This step is called the "nitrogen balance" and is indicated as step 1 in the diagram.

Step 2.—Knowing that for each mole of CO_2, there must be 1 mole C (since there is 1 molecule C per molecule of CO_2), we deduce that the molal

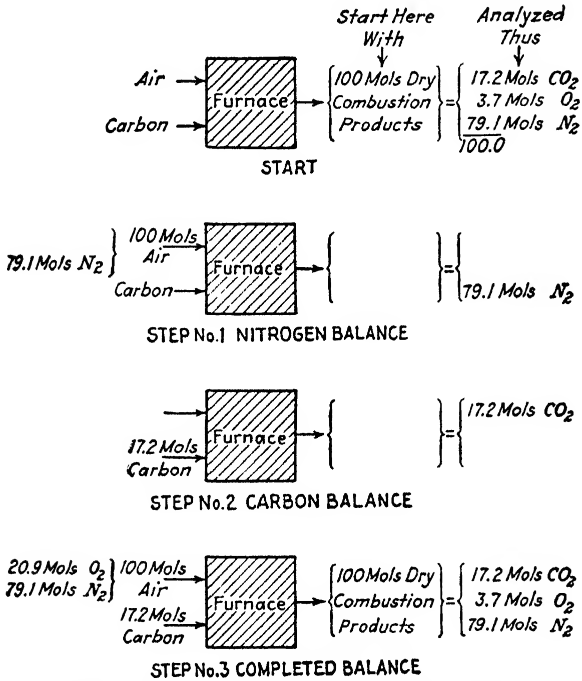


FIG. 42.—Example of combustion balance for coke.

quantity of C burned to produce 100 moles flue gas must be the same as the quantity of CO_2 in 100 moles flue gas, or 17.2 moles C. This step is called the "carbon balance" and is indicated as step 2 in the diagram.

Step 3.—Since 20.9 moles O_2 were supplied and 3.7 moles passed through the furnace unused, the fraction of excess oxygen (and, consequently, of excess air) was

$$\frac{3.7}{20.9 - 3.7} = \frac{3.7}{17.2} = 0.215 \text{ or } 21.5 \text{ per cent}$$

Example B.—Methane, CH_4, is burned in air, and the combustion products, analyzed by the Orsat apparatus, show 6.15 per cent CO_2, 9.94 per cent O_2, and 83.9 per cent N_2. (*a*) How many cubic feet of air were actually used per cubic foot of CH_4? (b) What was the percentage of excess air? (c) How many pounds of air per pound of CH_4?

Solution.—(a) The diagram of Fig. 43 shows the molal relations before and after passing through the furnace. It should be noted that the Orsat analysis gives the constituents of the dry flue gases only, the H_2O being condensed in the apparatus. Consider three steps in the calculation:

Step 1. *Nitrogen Balance.*—Taking 100 moles dry flue gas at the basis for the calculation, we have 83.9 moles N_2 on the right side of the diagram.

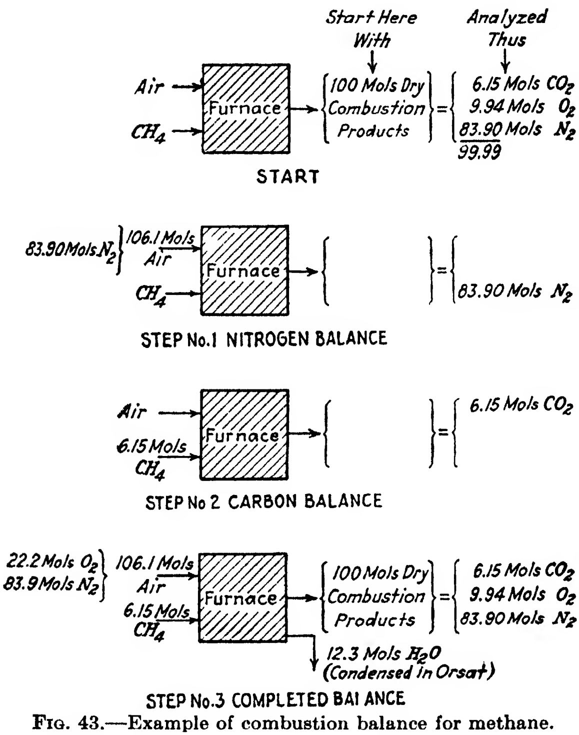


Fig. 43.—Example of combustion balance for methane.

Now, since these 83.9 moles N_2 come entirely from the air supplied for combustion which has 79.1 moles N_2 per 100 moles air, it follows that there were supplied $(83.9/79.1) \times 100$ moles $= 106.1$ moles air per 100 moles flue gas. That is,

$$\text{Moles air per mole flue gas} = 1.061.$$

This step is indicated as step 1 in the diagram.

Step 2. *Carbon Balance.*—Since CO_2 and CH_4 each contain 1 atom C, 1 mole CO_2 will be produced by 1 mole CH_4. The 6.15 moles CO_2 in the dry flue gas on the right side of the diagram were produced, therefore, by 6.15 moles CH_4 on the left side. This step is indicated as step 2 in the diagram.

Then the number of moles of CO_2, and therefore of CH_4, per mole of flue gas equals (6.15/100) = 0.0615. That is,

$$\text{Moles } CH_4 \text{ per mole flue gas} = 0.0615$$

Step 3.—Then the number of moles of air used per mole of fuel is represented by the quotient of these two ratios, that is,

$$\text{Moles air per mole } CH_4 = \frac{\text{moles air per mole flue gas}}{\text{moles } CH_4 \text{ per mole flue gas}}$$

or 1.061/0.0615 = 17.29. In other words, 17.29 cu ft air were used for the combustion of each cubic foot of CH_4.

(*b*) The 83.9 moles N_2 were accompanied by $(20.9/79.1) \times 83.9 = 22.2$ moles O_2, as shown on the left side of the diagram. Of these 22.2 moles O_2, 9.94 moles passed through the furnace unused, and the fraction of excess oxygen was

$$\frac{9.94}{22.2 - 9.94} = \frac{9.94}{12.26} = 0.811 \text{ or } 81.1 \text{ per cent}$$

It will be noticed that in this case, since the fuel contained no nitrogen, the excess air can be computed from Orsat analysis alone.

Another way to compute the excess air is as follows: Ideally, there are required 2 moles O_2 per mole CH_4, and this would require $(100/20.9) \times 2 = 9.56$ moles air. In other words, 9.56 cu ft air are ideally required for the combustion of 1 cu ft CH_4. Then, the percentage excess air used was

$$100\frac{17.29 - 9.56}{9.56} = 100\frac{7.73}{9.56} = 80.9 \text{ per cent}$$

The slight difference between these two results is due to slide-rule error.

(*c*) Since there were 17.29 moles air used per mole of CH_4, the **pounds** of air used per **pound** of CH_4 equals 17.29 times the ratio of molecular weights =

$$17.29 \times \frac{28.9}{16} = 31.2 \text{ lb air per pound } CH_4.$$

Another way to arrive at the same result is to multiply the theoretical weight of air per pound of CH_4 from Table III by one plus the fraction of excess air. That is,

$$17.29 \times 1.81 = 31.3 \text{ lb air per pound } CH_4$$

In the diagram it will be noted that 12.3 moles H_2O are produced by the 6.15 moles CH_4. This is determined by a hydrogen balance when the quantity of hydrogen in the fuel is known. In the case of fuels in which the percentage of hydrogen is not known, for example, coal or fuel oil, this percentage can be determined from the Orsat analysis, as follows:

In Example B it was calculated that there were 22.2 moles O_2 supplied of which 6.15 moles were used for combustion of C and 9.94 moles were excess oxygen. This left $22.2 - (6.15 + 9.94) = 6.11$ moles O_2 to unite with 12.22 moles H_2 to form 12.22

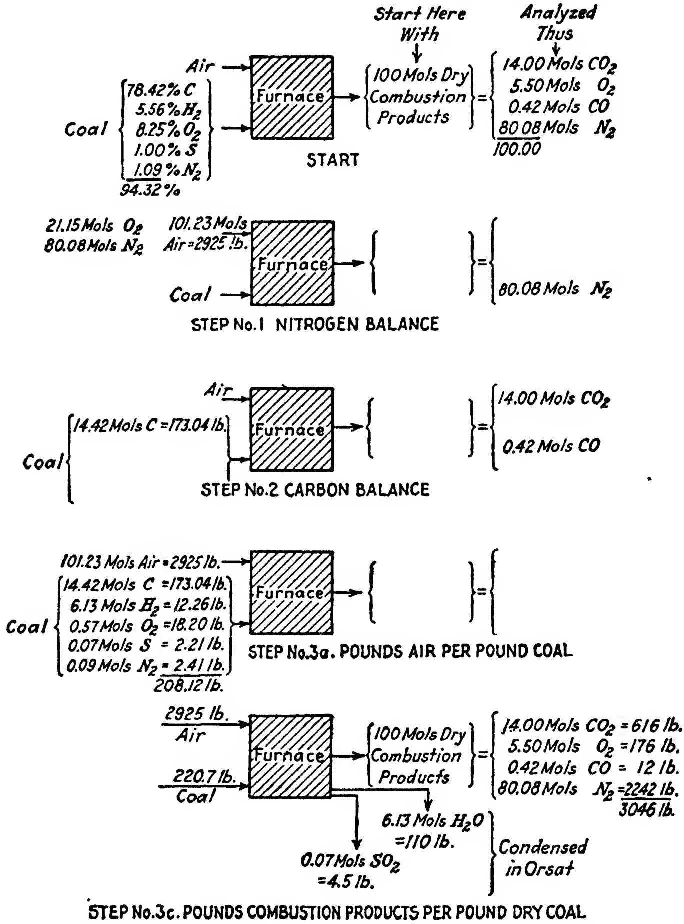


FIG. 44.—Example of combustion balance for a coal.

moles H_2O. This checks approximately the calculations from the quantity of CH_4.

Example C.—A coal has the following ultimate analysis: C, 78.42 per cent; H_2, 5.56 per cent; O_2, 8.25 per cent; N_2, 1.09 per cent; S, 1.0 per cent. This coal is burned in a furnace, and the flue-gas analysis by the Orsat is CO_2, 14.00 per cent; O_2, 5.5 per cent; CO, 0.42 per cent; and N_2, 80.08 per cent. (*a*) How many pounds of air were used per pound of fuel? (*b*) What was the percentage of excess air used? (*c*) What was the weight of each of the combustion products per pound of dry coal burned?

SOLUTION.—(*a*) Consider three steps in the calculation, referring to Fig. 44.

Step 1. *Nitrogen Balance.*—The 80.08 moles N_2 were supplied by the air, and, since air has 79.1 moles N_2 per 100 moles air, there must have been supplied $(80.08/79.1) \times 100 = 101.23$ moles or 2,925 lb air. These 101.23 moles carried $0.209 \times 101.23 = 21.15$ moles O_2. This step is indicated as step 1 in the diagram.

Step 2. *Carbon Balance.*—Since all the 14.42 moles C in CO_2 and CO of the flue gas came from the C of the fuel, we know there were 14.42 moles C in the coal consumed to produce 100 moles flue gas. This is equivalent to $14.42 \times 12 = 173.04$ lb carbon.

Step 3.—(*a*) The H_2 associated with this weight of C was (5.56/78.42) 173.04 = 12.26 lb or 6.13 moles; the associated O_2 was (8.25/78.42)173.04 = 18.20 lb or 0.57 mole; the associated S was (1.0/78.42)173.04 = 2.21 lb or 0.07 mole; and the associated N_2 was (1.09/78.42)173.04 = 2.41 lb or 0.09 mole. These quantities are shown on the left-hand side of Fig. 44 and indicated as step 3*a*.

The total weight of these coal substances, which constitute 94.32 per cent of the dry coal, was 208.12 lb, and the weight of dry coal burned was 208.12/0.9432 = 220.7 lb.

Then the weight of air supplied per pound of dry coal consumed is 2,925/220.7 = 13.25 lb. (*Ans. a.*)

(*b*) The theoretical quantity of air required is, by Eq. (1),

$$11.55(0.7842) + 34.64\left(0.0556 - \frac{0.0825}{8}\right) + 4.34 \times 0.01 = 9.05 + 0.16 + 0.04 = 9.25 \text{ lb air}$$

The excess is $13.25 - 9.25 = 4.00$ lb, and the percentage excess is $(4.00/9.25)100 = 43.3$ per cent. (*Ans. b.*)

(*c*) The *weights* of the several combustion products per 100 moles flue gas are obtained by multiplying the number of moles of each component by its molecular weight. In this manner we find we have 44(14.00) = 616 lb CO_2; 32(5.5) = 176 lb O_2; 28(0.42) = 11.8 lb CO; and 28(80.08) = 2,242 lb N_2.

Then the weights of the several components per pound of dry coal will be as follows:

616/220.7 = 2.79 lb CO_2; 176/220.7 = 0.798 lb O_2; 11.8/220.7 = 0.053 lb CO; and 2,242/220.7 = 10.65 lb N_2. (*Ans. c.*)

The number of moles of H_2O will be the same as the number of moles of H_2 that formed the water, namely, 6.13. The same will be true of the SO_2, and the number of moles of SO_2 will be 0.07. The *weight* of water formed per pound of coal will be nine times the weight of H_2, or 9(0.0556) = 0.5004 lb, and the *weight* of SO_2 per pound of coal will be twice the weight of S, or $2 \times 0.01 = 0.02$ lb.

As a check on the calculations, we may balance the oxygen on the two sides of the diagram. Thus on the left side we have 21.15 + 0.57 = 21.72 moles O_2. On the right side we have 14.00 + 5.50 + 0.21 + 3.065 + 0.07 = 22.85 moles O_2. The difference is due to error in the Orsat or fuel analyses or to the number of significant figures used in our calculations.

If it is desired to know the number of moles of each of the combustion products, per mole of fuel, we may divide the moles of each of the substances on the right-hand side of the diagrams by the number of moles of fuel. Thus in Example B we have 6.15/6.15 = 1 mole CO_2; 9.94/6.15 = 1.615 moles O_2; 8.39/6.15 = 13.64 moles N_2; and 12.3/61.5 = 2 moles H_2O per mole CH_4.

It will be noted that the "combustion balance" can be constructed almost completely if we know only the carbon content of the fuel. In most solid fuels the oxygen, nitrogen, and sulphur constituents are so small as to be negligible, certainly smaller than the usual error in the Orsat, and the hydrogen can be calculated by difference.

Further examples of the applications of the analysis of products of combustion will be shown in Chaps. IX and XX in connection with the performance of gas engines and boiler furnaces.

Appendix *A* shows a universal Orsat calculation chart which has been found expeditious and sure in the hands of persons who do this work only occasionally.

45. Ostwald's Combustion Triangle.—W. Ostwald has pointed out[1] that the relationship between the carbon dioxide, oxygen, and carbon monoxide measurements for a given fuel may be conveniently plotted within a triangle, the hypotenuse of which is determined by the hydrogen content of the fuel, the hypotenuse being the line of complete combustion. Figure 45 shows the triangles for four fuels with different carbon-hydrogen ratios, the triangle for methane having lines drawn parallel to the hypotenuse as a scale for CO percentages. On this diagram, for example, *AB* represents the Orsat per cent of CO, *A*8 that of O_2, and *A*3 that of CO_2, the excess air being 10 per cent. Ostwald's thought in designing the chart was that the only product of incomplete combustion was carbon monoxide, but, as was stated in Art. 43, free hydrogen and compounds containing hydrogen and carbon may be present as the excess air is reduced

[1] *Beitr. Graphischen Feuerungstechnik* Leipzig, 1920.

toward the theoretical amount, when burning hydrocarbons. With these fuels the use of the chart is limited to a small region near the complete combustion line. For combustion processes, such as coal furnaces or Diesel engines, which show no hydrogen combustibles in the flue gases, the chart serves as a valuable check on the manipulation of the Orsat.

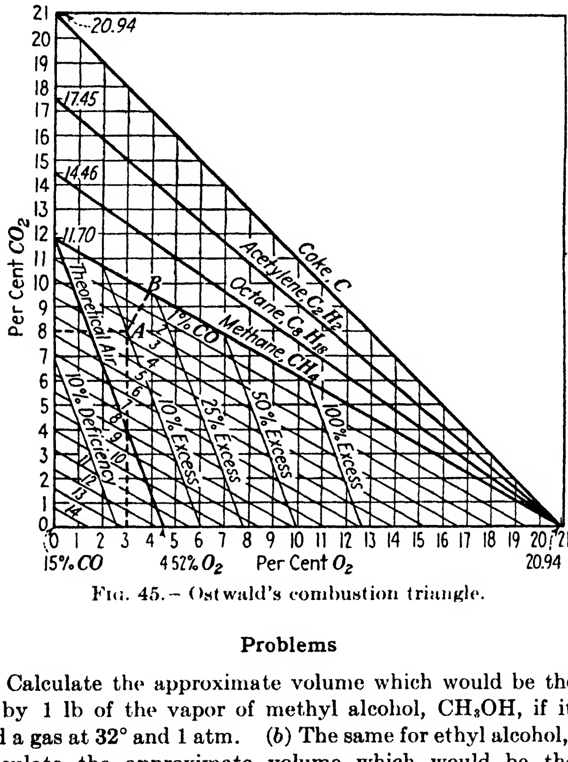


FIG. 45.—Ostwald's combustion triangle.

Problems

1. (*a*) Calculate the approximate volume which would be theoretically occupied by 1 lb of the vapor of methyl alcohol, CH_3OH, if it could be considered a gas at 32° and 1 atm. (*b*) The same for ethyl alcohol, C_2H_5OH.

2. Calculate the approximate volume which would be theoretically occupied by 1 lb gasoline vapor, considered as octane, C_8H_{18}, if it could exist as a gas at 32° and 1 atm.

3. A producer gas has the following percentage analysis by volume: H_2, 20; CO, 25; O_2, 0.5; CO_2, 5; N_2, 49.5. Calculate (*a*) the analysis by weight; (*b*) the specific volume at standard conditions (32° and 1 atm).

4. A natural gas has the following percentage analysis by volume: CH_4, 92.2; C_2H_4, 4.3; CO_2, 3.5. Calculate (*a*) the analysis by weight; (*b*) the specific volume at standard conditions.

5. A coke-oven gas has the following percentage analysis by volume: H_2, 53; CO, 6; CH_4, 35; C_2H_4, 2; CO_2, 2; N_2, 2. Calculate (*a*) the analysis by weight; (*b*) the specific volume at standard conditions.

6. Calculate the theoretical weight of air required for the combustion of (*a*) methyl alcohol; (*b*) ethyl alcohol.

7. Calculate (*a*) the theoretical weight of air required for the combustion of 1 lb octane; (*b*) the approximate heating value by Dulong's formula.

8. A fuel oil has the following percentage analysis by weight: C, 83.04; H_2, 11.58; O_2, 2.82; N_2, 0.6; S, 0.90. Calculate (*a*) the theoretical weight of air required for combustion; (*b*) the approximate heating value per pound.

9. A kerosene has the following percentage analysis by weight: C, 84.24; H_2, 13.44; O_2, 1.93; N_2, 0.2; S, 0.82. Calculate (*a*) the theoretical weight of air required for combustion; (*b*) the approximate heating value per pound.

10. A coal has the following percentage analysis by weight: C, 73; H_2, 7; O_2, 8. Calculate (*a*) the theoretical weight of air required for combustion; (*b*) the approximate heating value.

11. Calculate (*a*) the theoretical weight of air required for combustion per pound of the producer gas of Prob. 3; (*b*) the volume of air required per cubic foot of gas under standard conditions; (*c*) the heating value per pound; (*d*) the heating value per standard cubic foot.

12. Calculate (*a*) the theoretical weight of air required for combustion per pound of the natural gas of Prob. 4; (*b*) the volume of air required per cubic foot of gas under standard conditions; (*c*) the heating value per pound; (*d*) the heating value per standard cubic foot.

13. Calculate (*a*) the theoretical weight of air required for combustion per pound of the coke-oven gas of Prob. 5; (*b*) the volume of air required per cubic foot of gas under standard conditions; (*c*) the heating value per pound; (*d*) the heating value per standard cubic foot.

14. What would be the theoretical Orsat (volumetric) analysis of the products of perfect combustion of methyl alcohol in air?

15. What would be the theoretical Orsat (volumetric) analysis of the products of perfect combustion of ethyl alcohol in air?

16. What would be the theoretical Orsat (volumetric) analysis of the products of perfect combustion of methane in air?

17. What would be the theoretical volumetric analysis of the products of combustion of octane in air if 25 per cent excess air were used and no carbon monoxide were formed?

18. The coal of Prob. 10 is burned in air with 50 per cent excess air. What should be the Orsat analysis of the combustion products?

19. A fuel of oil having the percentage analysis: C, 83; H_2, 16; S, 1 is used in a Diesel engine, and the Orsat analysis of the exhaust gas shows the following percentages: CO_2, 7.2; O_2, 11.1; CO, 0.1; N_2, 81.6. Calculate (*a*) the weight of air theoretically required; (*b*) the weight of air actually used per pound of oil.

20. A coal containing the percentages: C, 80; H_2, 4; O_2, 2 is burned in a furnace, and the flue-gas analysis shows the percentages CO_2, 12; CO, 1; O_2, 7; N_2, 80 by volume. Calculate (*a*) the weight of air theoretically required; (*b*) the weight of air actually used per pound of coal.

21. A fuel oil contains 85 per cent carbon, and the Orsat analysis of the exhaust gases shows the percentages CO_2, 6; O_2, 13; N_2, 81. Calculate (*a*) the air actually used per pound of fuel; (*b*) the percentage of hydrogen in the oil.

22. The exhaust of a gasoline engine shows the following Orsat percentage analysis: CO_2, 11.56; O_2, 3.86; CO, 0.3; N_2, 84.3. Assuming the gasoline to be octane, what is (*a*) the weight of air used per pound of fuel; (*b*) the percentage of excess air used?

23. In Prob. 22, what is the weight of each of the combustion products, including water formed, per pound of gasoline?

24. Calculate the weight of each of the combustion products, including water formed, per pound of fuel oil of Prob. 21.

25. Calculate the weight of each of the combustion products, including water formed, per pound of coal of Prob. 20.

26. Ethane was burned in air, and the products of combustion analyzed by the Orsat gave CO_2, 10.77 per cent; O_2, 3.77 per cent; N_2, 85.48 per cent. What was (*a*) the percentage of excess air; (*b*) the weight of each of the combustion products, including water, per pound of fuel?

CHAPTER VI

WORK AND POWER

46. Definitions.—Work, in the physics sense, is the overcoming of resistance through a distance or

$$\text{Work} = \text{force} \times \text{distance}$$

In engineering, the usual unit of force is the *pound* and that of distance is the *foot*, and the product, *work*, is then expressed as *foot-pounds*.

Thus the work done in raising a weight is equal to the product of the weight times the vertical distance lifted, or $w \times l$.

In the case of water pumped into a standpipe, the work done equals the product of the weight of water pumped times the head in feet of the water pumped against, or $w \times h$. But since the weight of the water pumped equals the density of the water, d, in pounds per cubic foot times the volume pumped,

$$w = dV$$

and we have the work,

$$W = d \times V \times h$$

But since $d \times h$ equals pressure p, in pounds per square foot, we have

$$W = p \times V \qquad (3)$$

$p \times V$ then represents the work done on or by a fluid by a displacement of V cu ft against a pressure of p lb per sq ft and applies to gases and steam as well as to liquids.

Power is defined as the time rate of doing work, or

$$P = \frac{W}{t}$$

in which P = power in foot-pounds per second.
t = time in seconds.

The practical unit of power is the *horsepower* (hp), which equals 550 ft-lb per sec., or 33,000 ft-lb per min.

Thus the theoretical horsepower of a waterfall would equal

$$\text{Hp} = \frac{w_m \times h}{33{,}000}$$

in which w_m = the weight of water falling per minute.
h = the height of the fall.

In electrical terms the unit of power is the *watt*. One thousand watts is called the *kilowatt* (kw). One kilowatt equals 1.34 hp, or 1 hp equals 746 watts.

The power developed by an engine is measured by causing it to overcome resistance through a distance in some form of *brake* or *dynamometer*.

47. The Prony Brake.—Figure 46 shows a simple form of absorption dynamometer called the *Prony brake*. The driving pulley is fitted with two blocks A and B that may be clamped against its driving face with any desired force by means of the hand nuts C and D. The upper block has an arm E projecting out to the stand F which rests on the platform scales.

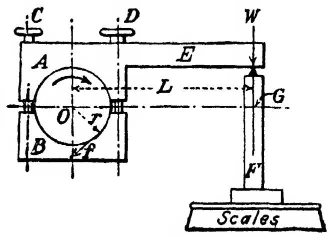


Fig. 46.—Prony brake.

When the engine runs in the direction of the arrow, the friction f of the pulley against the block produces a downward force W on the scales at a lever-arm distance OG represented by L. The work done per revolution by the engine equals the frictional force f times the circumference of the pulley, or $(2\pi r)f$. But, by the principle of the lever, $LW = rf$, and the work per revolution equals $2\pi LW$, which is the same as if a friction W were applied at the rim of a pulley of radius L. The work per minute equals $2\pi LWN$, where N represents revolutions per minute.

$$\text{Bhp} = \frac{2\pi LWN}{33{,}000} \tag{4}$$

and the constant part of Eq. (4), $2\pi L/33{,}000$, is called the "brake constant."

The "dead weight" of the stand F and the unbalanced part of the arm E must be deducted from the scale reading to obtain the true force W due to the power of the engine. This dead weight is measured by turning the engine over steadily by hand with the blocks loosened, first in the direction of the arrow and reading the scales and then in the opposite direction and reading the scales. The dead, or "tare," weight is the mean of these two readings.

Since the work done by the engine is converted into heat, some provision must be made for carrying away this heat, and for this purpose the rim of the pulley may be flanged inward and the space inside kept supplied with water.

For engines having flywheels or pulleys of large diameter, a more satisfactory form of brake is that shown in Fig. 47. This consists of a doubled length of large rope wrapped around the wheel, with the ends secured to the bottom of a frame F and the loop in the middle held by the screw hook H. By tightening up the hand nut N, the tension on the rope may be adjusted and the friction regulated. The length of the brake arm equals the radius of the wheel plus the radius of the rope.

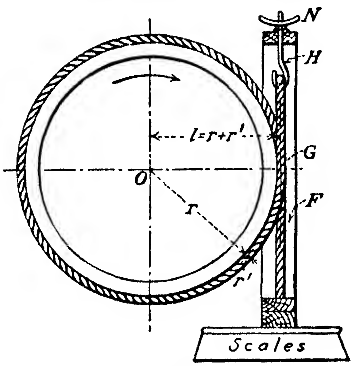


Fig. 47.—Rope brake.

48. Water Brakes.—Figure 48 shows a vertical section of a hydraulic dynamometer,[1] a type that is particularly adapted for use with high-speed gas engines or steam turbines. The shaft A carries a disk B having radial vanes C on its sides. The casing D likewise has radial vanes E and as the disk rotates with the shaft, water, which is admitted near the hub, is thrown by centrifugal action to the outer rim F of the casing, and the churning action set up tends to drag the casing around with the rotor. The torque necessary to hold the casing stationary is measured by a weight arm or a spring balance. As the water heats up due to absorption of the energy of the engine, a thermostat on the outlet at the bottom controls the temperature.

The entire dynamometer is supported on two pedestals with roller or ball bearings. To correct for the drag of these bearings the frictional torque of which would otherwise reduce the torque of the engine, a counterweight can be adjusted so as to give a

[1] See *Mech. Eng.*, October, 1937, p. 749, for a discussion of the performance of this type of dynamometer.

forward moment just equal to the frictional moment of the bearings.

49. The Electric Dynamometer.—Another type of brake especially adapted to high speeds and accurate regulation and, therefore, to automobile and airplane engines is the electric dynamometer. This is essentially an electric generator the frame of which is cradled so that the torque exerted upon the frame by

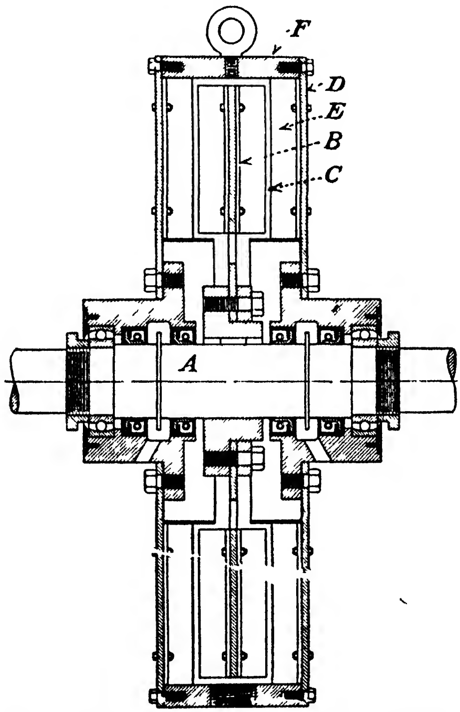


FIG. 48.—Hydraulic absorption dynamometer.

the armature current may be measured. The torque and consequently the load on the engine may be varied by changing the electrical output of the generator. The same consideration must be given to the friction of the cradle bearings as in the hydraulic brake.

Spring balances are frequently used instead of platform scales with the hydraulic and electric dynamometers, and for small powers the brake arm itself is sometimes graduated as a scale beam.

An electric generator that has been calibrated for friction and iron losses and has known electrical resistances provides a convenient method for loading an engine, since the generator output is easily measured by electrical instruments.

50. Pressures inside an Engine Cylinder.—For engineering purposes it is often important to know the pressure existing in an engine cylinder at various points of the cycle. Thus a pressure gage is sometimes screwed into the spark-plug hole of a gas engine to measure the "compression" pressure attained while the engine is being cranked. We know that the maximum pressure

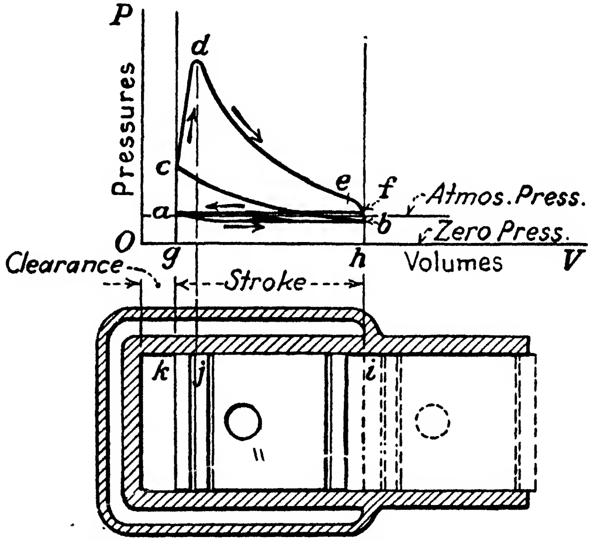


FIG. 49.—Pressures inside a gas-engine cylinder.

indicated by the gage occurs when the piston is at the end of the compression stroke, but the instrument does not tell the piston position corresponding to any other gage reading.

Figure 49 is a diagram of the cylinder of a gas engine having above it a graph in which the vertical distances *OP* represent gas pressure within the cylinder and the horizontal distances *OV* represent piston positions and therefore volumes occupied by the gas. At the beginning of the suction stroke the pressure in the combustion chamber is just about atmospheric, that is, at point *a* on the atmospheric line. When the piston moves outward, the pressure drops slightly, owing to the friction through the inlet passages, and the suction-pressure line is represented by *ab* slightly below the atmosphere.

The compression stroke is represented by the curved line *bc*, which rises at first gradually and then more rapidly as the piston nears the head end of the cylinder.

At *c* the spark occurs and combustion of the fuel takes place, the pressure rising, not instantly along a vertical line, but along some sloping line *cd* because the combustion requires an appreciable time, during which time the piston moves from *k* to *j*. The pressure at *d* may be as much as 500 lb per sq in. and depends upon the kind of fuel, the mixture richness, and the pressure at *c*. As the piston continues the power stroke, the pressure falls at first rapidly and then more slowly along the curve *de*.

The exhaust valve is opened at *e*, the pressure drops nearly to atmosphere, and then it is exhausted along the line *fa*, completing the cycle.

51. The Engine Indicator.—The instrument by means of which the graph of Fig. 49 is made is called an *engine indicator*. The indicator was originally devised by James Watt for use on a steam engine and is frequently called the steam-engine indicator, although it is used with equal success on displacement machinery of many kinds, such as gas engines, air compressors, ammonia compressors, and pumps.

In Fig. 50 is shown an engine indicator of a type particularly adapted for high-speed, internal-combustion engine work. By means of suitable piping, a connection is made by the union 7 to the clearance space at the end of the cylinder. The pressure within the indicator cylinder beneath the small piston 9 will, therefore, vary the same as the pressure in the engine cylinder. The vertical motion of the piston is controlled by a calibrated spring *E* at the top of the instrument. An increase in engine pressure will cause the piston 9 to rise and deflect the spring. This motion is communicated through link 18 to the pencil arm 20. The pencil arm is restrained by the links 17, 19, and 24, so that the motion of the piston is multiplied into a larger vertical, straight-line motion of the pencil point 52 at the end of the arm. This point will, therefore, move up and down in direct proportion to the changes of pressure within the engine cylinder. The controlling linkage is a carefully designed straight-line motion to cause the pencil point 52 to travel in a vertical straight line and to keep the motion of 52 in a constant ratio to the motion of the piston 9.

The drum *B*, carrying a sheet of paper on its outer surface, is caused to revolve in direct proportion to the movement of the

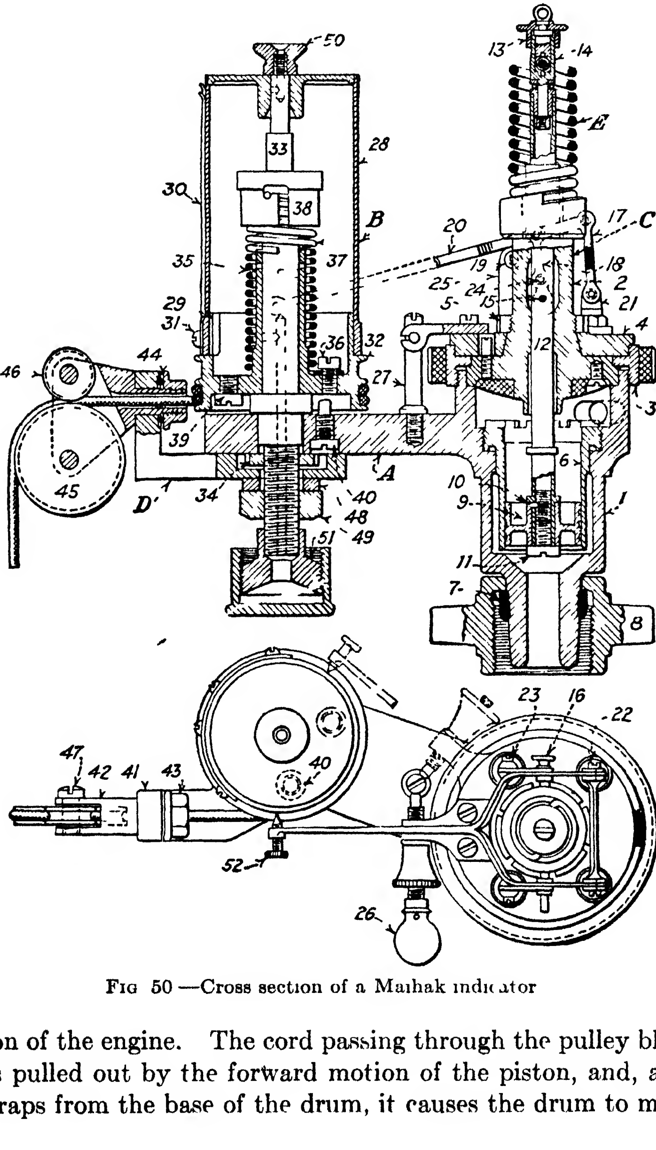


FIG 50.—Cross section of a Maihak indicator

piston of the engine. The cord passing through the pulley block 45 is pulled out by the forward motion of the piston, and, as it unwraps from the base of the drum, it causes the drum to make the greater part of a revolution. As the engine piston returns, the spring 37 inside the drum keeps the cord taut and causes the drum to return in proportion to the return stroke of the piston. We thus have the vertical motion of the pencil corresponding to the engine pressures, while the paper on the drum is being moved to correspond to the motion of the engine piston. The diagram drawn by the contact of the pencil against the paper will show,

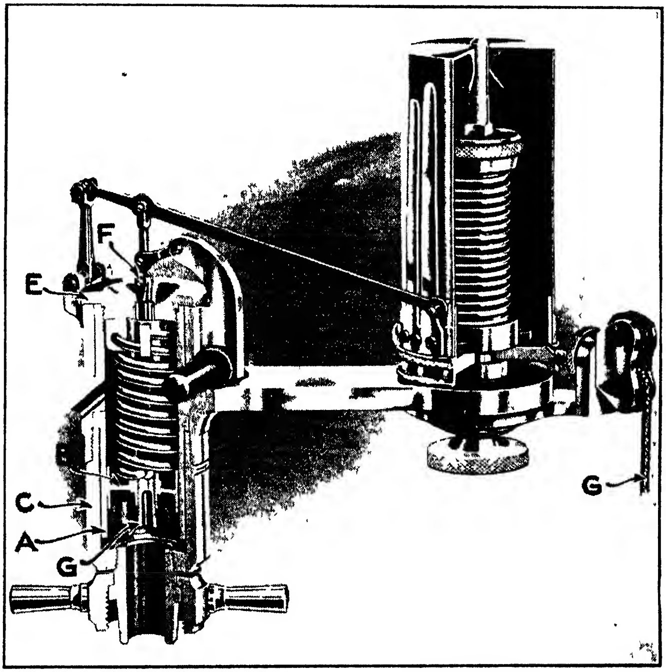


FIG. 51.– Crosby inside-spring engine indicator.

therefore, the pressure within the engine cylinder for each point in the forward and return strokes of the piston, and a diagram similar to that in Fig. 49 will be drawn.

The indicator springs *E* may be changed for different engine pressures. They are carefully calibrated and marked to show the pressure per square inch which will cause a motion of 1 in. of the pencil point. Thus an "80-lb" spring will be one which will permit the pencil point to move vertically 1 in. for a change in engine pressure of 80 lb per sq. in. The spring used should have a scale such that the height of the diagram is not over 2 in. and for high speeds not over 1½ in.

Figure 51 shows a Crosby indicator having the spring inside the instrument directly above the piston. Indicators of this type have the disadvantage that the heat from the working substance of the engine has a tendency to affect the elastic properties

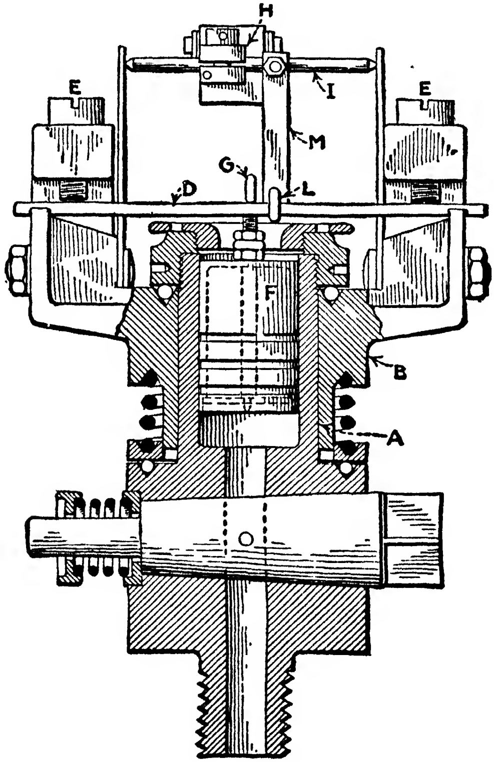


FIG. 52.—Hopkinson optical indicator.

of the spring and thus change its calibration. For high steam temperatures and for internal-combustion engines or in cases where great accuracy is desired, the outside-spring type should be used. Most makers will furnish either type.

For engine speeds over 600 rpm and for very sudden changes in pressure it is difficult to obtain reliable results from the ordinary form of indicator, owing to the inertia effects of the pencil mechanism and the paper drum. Figure 52 shows a "manograph" type of indicator that is especially designed for high-speed gas engines. The body of the instrument A which carries the cock is screwed into the cylinder. Part B is capable of rotating about the vertical axis, rolling on the ball bearings at top and bottom, the tension of the large coil spring eliminating lost motion. The indicator cord transmitting the piston motion is wound around the cylindrical part at B. The cylinder pressure lifts the piston F against the flat spring D, which lifts the

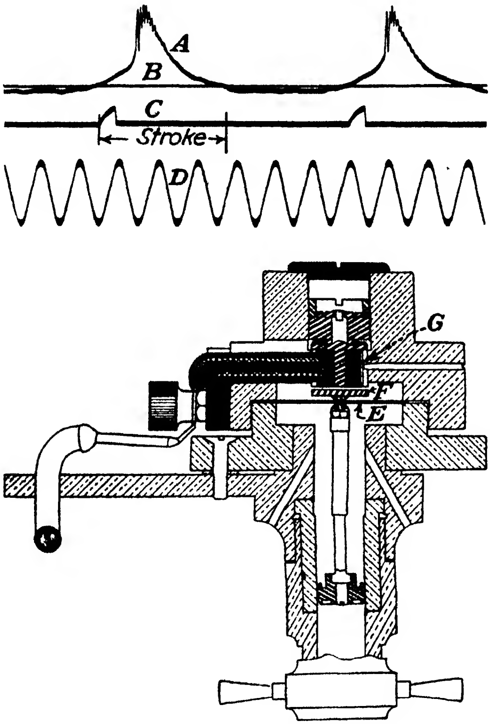


Fig. 53.—High-speed engine indicator and pressure-time cards.

strut M, thus turning the spindle E on its pivot points. H is a mirror holder, and it will be seen that a beam of light from a lamp, falling on the mirror, will be moved vertically by the pressure and horizontally by the piston motion. The card can be made on a ground glass for observation or on a photographic plate for recording.

Figure 53 shows a type of indicator[1] that is used for very high-speed engines. The spring E is a flat disk of high natural

[1] *Mech. Eng.*, May, 1936, p. 283.

frequency and on top of the disk is an iron armature F. An electromagnet G is placed above F with a small air gap between the two, the coil being connected through an a-c bridge circuit to an oscillograph. Now variations in pressure under the piston cause the air gap to change, which in turn produces variations in the reactance of the coil and thus the deflections are proportional to the pressure.

The photographic film on which the record is made moves at a uniform speed through the oscillograph and receives a record of the a-c timing wave D and the ignition line C. Since the only

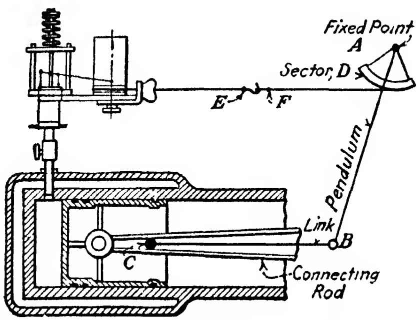


FIG. 54.—Pendulum-and-sector reducing motion.

mechanical motion in this instrument is that of the diaphragm, which is only a few thousandths of an inch, it is especially useful in a study of explosion ripples as shown on the pressure line A.

52. Reducing Motions.—The drums B, Fig. 50, are of such size that the length of the diagram drawn is only about 3 in. It is therefore necessary to have an accurate reducing motion between the piston of the engine and the drum, so designed that the motion of the drum bears a constant relation to the motion of the piston. Figure 54 shows such a reducing motion attached to an engine. It consists of a pendulum arm AB swinging about point A and having its lower end connected to the piston, connecting rod, or crosshead of the engine by the link BC. At its upper end it has a sector D whose center is A, and from this sector a cord is run to the indicator drum. This type, when properly proportioned, gives a nearly perfect reduction in the ratio of BA to DA. The string has a hook E and a loop F, so that the drum may be connected or disconnected while the engine is in motion.

A simple reducing motion is shown in Fig. 55. A small crankpin G is set into the end of the engine crankshaft Q and connected to a sliding rod HK by means of a connecting rod GH whose length is to the length of the engine connecting rod as QG is to the length of the main crank. A cord fastened to K is led over pulleys to the indicator drum. One objection to this motion is the long cord necessary, which introduces error due to stretching

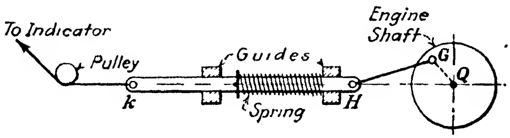


FIG. 55 Proportional-crank reducing motion.

Figure 56 shows a geared reducing wheel which is supported directly by the indicator.

A fourth type of reducing motion is a pantograph or "lazy tongs," one end of which is fixed, the other being connected to the crosshead of the engine. The drum motion is taken off at some point between the two ends that will give the desired ratio

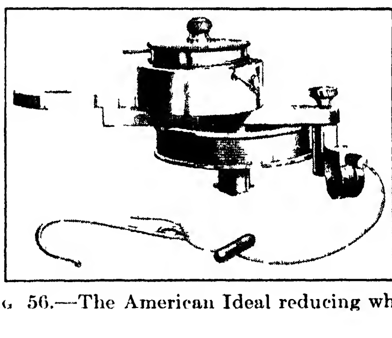


FIG. 56.—The American Ideal reducing wheel.

For low-speed steam engines the drum motion may be as much as 4 in., while for high-speed engines the motion should not be over $2\frac{1}{2}$ in.

53. Manipulation of the Indicator.—The pencil motion should be lubricated with fine oil, and the piston with the kind of oil used for the engine piston—cylinder oil for steam engines and gas-engine oil for gas engines.

The mechanism should be tested for freedom by having the spring removed and allowing the piston to drop by its own weight, but there should be no lost motion. The method of putting in the spring should be carefully studied, for it is frequently necessary to change springs after a test has been started and when the instrument is hot.

In attaching the instrument to the engine, the pipe coupling should be screwed down snugly so that it will not shift, and the string should be adjusted so that the drum will not strike the stops at either end of the stroke. Long connections between the engine and the indicator should be made with cord that will not stretch or with fine wire or cable.

The pencil point should be sharpened frequently, so as to make a fine line with a very light pressure on the paper. Generally, only one complete cycle should be traced on each card in order to keep the lines clear.

For accurate work the indicator spring should be calibrated at the temperature at which it works on the engine.

54. Significance of the Card Area.—It was shown in Art 46, Eq. (3), that the product of pressure by volume equals work. Now the height of an indicator card represents pressure, and the length represents distance traveled by the engine piston, which in turn represents volume swept through. The area of the indicator diagram is, therefore, proportioned to pV and, consequently, to the work done during the cycle.

The average pressure inside the engine cylinder that is effective in doing work is measured by the average ordinate of the indicator card multiplied by the scale of the spring. This *mean effective pressure,* as it is called, is the average pressure difference between the working and back strokes. The mean ordinate is found by dividing the area of the card by its length.

55. Measurement of Indicator Cards.—The usual method for determining the area of irregular figures is to use a planimeter, a simple form of which is shown in Fig. 57. The indicator card is tacked to a board covered with a fresh sheet of drawing or letter paper. The planimeter is anchored on the board so that the arms are roughly at right angles to each other and the tracing point placed at some easily identified or marked starting point. The dial and vernier are read and the reading recorded on the indicator card. The tracing point is then traced around the entire cycle in a clockwise direction and returned to the starting point, when the reading is again taken and recorded above the first reading. The cycle should then be retraced in the counter-clockwise direction, when the vernier should read the same as at the beginning. For ordinary work the discrepancy between the first and third readings should not exceed two places on the vernier and in case of greater error the card should be retraced. The difference between the mean of the first and third readings

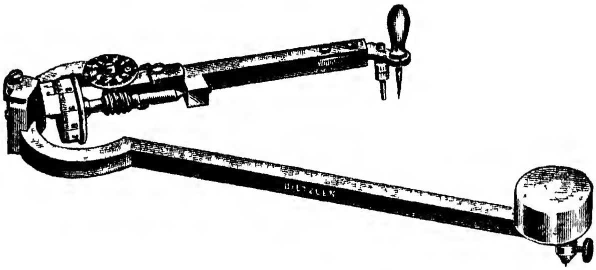


FIG. 57.—Dietzgen polar planimeter

and the second reading is taken as the area of the card in square inches.

If there are loops in the diagram, as is the case with an ordinary gas-engine card, these will represent **negative work,** that is, work done by the piston on the substance in the cylinder. These areas are automatically subtracted from the positive area of the card

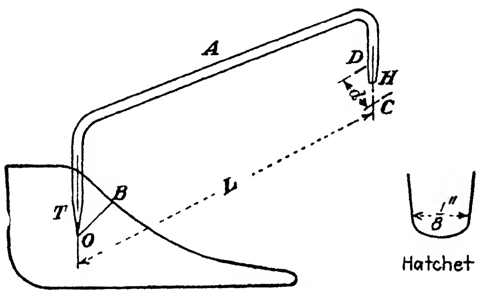


FIG. 58.—Hatchet planimeter.

by the planimeter, since they are traced in the opposite direction to that of the main area.

A simple instrument for measuring areas is the "hatchet" planimeter shown in Fig. 58. It consists of a piece of steel wire A about 7 in. long with about 1½ in. of the ends bent down to form the tracing point T and the hatchet H. The hatchet is formed as shown in the enlarged detail and is ground on an oil stone to a sharp cutting edge. The tracing point T must lie in the plane of the hatchet. To measure an area of any diagram, find, approximately, the center of gravity O of the area and draw any line OB to the boundary. With the planimeter set up as in Fig. 58, make a dent in the paper at H, and then trace the point from O to B and around the area in the right-hand direction to B and then return to O. Make another dent with the hatchet, which has arrived at some point C. Return the hatchet to its starting dent H, trace the card in the left-hand direction, and make a dent at D. The area of the card is 98 per cent of the distance d between C and D times one-half L. Care must be used to keep the instrument upright and to measure CD with great accuracy.[1]

56. The Trapezoid Method for Mean Effective Pressure.—If a planimeter is not available, the mean ordinate may be determined with reasonable accuracy by the trapezoid method. Limiting lines at the ends of the diagram are first drawn perpendicular to the atmospheric line. The length of the card is then divided into, say, 10 equal parts by means of a scale, and the area is divided into 10 trapezoids. Of course, it is not necessary to draw more than the middle ordinate of the trapezoids, and these are erected at the scale readings of $\frac{1}{2}$, $1\frac{1}{2}$, $2\frac{1}{2}$, etc. The sum of these middle ordinates divided by 10 and multiplied by the scale of the spring gives the mean effective pressure with sufficient accuracy for most calculations. Greater accuracy would result from using more than 10 trapezoids.

57. Indicated Horsepower.—Multiplying the mean effective pressure from an indicator diagram by the area of the corresponding face of the piston will give the total mean pressure, which may be considered as acting upon that side of the piston during the working stroke. This product times the length of the stroke, **in feet,** will give the foot-pounds of work done. This product, in turn, multiplied by the number of working strokes per minute will give the work done per minute, in foot-pounds.

This may be expressed by the following formula:

$$W = p_m LAN$$

[1] See *Amer. Mach.*, July 17, 1902.

in which W = the number of foot-pounds of work per minute.
p_m = the mean effective pressure in pounds per square inch.
L = the length of stroke in feet.
A = the area of the piston in square inches.
N = the number of working strokes per minute (equals the revolutions per minute in a single-acting steam engine and the number of explosions per minute in a gas engine).

The arrangement of the factors to spell the word *plan* makes the formula easy to remember. The fact that the area must be expressed in square inches while the stroke must be expressed in feet should be particularly noted.

Since 1 hp is a rate of 33,000 ft-lb per min, it follows that the indicated horsepower of each end of a cylinder is obtained by the use of the formula.

$$\text{Ihp} = \frac{p_m L A N}{33{,}000} \tag{5}$$

The total indicated horsepower is, of course, the sum of the values obtained for the two ends of the cylinder of a double-acting engine. In the crank end of a double-acting engine the area A subject to steam pressure is less than at the head end by the cross-sectional area of the piston rod.

Example A.—A 16 by 36-in. steam engine is running at 110 rpm. The piston rod is 3 in. in diameter. The indicator cards show mean effective pressures of 48 and 46 lb for the head and crank ends, respectively. What is the indicated horsepower of the engine?

NOTE: The meaning of 16 by 36 in. is that the cylinder bore is 16 in. and the stroke 36 in.

SOLUTION:

$$\text{Ihp (head end)} = \frac{48 \times 36/12 \times 16^2 \times 0.7854 \times 110}{33{,}000} = 96.5$$

$$\text{Ihp (crank end)} = \frac{46 \times 36/12 \times (16^2 - 3^2) \times 0.7854 \times 110}{33{,}000} = 89.2$$

$$\text{Total Ihp} = 185.7$$

Example B.—A four-cylinder, four-stroke automobile engine has a $3\frac{3}{4}$-in. bore and 4-in. stroke. When running at 1,000 rpm and having a mean effective pressure of 67 lb per sq in., what indicated horsepower is it developing?

SOLUTION:

$$\text{Ihp} = \frac{67 \times \frac{4}{12} \times 3.75^2 \times 0.7854 \times \left(\frac{1{,}000}{2} \times 4\right)}{33{,}000} = 14.95$$

58. Friction Horsepower.—The power lost in an engine, as shown by the difference between indicated horsepower and brake horsepower, is evidently the power consumed in overcoming the friction of the moving parts of the engine. It is generally determined by finding the difference between the indicated and brake horsepowers.

$$\text{Fhp} = \text{ihp} - \text{bhp}$$

If an engine is operated without any dynamometer or other load, the indicated horsepower developed will all be used in

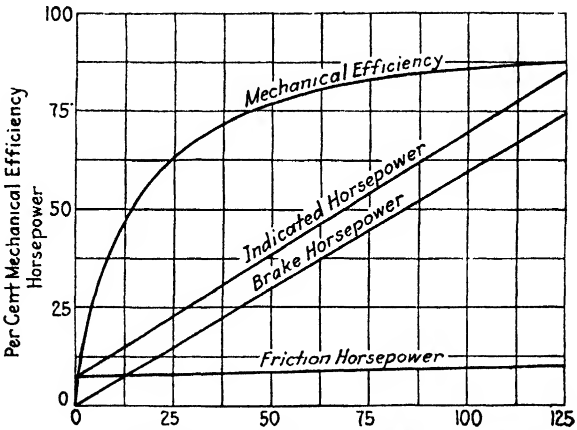


FIG. 59.—Horsepower and mechanical efficiency curves for a simple steam engine.

friction. We can, therefore, determine the friction horsepower at no load by determining the indicated horsepower under this condition. The friction horsepower will be greater under load, because of the greater forces transmitted and the resultant greater bearing pressures. The only reliable way of determining friction horsepowers under load is by the difference between indicated and brake horsepowers.

59. Mechanical Efficiency.—The mechanical efficiency of any piece of power machinery is the ratio of output to input. In an engine, it is obviously the ratio of brake horsepower to indicated horsepower.

$$\text{Mechanical efficiency} = \frac{\text{bhp}}{\text{ihp}}$$

At no load, the mechanical efficiency of any engine is clearly zero, for there is no output. At full, or rated, load, the mechanical efficiency of a steam engine should be at least 85 per cent and for a well-built engine may reach 95 per cent. In a gas or gasoline engine, the full-load mechanical efficiency may vary from 75 to 90 per cent in the same way.

Figure 59 shows for a steam engine of 60 rated bhp, the curves of brake horsepower, indicated horsepower, friction horsepower, and mechanical efficiency for different percentages of rated-horsepower output.

Problems

(All steam engines double acting.)

1. What is the potential horsepower of the Boulder project on the Colorado River at 550 ft head and 20,000 cu ft water per second? The same in kilowatts?

2. A pump delivers 10,000 lb water per hour to a boiler against a pressure of 175 lb per sq in. What is the net horsepower of the pump?

3. The lever arm of a Prony brake is 8 ft long, and during an engine test the net brake load was 1,035 lb. Engine speed, 150 rpm. What was (*a*) the brake horsepower; (*b*) the "brake constant?"

4. A Prony brake with an arm 6 ft long is placed on an engine. Engine speed, 200 rpm; gross load on scales, 205 lb. When the engine was turned over by hand forward and then backward, the scale readings were 47 and 29 lb respectively. What was the horsepower developed during the engine test?

5. An engine has a flywheel 52 in. in diameter and is fitted with a rope brake made of 1½-in. rope. Tare weight of brake is 42 lb. During a test, the gross load was 343 lb and the speed was 290 rpm. (*a*) What is the "brake constant?" (*b*) What was the horsepower developed during the test?

6. What must be the length of a Prony-brake arm to give a brake constant of 0.001?

7. A steam turbine is fitted with a hydraulic brake for a test, the arm being 38 in. and the tare weight 191 lb. During the test, the speed was 3,025 rpm, and the gross brake load was 1,887 lb. Calculate the horsepower developed.

8. An electric motor, direct connected to a fan, is "cradled" and has a torque arm 24 in. long. During a test, the net brake load was 2.73 lb and the speed 1,774 rpm. What was the power input to the fan?

9. A 20,000-kw generator has an over-all efficiency of 94.7 per cent. What must be the shaft horsepower of the turbine that drives it?

10. A direct-current generator is direct connected to a steam turbine. The load on the generator was 59.4 amp at 249 volts. The shunt-field current (not flowing through the load ammeter) was 0.47 amp. The resistance of armature, brushes, and series field is 0.149 ohm, and a calibration test shows the stray power loss (windage, friction, and iron losses) to be 1,250 watts. What was the output of the turbine in kilowatts; in horsepower?

11. A 400-kw alternator is direct connected to a steam engine. The over-all efficiency of the alternator is 91 per cent at full load, and the mechanical efficiency of the engine is 89 per cent. What is the indicated horsepower of the engine?

12. A single-cylinder, four-stroke Diesel engine gave an indicator card 3.45 in. long having an area of 0.96 sq in. Scale of spring 353 lb per in. (*a*) What was the mean effective pressure? (*b*) The engine dimensions were 17 by 19 in. and the speed 255 rpm. What was the indicated horsepower?

13. The average mean effective pressure of a 21 by 37-in., single-tandem, double-acting, four-stroke gas engine running at 131.3 rpm was 70.3 lb per sq in. Diameter of piston rod and tail rod 5 in. Mechanical efficiency 89 per cent. What were the indicated and shaft horsepowers?

14. A four-cylinder, double-acting, two-stroke Diesel engine has 28-in. bore and 40 in. stroke and runs at 95.4 rpm. The mean effective pressure during a test was 64.8 lb per sq in. What was the indicated horsepower? Neglect piston rods.

15. The area of an indicator card taken on an 8 by $11\frac{1}{2}$-in., single-cylinder, four-stroke gasoline engine running at 304 rpm is 1.37 sq in., and the length was 3.18 in. The scale of the spring was 200 lb. What was the indicated horsepower?

16. A high-speed steam engine is direct connected to a centrifugal pump. During a test, the indicated horsepower was 125, while the pump delivered 11,100 gal water per minute against a total head of 30 ft. Dynamometer tests of the pump show its efficiency to be 81 per cent at this load. Find the mechanical efficiency of the engine.

17. A large pumping engine delivers 18,000,000 gal water per 24 hr against a discharge pressure of 60 lb per sq in. The discharge gage is 11 ft above the surface in the supply reservoir. The indicated horsepower is 572. What is the over-all efficiency of the pumping engine?

18. A 9 by 12-in. single-acting, three-cylinder, four-stroke gas engine running at 300 rpm is fitted with a Prony brake with a 48-in. arm. What is the net load on the scales if the engine is delivering 37 hp to the brake?

19. A 5 by 6-in., six-cylinder gasoline engine of the four-stroke, single-acting type has a piston speed of 1,260 ft per min. Indicator cards taken with a 197-lb spring are 2.77 in. long and have an area of 1.54 sq in. The net brake load was 69 lb and the brake constant 0.001. Find (*a*) the indicated horsepower; (*b*) the brake horsepower; (*c*) the mechanical efficiency.

20. A 10 by 12-in. steam engine runs at 250 rpm. The piston-rod diameter is 1.12 in. The mean effective pressure for the head-end indicator card is 52 lb, and the crank-end card is 54 lb. What indicated horsepower does the engine develop?

21. The indicator card from the head end of a steam-engine cylinder is 2.0 sq in. in area. That from the crank end is 2.1 sq in. in area. Both are 3 in. long. The cards were taken with an 80-lb spring. The engine is 16 by 36 in.; 95 rpm; piston-rod diameter, 3 in. What indicated horsepower was developed?

22. A steam engine is 14 by 16 in. and is run at 250 rpm. The diameter of the piston rod is 2⅜ in. The area of the head-end card is 1.48 sq in. and of crank-end card 1.54 sq in. Both were taken with a 60-lb spring, and the cards are 3.20 in. long. The calibration of the spring shows that a correction factor of plus 3 per cent is necessary. What indicated horsepower is developed by the engine?

23. A steam engine is 18 by 24 in. and runs at 150 rpm. Piston-rod diameter 3 in. The indicator cards are 3 in. long. Area of head-end card is 1.87 sq in. and of crank-end card 1.76 sq in. Scale of spring, 100 lb. (*a*) What indicated horsepower is developed? In the test, a Prony brake with a lever arm of 8 ft was used. The net load on the scales exerted by this brake was 1,035 lb. (*b*) What brake horsepower is delivered by the engine? (*c*) What was the mechanical efficiency of the engine?

24. On a no-load test, the mean ordinate of the head-end card of a steam engine is found to be 0.12 in. and of the crank-end card 0.105 in. The scale of the indicator spring is 60 lb. The engine is a 16 by 36 in. with 3-in. piston rod. It is run at 100 rpm. What is the friction horsepower?

25. The pressure-time indicator card of Fig. 53 was taken on a two-stroke engine running at 554 rpm with 25 deg spark advance. Timing wave was 60 cycle. Determine, from the diagram, (*a*) the engine speed and (*b*) the frequency of the combustion ripples.

CHAPTER VII

HEAT AND WORK

ELEMENTARY THERMODYNAMICS

60. Definitions.—For most engineering work, the Fahrenheit scale of temperatures is standard, although the centigrade scale is sometimes used in special cases, as, for example, to express the temperature rise of electrical machines. When temperatures are referred to the ordinary zero (32°F. below the freezing point of water), they are designated as °F., and, when referred to absolute

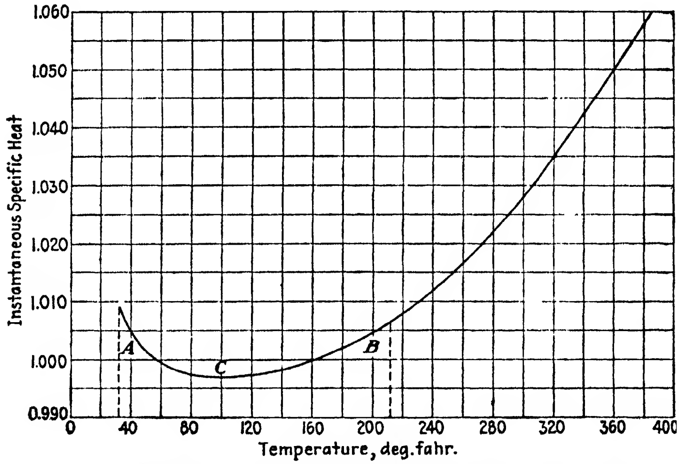


Fig. 60.—Variation of specific heat of water with temperature.

zero (492° below the freezing point of water), they are designated as °R., that is, *degrees Rankine.*

The British thermal unit (Btu), which is the unit of heat used in engineering, is the heat required to raise the temperature of 1 lb water 1°F. Since the heat capacity of water is slightly different at different temperatures, the Btu has been specifically defined by the American Society of Mechanical Engineers as $\frac{1}{180}$ of the heat required to raise 1 lb water from 32 to 212°F.

The variation in heat capacity of water is shown by the curve of Fig. 60. According to the definition of a Btu, the area C is just equal to the sum of areas A and B.

Specific heat is the heat capacity of a substance compared to that of water, or, in other terms, it is the heat (Btu) required to raise the temperature of 1 lb of the substance 1°F. **Molal specific heat** is the heat required to raise 1 mole of the substance 1°.

Table IV gives the specific heats of most of the common substances referred to in this book.

TABLE IV.—INSTANTANEOUS SPECIFIC HEATS AT 60°F. AND 30 IN. HG[1]

Substance	Symbol	Molecular weight	Specific heats, Btu per lb per °F.		Ratio[2] $C_p:C_v$	Molal specific heats, Btu per mole per °F.	
		M	C_p	C_v	k	$M \times C_p$	$M \times C_v$
Helium	He	4.00	1.250	0.753	1.659	5.000	3 012
Oxygen	O_2	32.00	0.2178	0.1555	1.401	6.970	4.976
Nitrogen	N_2	28.02	0.2485	0.1771	1.404	6 964	4.960
Air	....	28.94	0.2400	0.1711	1.403	6.945	4.950
Hydrogen	H_2	2.016	3.389	2.403	1.410	6.830	4.843
Carbon monoxide	CO	28.01	0.2479	0.1765	1.404	6.942	4.945
Carbon dioxide	CO_2	44.02	0.1989	0.1525	1.304	8.760	6.718
Sulphur dioxide	SO_2	64.07	0.1516	0.117	1.29	9.61	7.53
Hydrogen sulphide	H_2S	34.08	0.2533	0.192	1.32	8 63	6.54
Ammonia	NH_3	17.03	0.5230	0.399	1.31	8.90	6.80
Methane	CH_4	16.04	0.5288	0.408	1.31	8.48	6.48
Acetylene	C_2H_2	26.03	0.3829	0.308	1.26	9.97	7.91
Ethylene	C_2H_4	28.05	0.3588	0.285	1.26	10.06	7.98
Ethane	C_2H_6	30.07	0.3858	0.316	1.22	11.59	9.50

[1] Computed from International Critical Tables, Vol. III, p. 3; Vol. V, p. 78.
[2] Determined from velocity of sound.

Expressed in mathematical terms, specific heat is the rate of change of heat with temperature, or

$$c = \frac{dQ}{dt} \tag{6}$$

Joule's equivalent, or the **mechanical equivalent of heat,** is the number of foot-pounds of work necessary to produce 1 Btu, the value of which is generally taken as 778 ft-lb. The first International Steam Table Conference, in London, 1929, fixed the value of a kilocalorie as $\frac{1}{860}$ of an international kilowatt-hour, thus automatically fixing the value of the Btu at 778.26 ft-lb.[1] The recent steam tables by Keenan and Keyes (Art. 103), calculated on this basis, give the heat required to raise 1 lb water from 32 to 212° as 180.07 Btu. The symbol for this constant is J.

Since 1 hp is the ability to do 33,000 ft-lb of work per minute, this is equivalent of 33,000/778 Btu per min., or (33,000/778) 60 = 2545 Btu per hr. That is, 2545 Btu is the heat equivalent of 1 hp used for 1 hr, or 1 hp-hr.

61. Measurement of Pressure.—Fluid pressures are ordinarily expressed in engineering in pounds per square inch or per square foot, and measurement is made by means of some sort of dial gage (see Art. 202). For low pressures where a high degree of accuracy is desired, pressures are measured in inches of height of mercury column which the pressure will sustain; and for extremely low pressures, a water column is used.

In the usual engineering situation the apparatus is surrounded by the pressure of the atmosphere, and so the total pressure or absolute pressure is the pressure indicated by the dial or mercury-column gage plus the pressure of the atmosphere. The pressure of the atmosphere is measured by the height of a mercury column it will force up into the perfect vacuum of the barometer. To reduce pressures in inches of mercury to pounds per square inch, multiply by 0.4908.

Consider the two tanks and the barometer of Fig. 61. Tank A contains a gas under a pressure of p lb per sq in, as indicated by the gage G. The absolute pressure in the tank is $(p + 0.4908b)$ lb per sq in., in which b is the barometric height in inches of mercury.

The tank B contains a gas under a partial vacuum such that the atmosphere forces the mercury column in the manometer M up to a height h. In this case, the manometer is a mercury vacuum gage. (*Vacuums are always measured in inches of*

[1] For a historical summary of this constant see *Mech. Eng.*, June, 1934, p. 347.

mercury.) But there is still some pressure in tank B; otherwise, the height h would equal b. Then the absolute pressure in B equals (b-h) in. Hg or 0.4908 (b-h) lb per sq in.

These pressures are illustrated graphically in Fig. 62.

The atmospheric pressure is continually changing and at altitudes is lower than at sea level, however, by definition, a

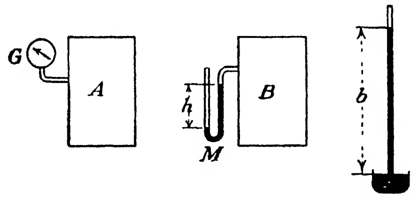


FIG. 61.— Gage and absolute pressures; apparatus.

"standard" atmosphere will support a column of mercury 760 mm high, the mercury having a density of 13.5951 g per cu cm (at 32°F.), at standard gravity of 980.665 cm per sec per sec. This corresponds to 29.92 in. The standard engineering reference pressure for many purposes is 30 in. (see Arts. 39, 40).

62. Boyle's and Charles's Laws.—*Boyle's law* states that, for a "perfect" gas, the volume of a given mass of the gas varies inversely as the pressure, the temperature remaining constant.

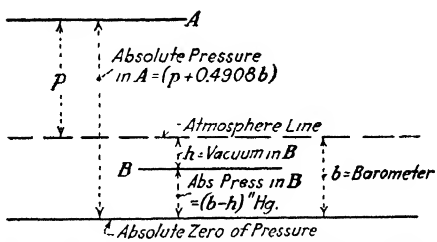


FIG. 62.—Gage and absolute pressures; graph.

This is illustrated graphically by Fig. 63(A) in which the pressure of, say, 1 lb of a gas is plotted against its volume, the temperature being constant. Beginning with a pressure p_1 and volume v_1, if the volume be doubled to v_2, the pressure will be p_2 which is one-half p_1; and, if the volume is quadrupled to v_3, the pressure p_3 will be one-fourth p_1.

Stated in symbols, we have

$$\frac{p_1}{p_2} = \frac{v_2}{v_1} \text{ or } p_1v_1 = p_2v_2$$

and, likewise,

$$\frac{p_2}{p_3} = \frac{v_3}{v_2} \text{ or } p_2v_2 = p_3v_3$$

That is,

$$p_1v_1 = p_2v_2 = p_3v_3 = p\,v = \text{a constant} \tag{7}$$

This is the equation of an equilateral hyperbola.

Charles's law states that, if the volume of a given mass of a "perfect" gas is kept constant, the pressure will vary directly as the *absolute* temperature or, if the pressure is kept constant, the volume varies directly as the *absolute* temperature. Figure 63(*b*) shows Charles's law graphically. If the temperature is changed from T_1 to T_2, the pressure will change from p_1 to p_2, and

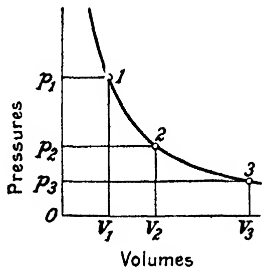


FIG. 63*A*.—Boyle's law.

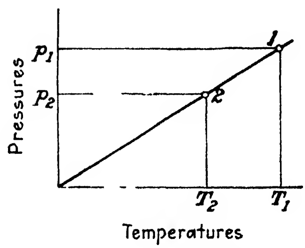


FIG. 63*B*. Charles's law.

the line joining points 1 and 2 would intersect the two axes at absolute zero.

Stated mathematically, Charles's law is

$$\frac{T_1}{T_2} = \frac{p_1}{p_2} \text{ if } v \text{ is constant} \tag{8}$$

or

$$\frac{T_1}{T_2} = \frac{v_1}{v_2} \text{ if } p \text{ is constant}$$

A "perfect" gas may be defined as one which follows Boyle's and Charles's laws perfectly. For most engineering purposes, air, hydrogen, nitrogen, and oxygen are considered as perfect gases, but some exceptions to this occur in which the variations of the gases from Boyle's and Charles's laws must be considered.

63. The Characteristic Equation of a Perfect Gas.—In the usual case pressure, volume, and temperature may all change at the same time. For example, the volume occupied by a pound of air may change owing to a simultaneous change in the barometric pressure and the temperature.

In Fig. 64 let OP, OV, and OT represent three axes of pressure, volume, and temperature mutually at right angles to each other, and suppose the point 1 in this space diagram represents the pressure p_1, the temperature T_1, and the volume v_1 of 1 lb of a perfect gas at some initial instant. At some later instant we find that the pressure is p_2, the temperature T_2, and the volume v_2 as represented by point 2 in the space diagram. This change may have taken place over any devious space path, such as the dotted line between points 1 and 2. By proper control, however, we might cause the temperature to remain constant at T_1 while the pressure drops to p_x and the volume increases to v_2 following Boyle's law. Then, if we should keep the volume constant at v_2, we could drop the pressure from p_x to p_2, following Charles's law.

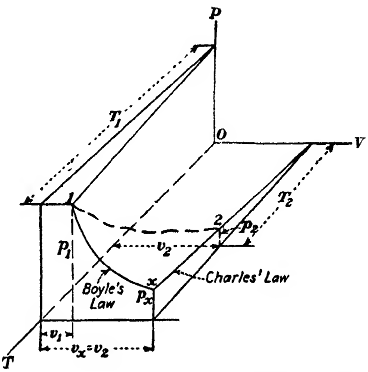


FIG. 64.—Boyle's and Charles's laws combined.

Now, by Boyle's law applied to path 1-x,

$$p_1v_1 = p_xv_2 \text{ (since } v_2 = v_x)$$

and

$$\frac{p_1v_1}{v_2} = p_x \tag{9}$$

Also, by Charles's law, applied to path x-2,

$$\frac{T_1}{T_2} = \frac{p_x}{p_2} \text{ (since } T_1 = T_x)$$

and

$$\frac{T_1}{T_2}p_2 = p_x \tag{10}$$

Combining Eqs. (9) and (10), we have

$$p_x = \frac{p_1v_1}{v_2} = \frac{T_1}{T_2}p_2$$

or

$$\frac{p_1v_1}{T_1} = \frac{p_2v_2}{T_2}$$

and, since point 2 was an arbitrary point, we might write, by analogy,

$$\frac{p_1v_1}{T_1} = \frac{p_2v_2}{T_2} = \frac{p_3v_3}{T_3} = \frac{pv}{T} = R = \text{a constant} \qquad (11)$$

This constant is conventionally denoted by R and is called the "gas constant." Its numerical value is determined for any gas by measuring simultaneously the pressure, volume, and temperature of 1 lb of the gas and substituting in Eq. (11). Thus it has been found by experiment that for air,

if $p = 14.69$ lb per sq in. or (14.69×144) lb per sq ft.

$v = 12.387$ cu ft when the temperature is 32°F.

$T = 32 + 459.6 = 491.6$°R.

we have

$$R = \frac{(14.69 \times 144) \times 12.387}{491.6} = 53.37 \text{ for air}$$

The customary form of Eq. (11) is

$$pv = RT \qquad (12)$$

and this is called the "characteristic" equation of a perfect gas. If more than 1 lb of the gas is considered, it takes the form

$$pv = wRT \qquad (13)$$

in which w is the weight of the gas.

Equation (13) is very useful in determining any one of the four properties p, v, w, or T of a given mass of a gas if the other three are known.

Example A.—What is the weight of air contained in a tank having a volume of 100 cu ft if the pressure is 300 lb per sq in and the temperature is 115°F.?

SOLUTION:

$$w = \frac{pv}{RT} = \frac{(300 \times 144) \times 100}{53.37 \times (115 + 460)} = 1{,}408 \text{ lb.}$$

64. Paths of Expansion.—In Fig. 65 consider a perfectly insulated cylinder having a conducting bottom and fitted with a frictionless piston having an area A. Assume that the total weight due to the atmospheric pressure and weight of piston is equal to P. Assume further that the cylinder contains under the piston 1 lb of a perfect gas and that the gas may receive heat from some heat source represented by the candle.

Now suppose that in a given small interval of time, the candle supplies dQ Btu to the gas. This energy will be absorbed by the gas in two different ways: (1) by increasing its temperature and (2) by doing external work, that is, by raising the piston. Heat used to increase the temperature is called "sensible" heat, because it can be sensed, or preferably "internal energy," and is generally denoted by dE. Heat used in doing external work in raising the weight is denoted by dW. Then

$$dQ = dE + dW$$

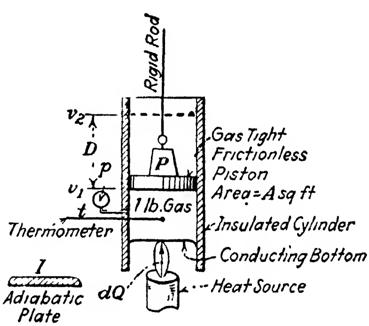


FIG. 65.—Adding heat to a mass of gas.

That is to say, the thermal energy possessed by the gas consists of internal energy E and external energy W, both measured above some arbitrary datum of temperature and pressure. The sum of these two energies is called the *enthalpy*[1] of the substance and its symbol is H. The usual expression for enthalpy is then

$$H = E + \frac{pv}{J} \tag{14}$$

since pv/J represents external energy in thermal units.

In actual gases there is always a very small amount of internal friction, which is ignored in this discussion by considering the gas as perfect.

While heat is being added to the pound of the perfect gas in Fig. 65, the expansion of the gas may be controlled in several ways:

1. We may allow the piston to move up freely, and, since there is no friction, the pressure inside will remain constant while the temperature rises. This heat is added at **constant pressure.**

[1] Formerly called variously, "heat content," "total heat," "total heat content," "thermal potential."

2. If the rod on top of the weight is held so that the piston cannot move, the heat will be added at **constant volume.**

3. It would be possible to pull up on the rod while heat is being added at just such a rate that the temperature of the gas remains constant. This is called **constant temperature** or **isothermal** expansion, and, while it proceeds, the pressure is gradually reduced.

4. It would also be possible, and this is the usual case in practice, to allow the piston to move upward more rapidly or more slowly than necessary for constant-temperature expansion. If the piston is pulled up more rapidly, the pressure is reduced more rapidly, and, if it be pulled up more slowly, the pressure is reduced more slowly than in constant-temperature expansion. Any expansion, in which the temperature changes, is called **polytropic** expansion.

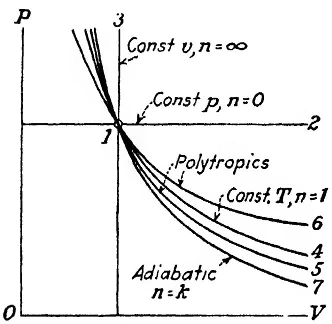


FIG. 66. Some possible pressure-volume paths for gases.

5. Suppose now that the candle be removed and an insulating plate I be put under the cylinder. Then, if the piston be pulled up, the pressure of the gas will be reduced, and, since it can receive no heat from an outside source, the temperature must also fall. This is called **adiabatic** or **isentropic** or **constant-entropy** expansion. The word "adiabatic" means "not passing through." "Entropy" will be explained in Art 72.

These five processes may be illustrated graphically, as in Fig. 66 in which pressures are plotted against volumes, the lines 1-2, 1-3, 1-4, etc., being called "paths" of expansion.

65. Equations of Paths.—It will be noted that the path 1-4 in Fig. 66 is the constant-temperature curve of Fig. 62 described in Art. 62 and that its form is an equilateral hyperbola the equation for which is

$$pv = \text{a constant}$$

The path 1-2 being a straight line parallel to the volume axis has the equation

$$p = \text{a constant}$$

and the path 1-3 being parallel to the pressure axis has the equation

$$v = \text{a constant}$$

Now it is possible to show that all these paths are variants of the general path, the polytropic, which has the equation

$$pv^n = \text{a constant}$$

in which the exponent n may have various values. Thus the equation of the isothermal, or constant-temperature, path may be expressed as

$$pv^1 = \text{a constant}$$

in which $n = 1$. The constant-pressure and constant-volume paths may be expressed, respectively, as

$$pv^0 = \text{a constant}$$

and

$$pv^\infty = \text{a constant}$$

Paths falling between 1-4 and 1-2 have an exponent n between 1 and zero, and those falling below 1-4 have an exponent greater than 1. The value of n for the adiabatic path will be discussed in Art. 70.

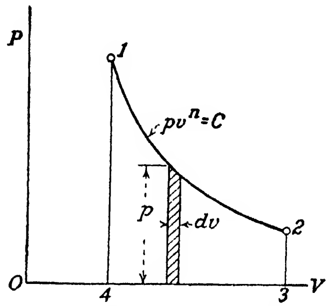


FIG. 67.—Work area under a polytropic path.

66. Work Done during Expansion.— As stated in Art. 46, when a gas expands against an external pressure, external work is done. Figure 67 illustrates a polytropic curve whose equation is

$$pv^n = C, \text{ a constant}$$

and it is desired to determine the work done by the gas when it is allowed to expand from p_1v_1 to p_2v_2 along this path.

Since areas on the p-v diagram represent work, the work done during expansion dv at pressure p is

$$dW = pdv$$

and the work done between states 1 and 2 is

$$W_{1-2} = \int_{v_1}^{v_2} pdv \tag{15}$$

Now, since the equation of the curve is

$$pv^n = C$$

$$p = \frac{C}{v^n} \tag{16}$$

substituting Eqs. (16) in (15), we have

$$W_{1-2} = \int_{v_1}^{v_2} \frac{C}{v^n} dv$$

which, integrated, gives

$$W_{1-2} = \frac{C}{1-n}\Big[v^{(1-n)}\Big]_{v_1}^{v_2}$$

$$= \frac{C}{1-n}[v_2^{(1-n)} - v_1^{(1-n)}]$$

$$= \frac{1}{1-n}[Cv_2^{(1-n)} - Cv_1^{(1-n)}] \tag{17}$$

Now, substituting in Eq. (17) the values $p_2v_2^n$ and $p_1v_1^n$ for the constant C, we have

$$W_{1-2} = \frac{1}{1-n}[p_2v_2^nv_2^{(1-n)} - p_1v_1^nv_1^{(1-n)}]$$

$$= \frac{1}{1-n}[p_2v_2 - p_1v_1]$$

and changing signs, we have, finally,

$$W_{1-2} = \frac{p_1v_1 - p_2v_2}{n-1} \tag{18}$$

By substituting for p_1v_1 and p_2v_2 in Eq. (18) their values RT_1 and RT_2 from $pv = RT$, we have

$$W_{1-2} = \frac{R(T_1 - T_2)}{n-1} \tag{19}$$

Example A.—How much external work is done when 1 lb air (considered a perfect gas) expands from 100 lb per sq in. abs to 50 lb per sq in. abs along the path $pv^{1.4} = C$, the initial volume being 1.5 cu ft and the final volume 2.46 cu ft?

SOLUTION:

$$W_{1-2} = \frac{p_1v_1 - p_2v_2}{n-1}$$

$$W_{1-2} = \frac{144(100 \times 1.5 - 50 \times 2.46)}{1.4 - 1}$$

$$= \frac{144(150 - 123)}{0.4} = \frac{144 \times 27}{0.4} = 9{,}720 \text{ ft-lb}$$

It must be remembered that p is pounds per square foot in these equations.

Applying the general solutions of Eqs. (18) and (19) to the specific cases of Art. 65, we have for path 1-2 of Fig. 66, *constant-pressure* expansion,

$$W_{1-2} = \frac{p_1v_1 - p_1v_2}{0 - 1} = p_1(v_2 - v_1) \tag{20}$$

or

$$W_{1-2} = \frac{R(T_1 - T_2)}{0 - 1} = R(T_2 - T_1) \tag{21}$$

For path 1-3 of Fig. 66, *constant volume*, we have

$$W_{1-2} = \frac{p_1v_1 - p_2v_1}{\infty - 1} = \frac{v_1(p_1 - p_2)}{\infty - 1} = 0$$

which we know by experience must be true, since there is no motion of the piston and hence no work is done.

For path 1-4 of Fig. 66, *constant temperature*, we have

$$W_{1-2} = \frac{p_1v_1 - p_2v_2}{1 - 1} = \frac{p_1v_1 - p_2v_2}{0}$$

which is indeterminate. We must resort, therefore, to the following method: Suppose that in Fig. 67 we let the path be the isothermal, the equation for which is $pv = C$. Then $p = C/v$, which, substituted in

$$W_{1-2} = \int_{v_1}^{v_2} pdv$$

gives

$$W_{1-2} = C\int_{v_1}^{v_2}\frac{dv}{v} = C[\log_e v]_{v_1}^{v_2} = C\log_e\frac{v_2}{v_1} \tag{22}$$

Now, substituting in Eq. (22) the value p_1v_1 for the constant C, we have

$$W_{1-2} = p_1v_1 \log_e\frac{v_2}{v_1} \tag{23}$$

or, substituting RT_1 for p_1v_1, this becomes

$$W_{1-2} = RT_1 \log_e\frac{v_2}{v_1} \tag{24}$$

Example B.—If 1 lb air is expanded isothermally from $p_1 = 300$ lb per sq in. abs and $v_1 = 0.6$ cu ft to a volume of 7 cu ft, how much external work is done?

SOLUTION:

$$W_{1-2} = p_1v_1 \log_e \frac{v_2}{v_1}$$
$$= (144 \times 300) \times 0.6 \times 2.3 \log_{10}\left(\frac{7}{0.6}\right)$$
$$= 100{,}200 \text{ ft-lb}$$

67. The Two Specific Heats.—In Art. 64 it was shown that the heat added to the body of gas by the heat source,

$$dQ = dE + dW$$

or, in finite terms,

$$Q = E + W \tag{25}$$

In order to determine the internal energy E, it would be necessary, in the apparatus of Fig. 65, to hold the piston in a fixed position so that no external work would be done and then to measure the heat added in order to raise the temperature from T_1 to T_2. Then this internal energy would be

$$E = c_v(T_2 - T_1) \tag{26}$$

the specific heat at constant volume being used because the volume was kept constant. It can be shown that the change in internal energy is always the specific heat at **constant volume** times the change in temperature, because this energy is due to the velocity of the molecules and the velocity of the molecules is the same for a given temperature whether they have a larger space or a smaller one in which to move.

If we apply Eq. (25) to the constant-pressure process, we have for the heat added to 1 lb gas while the temperature rises from T_1 to T_2 and the volume increases from v_1 to v_2

$$Q_{1-2} = E_{1-2} + W_{1-2}$$

or, from Eqs. (26) and (20),

$$= c_v(T_2 - T_1) + \frac{1}{J}p_1(v_2 - v_1) \tag{27}$$

$1/J$ being inserted to convert work in foot-pounds into Btu. Also, using values of E and W from Eqs. (26) and (21),

$$Q_{1-2} = c_v(T_2 - T_1) + \frac{1}{J}R(T_2 - T\ ) \tag{28}$$

But for constant pressure

$$Q_{1-2} = c_p(T_2 - T_1) \tag{29}$$

in which c_p = the specific heat of the gas at constant pressure. Then, combining Eqs. (28) and (29), we have

$$c_p(T_2 - T_1) = c_v(T_2 - T_1) + \frac{R}{J}(T_2 - T_1)$$

and

$$c_p = c_v + \frac{R}{J}$$

or

$$c_p - c_v = \frac{R}{J} \tag{30}$$

This very significant relation means that the constant R equals the difference between the two specific heats at constant pressure and constant volume when expressed in foot-pounds.

If one uses the values for the *molal* specific heats as given in Table IV, page 97, in Eq. (30) it will be found that R will have a nearly constant value for all gases. This value is about 1,544 ft-lb and is called the "universal gas constant" (see Art. 79). The symbol r is used for this constant.

68. Heat Added on Any Path.—Applying Eq. (25) to the general expansion curve, the polytropic, assuming 1 lb of the gas, we have

$$Q_{1-2} = E_{1-2} + W_{1-2}$$

From Eqs. (26) and (18),

$$= c_v(T_2 - T_1) + \frac{1}{J}\left(\frac{p_1v_1 - p_2v_2}{n - 1}\right)$$

or, from Eq. (19),

$$= c_v(T_2 - T_1) + \frac{1}{J(n - 1)}R(T_1 - T_2)$$

whence,

$$Q_{1-2} = \left[-c_v + \frac{R}{J(n - 1)}\right](T_1 - T_2) \tag{31}$$

Example A.—How much heat must be added to 1 lb air to cause it to expand from p_1 = 200 lb per sq in., v_1 = 0.985 cu ft, and T_1 = 530°R. to p_2 = 20 lb per sq in. and v_2 = 12.7 cu ft along the path $pv^{0.9} = C$?

SOLUTION:

$$Q_{1-2} = \left[-c_v + \frac{R}{J(n-1)} \right](T_1 - T_2)$$

$$T_2 = \frac{p_2v_2}{R} = \frac{144 \times 20 \times 12.7}{53.37} = 685°\text{R.}$$

Then

$$\begin{aligned} Q &= \left[-0.1711 + \frac{53.37}{778(0.9-1)} \right](530 - 685) \\ &= [-0.1711 - 0.687](-155) \\ &= 0.858 \times 155 = 132.0 \text{ Btu} \end{aligned}$$

69. Heat Added on the Isothermal Path.—Referring again to Eq. (25),

$$Q_{1-2} = E_{1-2} + W_{1-2}$$

it will be seen that, for an isothermal path, the change in internal energy is zero, since there is no change in temperature. Then in this case

$$Q_{1-2} = W_{1-2} \tag{32}$$

and, stated in words, this means that on an isothermal path, the heat added is equal to the work done.

70. The Adiabatic Path.—Reference was made in Art. 64 to an adiabatic path as one along which the gas was insulated so that no heat could be given to the gas or abstracted from it during the expansion from p_1v_1 to p_2v_2. For the adiabatic then Q_{1-2} is zero, and Eq. (25) becomes

$$0 = E_{1-2} + W_{1-2} \tag{33}$$

Transposing,

$$-E_{1-2} = W_{1-2} \tag{34}$$

the negative sign before E indicating that there is a reduction in the internal energy of the gas because of the work done and that the temperature falls. In other words, on an adiabatic path a gas does external work at the expense of its own internal energy.

Now, substituting in Eq. (34) for E and W their values from Eqs. (26) and (19), respectively, we have

$$-c_v(T_2 - T_1) = \frac{R(T_1 - T_2)}{J(n-1)}$$

or

$$c_v(T_1 - T_2) = \frac{R}{J(n-1)}(T_1 - T_2)$$

and

$$c_v = \frac{R}{J}\left(\frac{1}{n-1}\right) \tag{35}$$

Now, from Eq. (30), $R/J = c_p - c_v$ which, substituted in Eq. (35), gives

$$c_v = (c_p - c_v)\frac{1}{n-1}$$

and

$$n - 1 = \frac{c_p - c_v}{c_v} = \frac{c_p}{c_v} - 1$$

Therefore for an adiabatic path

$$n = \frac{c_p}{c_v} \tag{36}$$

This means that the exponent of v for an adiabatic expansion is the ratio of the two specific heats, which ratio is conventionally represented by k. Then

$$pv^k = \text{a constant} \tag{37}$$

is the equation of a perfect gas adiabatic. For air at normal temperature, $k = 1.402$. For value of k for other gases, see Table IV.

The work done under an adiabatic, then, is obtained by substituting k for n in Eq. (18), or,

$$W = \frac{p_1v_1 - p_2v_2}{k-1} \tag{38}$$

71. Relations between p, v, and T along a Polytropic.—Taking as the equation of the polytropic of Fig. 66 or 67,

$$pv^n = C$$

we have

$$p_1v_1^n = p_2v_2^n$$

from which

$$\frac{p_1}{p_2} = \left(\frac{v_2}{v_1}\right)^n \tag{39}$$

Also

$$\left(\frac{p_1}{p_2}\right)^{\frac{1}{n}} = \frac{v_2}{v_1} \tag{40}$$

Now, from the characteristic equation of a perfect gas, $pv = RT$, we have

$$\frac{p_1v_1}{T_1} = \frac{p_2v_2}{T_2}$$

and

$$\left(\frac{p_1}{p_2}\right)\left(\frac{v_1}{v_2}\right) = \frac{T_1}{T_2} \tag{41}$$

Substituting Eq. (39) in Eq. (41), we have

$$\left(\frac{v_2}{v_1}\right)^n\left(\frac{v_1}{v_2}\right) = \frac{T_1}{T_2}$$

and

$$\left(\frac{v_2}{v_1}\right)^{n-1} = \frac{T_1}{T_2} \tag{42}$$

Likewise, substituting Eq. (40) in Eq. (41), we have

$$\left(\frac{p_1}{p_2}\right)\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}} = \frac{T_1}{T_2}$$

and

$$\left(\frac{p_1}{p_2}\right)^{1-\frac{1}{n}} = \left(\frac{p_1}{p_2}\right)^{\frac{n-1}{n}} = \frac{T_1}{T_2} \tag{43}$$

Equations for the adiabatic would, of course, have the same forms with k substituted for n in each case.

Example A.—Air is expanded adiabatically from $p_1 = 125$ lb per sq in. and $T_1 = 600°$R. to $p_2 = 12.5$ lb per sq in. What is its final temperature?

SOLUTION.—For an adiabatic

$$\left(\frac{p_1}{p_2}\right)^{\frac{k-1}{k}} = \frac{T_1}{T_2}$$

$$T_2 = \frac{T_1}{\left(\frac{p_1}{p_2}\right)^{\frac{k-1}{k}}} = \frac{600}{\left(\frac{125}{12.5}\right)^{\frac{0.403}{1.403}}} = \frac{600}{10^{0.287}} = \frac{600}{1.93} = 311°\text{R.}$$

$$t_2 = 311 - 460 = -149°\text{F.}$$

72. Entropy.—Consider again the cylinder of Fig. 65 (Art. 64), and suppose a small quantity of heat dQ is added to the gas while the piston is held stationary. Then the heat added is equal to the specific heat at constant volume times the rise in temperature, or

$$dQ = c_v dT \tag{44}$$

Now we may devise a new thermal quantity by dividing this heat change dQ by the absolute temperature at which the heat change takes place, which will give us

$$\frac{dQ}{T} = \frac{c_v dT}{T} \tag{45}$$

and we may call this new quantity "entropy change." It should be noted that this quantity dQ/T is a purely mathematical expression of the same nature as the moment of inertia of a beam section and, so far as this discussion is concerned, need not be considered to have a real existence any more than "moment of inertia" has. Entropy is, however, a tool as useful in solving heat problems as is moment of inertia in solving beam problems, and its use should be mastered by the student. If we denote entropy change by ds, Eq. (45) becomes

$$ds = \frac{dQ}{T} = c_v \frac{dT}{T} \tag{45a}$$

and, integrating,

$$\int_{s_1}^{s_2} ds = c_v \int_{T_1}^{T_2} \frac{dT}{T} = c_v [\log_e T]_{T_1}^{T_2}$$

and

$$s_2 - s_1 = c_v \log_e \frac{T_2}{T_1} \tag{46}$$

for heat added at constant volume. For heat added at constant pressure, we should have

$$s_2 - s_1 = c_p \log_e \frac{T_2}{T_1} \tag{47}$$

For any substance, iron, stone, or water, having a single value of specific heat, we should have a similar expression, which, in general terms, is

$$s_2 - s_1 = c \log_e \frac{T_2}{T_1} \tag{48}$$

It should be noted that ds is *change* of entropy and that there is no zero of entropy, although we might take any arbitrary temperature above which entropy is calculated. For water this temperature is taken as 32°, see Art. 101.

The symbol S is used for any weight and s for unit weight of substance. For w weight of substance, $S = ws$.

Another important item to note is that the temperature by which the heat change is divided is the *absolute* temperature.

Example A.—What is the change in entropy of 1 lb water if the temperature is changed from 32 to 102°F.?

SOLUTION.—The change in temperature is from 492 to 562°R.

$$\begin{aligned} s_{2-1} &= c \log_e \frac{T_2}{T_1} \\ &= 1 \times \log_e \frac{562}{492} = 1 \times \log_e 1.142 \\ &= 0.1327 \text{ unit} \end{aligned}$$

Example B.—What is the change in entropy of 1 lb gas when heated at constant pressure from 500 to 750°R., c_p remaining constant at 0.24?

SOLUTION:

$$\begin{aligned} s_{2-1} &= 0.24 \log_e \frac{750}{500} = 0.24 \times 2.3 \times 0.1761 \\ &= 0.0972 \end{aligned}$$

If this solution be made in five 50° steps, we should have the following entropy changes:

			Units
	From 500 to 550°	=	0.0222
	From 550 to 600	=	0.0210
	From 600 to 650	=	0.0193
	From 650 to 700	=	0.0179
	From 700 to 750	=	0.0167
Total	From 500 to 750	=	0.0971, as before

These entropy changes are plotted cumulatively as abscissas against temperatures as ordinates in Fig. 68, the locus being, of course, a logarithmic curve.

Since by definition

$$dS = \frac{dQ}{T}$$

it follows that

$$TdS = dQ$$

That is, the temperature times the entropy change equals the heat change, and, if we plot temperatures against entropies, the areas on this plane will represent heat in the same way that areas on the pressure-volume plane represent work. Thus, in Fig. 68, if the changes in entropy and temperature of 1 lb of air are shown by the curve 2-3, the area 2-3-8-7 under this curve represents the heat change. If we assume 2-3 to be a straight line, the area equals $(0.0210) \times [(550 + 600)/2] = 0.021 \times 575 = 12.07$

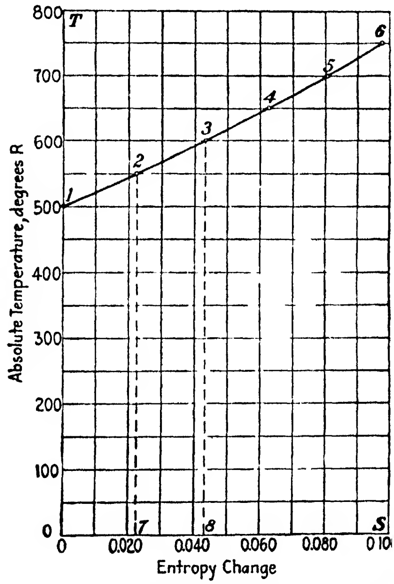


FIG. 68.—Temperature-entropy (T-S) diagram.

Btu. This agrees nearly with what we know to be true, namely' that the heat change between 550 and 600°R. is

$$(600 - 550)(0.24) = 12 \text{ Btu}$$

Stated mathematically, we have

$$Q = \int_{s_1}^{s_2} TdS$$

In Art. 64 it was said that an adiabatic path for a perfect gas is a constant-entropy path. Since by definition

$$dS = \frac{dQ}{T}$$

it follows that if there is no heat added or taken away from the substance, dQ becomes zero, and, consequently, there can be no change in entropy. This means that an adiabatic path on the T-S diagram is a vertical or constant-entropy line, as, for example, 2-7 or 3-8 of Fig. 68.

Problems

1. A salt solution has a specific heat of 1.04; 19,000 lb of it are pumped per hour through a cooler against a total head of 21 ft, the temperature rising from 5 to 17°F. (*a*) How many Btu are absorbed by the cooler per hour? (*b*) What is the net horsepower required to drive the pump?

2. If 1,576,000 Btu are given to an engine in an hour and if the engine can convert 6 per cent of this heat into work, what is the horsepower of the engine?

3. If the pressure gage on a boiler reads 150 lb and the barometer reads 28.3 in., what is the absolute pressure in the boiler in pounds per square inch?

4. The vacuum gage on a condenser reads 27.2 in. and at the same time the barometer reads 29.1 in. What is (*a*) the absolute pressure in the condenser in pounds per square inch; (*b*) the vacuum-gage reading reduced to a 30-in. basis?

5. A boiler-feed pump is located 14 ft below the water line of the boiler. The pump draws water from a tank located 4 ft above the pump cylinder. If the pressure in the boiler is 40 lb gage, neglecting friction losses, what is the least total head against which the pump must act (*a*) in feet; (*b*) in pounds per square inch? (*c*) What is the least height of water level in a standpipe above the boiler in order that the water will flow into the boiler by gravity?

6. An engine is given 10,000,000 Btu per hr by the steam supplied. The over-all efficiency being 8.5 per cent, (*a*) what is the horsepower of the engine? (*b*) The engine is direct connected to a centrifugal pump having an efficiency of 75 per cent. The delivery pressure gage reads 15.5 lb, the suction gage reads 16 in. of vacuum, the pressure gage being 4 ft higher than the suction gage. The barometer is 28.9 in. How many gallons of water will the pump deliver per hour?

7. A certain auditorium will seat 3,000 people. If each person is supplied with 30 cu. ft. of air per minute for ventilation, the outside temperature being 0°F. and that in the hall being 70°, how many pounds of air will be admitted per hour, and how many Btu will be required to heat it?

8. The compressed-air tank on a street car has a volume of 250 cu ft. The pressure in the tank is 250 lb gage and the temperature is 60°F. There are two air cylinders each 8 by 10 in. The brakes take air at 40 lb gage pressure and 60° temperature. How many times will the tank operate the brakes at full pressure of 40 lb per sq in.?

9. (*a*) How many pounds of oxygen may be shipped in a steel bottle having a volume of 1.6 cu ft, the maximum pressure allowable being 1,500 lb per sq in. abs at a temperature of 120°F.? (*b*) If the temperature rises to 150°F., what is the pressure of the gas?

10. The oxygen tank for a bomb calorimeter contains 0.3 cu ft oxygen at 1,200 lb per sq in. absolute and 72°F. The minimum pressure used in the calorimeter is 275 lb per sq in. absolute. How many pounds of oxygen are available for use?

11. In a 10-in. gas-transmission pipe line the pressure is 500 lb per sq in. gage and the entrance velocity 150 ft per sec. How many cubic feet of gas at 1 lb gage pressure are delivered to the receiver per day? Barometer is 29 in.

12. (*a*) How many cubic feet of "free air," that is, atmospheric pressure, at 14.7 lb per sq in. and 60°F. must be compressed to fill a 1,000-cu ft tank at 100 lb per sq in. gage and 75°F.? (*b*) How many pounds of air?

13. What is the value of the exponent n in the equation of a perfect-gas expansion if $p_1/p_2 = 10$ and $v_2/v_1 = 9$?

14. What is the value of the exponent n in $pv^n = C$ when air is compressed from 13 to 80 lb per sq in. abs, the volume at the end of compression being one-sixth that at the beginning?

15. A pound of air is expanded along the path $pv^n = C$ from 200 lb per sq in. abs and 203°F. to 12.3 lb per sq in. abs and a volume of 18 cu ft. Calculate (*a*) the initial volume; (*b*) the value of n; (*c*) the final temperature; (*d*) the work done during the expansion; (*e*) the heat added or abstracted during the expansion

16. One pound of air is compressed polytropically from 14 to 112.3 lb per sq in. abs, the initial temperature being 60°F. and the final temperature being 100°F. Calculate (*a*) the initial and final volume; (*b*) the exponent n; (*c*) the work of compression; (*d*) the heat added or subtracted during the process.

17. The ratio of compression (volume ratio) in a gas engine is 4.5:1. What are the final temperature and pressure if the initial temperature is 90°F. and the initial pressure 11 lb per sq in. abs, assuming that the path of compression is $pv^1 = C$?

18. A pound of air expands adiabatically from 100 lb per sq in. abs and 870°R. to 10 lb per sq in. abs. Calculate (*a*) the final temperature; (*b*) the initial and final volumes; (*c*) the external work done.

19. How many foot-pounds of work are done in pumping up an automobile tire from atmospheric pressure of 14 to 60 lb gage? The air space in the tube when inflated is a torus whose dimensions are 4 and 30 in. Assume the air temperature to be 60°F. and the air in the tire to have a final temperature of 100°F. Ignore the heat lost and that given to the tire.

20. One cubic foot of air expands adiabatically from 14.7 lb and 60°F. to 2 lb absolute. Find (*a*) temperature; (*b*) volume; (*c*) external work done; (*d*) internal energy converted into work.

21. One pound of air at 12.3 lb and 60°F. passes through the following cycle: compressed isothermally to 100 lb abs, then compressed at constant pressure to 0.5 cu ft, expands along the path $n = 1.2$ to 12.3 lb pressure, and then at constant pressure to its original volume. Compute (*a*), p, v, t at end of each step; (*b*) work expanded during each step; (*c*) net work area of diagram.

22. A pound of air at atmospheric pressure and 62° is compressed along the path, $pv^{1.32} = C$ to 150 lb gage, expanded isothermally to its original volume, and then pressure and temperature dropped at constant volume to its original state. Compute (*a*), p_1, v_1, T_1, p_2, v_2, T_2, p_3, v_3, T_3, (*b*) heat added or abstracted as heat during each step; (*c*) net work area of cycle.

23. One pound of air is passed through the following cycle beginning at 32°F. and 12.39 cu ft: (1) compressed along the path $pv^{1.2} = C$ to 100 lb per sq in. abs, (2) pressure is dropped to initial pressure at constant volume, (3) expansion at constant pressure to the initial state, 12.39 cu. ft. Calculate (*a*), p, v, and T at the end of each step; (*b*) heat added or abstracted during each step; (*c*) net work area of the cycle.

24. One pound of air at 60°F. and 14 lb per sq in. abs is passed through the following cycle: (1) compressed adiabatically to one-fifth of its initial volume, (2) has 1000 Btu added at constant volume, (3) expands adiabatically to its initial volume, (4) is cooled at constant volume to initial temperature and pressure. (*a*) Sketch the cycle and calculate the pressure, volume, and temperature at the end of each process. (*b*) Calculate the work area of the cycle. Assume the specific heats to be those given in Table IV.

25. One pound of air is passed through the following cycle from 100°F. and 13.5 lb per sq in. abs.: (1) compressed adiabatically to one-fifteenth of its initial volume, (2) has 1000 Btu added at constant pressure, (3) expands adiabatically to its initial volume, (4) is cooled at constant volume to its initial temperature and pressure. (*a*) Sketch the cycle and calculate the pressure, volume, and temperature at the end of each process. (*b*) Calculate the work area of the cycle. Assume the specific heats to be those given in Table IV.

26. What is the change in entropy when 10 lb of a substance having a specific heat of 0.11 are heated from 60 to 270°F.?

27. Calculate the changes in entropy when 1 lb of air is heated from 32 to 200°F. (*a*) at constant pressure; (*b*) at constant volume.

28. Calculate in six 300° steps the change in entropy when 1 lb of air is heated at constant volume from 200 to 2000°F. Plot temperature against entropy.

29. Calculate the changes in entropy by a number of steps for each of the four processes of the cycle of Prob. 24 and plot the cycle on the temperature-entropy diagram.

30. Calculate the change in entropy for 1 lb of water from 32 to 212°F. in nine 20° steps, using the average specific heat for each step as shown by the curve of Fig. 60. Plot the temperature-entropy diagram.

CHAPTER VIII

ANALYSIS OF INTERNAL-COMBUSTION ENGINE CYCLES

73. Theoretical Efficiencies.—It is the purpose of the present chapter to examine into the characteristics of the ideal Otto and Diesel cycles in order to obtain standards by which we may measure the degree of perfection attained by the actual engines.

The general definition for efficiency of any machine is

$$\text{Efficiency} = \frac{\text{output}}{\text{input}}, \text{ or } \frac{\text{input} - \text{losses}}{\text{input}} \tag{49}$$

and in heat machinery this becomes

$$e = \frac{\text{equivalent B.t.u. output}}{\text{equivalent B.t.u. input}} \tag{50}$$

In heat engines this may be expressed in symbols, as

$$\begin{aligned} e &= \frac{W}{Q_1} \\ &= \frac{Q_1 - Q_2}{Q_1} \end{aligned} \tag{51}$$

in which W = the net heat equivalent of the work output.
Q_1 = the total heat input.
Q_2 = the loss in wasted or rejected heat.

THE OTTO CYCLE

74. The Ideal Otto Cycle.—In Fig. 49 was shown an indicator card from an Otto four-stroke gasoline engine. Figure 69 shows this same indicator card "idealized" by making the following assumptions:

1. The stroke AB is the same as for the actual card.
2. The clearance FA is same as for the actual engine.
3. The suction and exhaust lines are coincident at atmosphere pressure.
4. The compression and expansion lines are adiabatics.
5. The explosion line CD and the release line EB are constant-volume lines. At release the same result would be obtained if the gases were simply cooled instead of being exchanged for a new charge.

6. The working substance is 1 lb air with constant specific heats.

7. The heat added at C is the heat value of the fuel in the cylinder. The quantity of fuel is considered so small as not to dilute the working substance, air.

8. There are no losses due to radiation, friction, or other causes.

75. Heat Added during Constant-volume Combustion.—The heat added to the working substance at C is the heat value of the fuel mixed with the working substance. This heat quantity Q_1 might be found from a knowledge of the quantity of fuel consumed per combustion stroke. This heat is accounted for in the cycle of Fig. 69 by the heat quantity, in Btu.

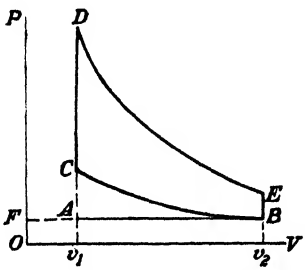


FIG. 69.—Idealized indicator card for Otto cycle.

$$Q_1 = c_v(T_d - T_c) \qquad (52)$$

76. Work Done during the Cycle. The work done during the theoretical Otto cycle may be found in two different ways:

1. If we subtract the work area under CB from that under DE, we have, from Eq. (38), the work of the ideal cycle in foot-pounds,

$$W_t = \frac{p_d v_d - p_e v_e}{k - 1} - \frac{p_c v_c - p_b v_b}{k - 1}$$

which, expressed in temperatures and Btu, becomes

$$W_t = \frac{RT_d - RT_e - RT_c + RT_b}{J(k - 1)}$$

$$= \frac{R}{J(k - 1)}[T_d - T_e - T_c + T_b] \qquad (53)$$

but since, by Eq. (30),

$$\frac{R}{J} = c_p - c_v$$

and, by Art. 60,

$$k = \frac{c_p}{c_v}$$

Then

$$\frac{R}{J(k - 1)} = \frac{c_p - c_v}{\dfrac{c_p}{c_v} - 1} = c_v$$

Therefore

$$W_i = c_v[T_d - T_e - T_c + T_b] \text{ Btu} \tag{54}$$

2. The work done by a gas during an adiabatic expansion is done at the expense of its own internal energy, and the work done equals the change in internal energy, see Art. 70. Then the change in internal energy along DE is

$$c_v(T_d - T_e) \tag{55}$$

which is equal to the work done *by* the gas in expanding. Again the change in internal energy along BC is

$$c_v(T_c - T_b) \tag{56}$$

which is equal to the work done *upon* the gas during compression. The net work is then the difference of Eq. (55) or (56), or

$$\begin{aligned} W_i &= c_v(T_d - T_e) - c_v(T_c - T_b) \\ &= c_v[T_d - T_e - T_c + T_b] \text{ Btu} \end{aligned} \tag{57}$$

as before.

77. Efficiency of Theoretical Cycle.—Since efficiency, from Art. 73, equals output divided by input, we have as the efficiency of the ideal cycle

$$e_i = \frac{W_i}{Q_1} = \frac{c_v(T_d - T_e - T_c + T_b)}{c_v(T_d - T_c)} \tag{58}$$

$$= \frac{T_d - T_c}{T_d - T_c} - \frac{T_e - T_b}{T_d - T_c} = 1 - \frac{T_e - T_b}{T_d - T_c} \tag{59}$$

Now, since DE and CB are adiabatics, from Art. 71,

$$\frac{T_e}{T_d} = \left(\frac{v_1}{v_2}\right)^{k-1} \qquad \frac{T_b}{T_c} = \left(\frac{v_1}{v_2}\right)^{k-1}$$

Therefore

$$\frac{T_e}{T_d} = \frac{T_b}{T_c} = \left(\frac{v_1}{v_2}\right)^{k-1}$$

and we may write, by the principle of subtraction,

$$\frac{T_e - T_b}{T_d - T_c} = \frac{T_e}{T_d}, \text{ or } = \frac{T_b}{T_c} \tag{60}$$

Substituting Eq. (60) in (59), we have

$$e_t = 1 - \frac{T_e}{T_d} = 1 - \frac{T_b}{T_c}$$

$$= 1 - \left(\frac{v_1}{v_2}\right)^{k-1} \tag{61}$$

And, if we let $v_2/v_1 = r$, the compression ratio, we have

$$e_t = 1 - \frac{1}{r^{(k-1)}} \tag{62}$$

This equation, Eq. (62), shows the very significant relation that the efficiency of an ideal Otto-cycle engine depends entirely

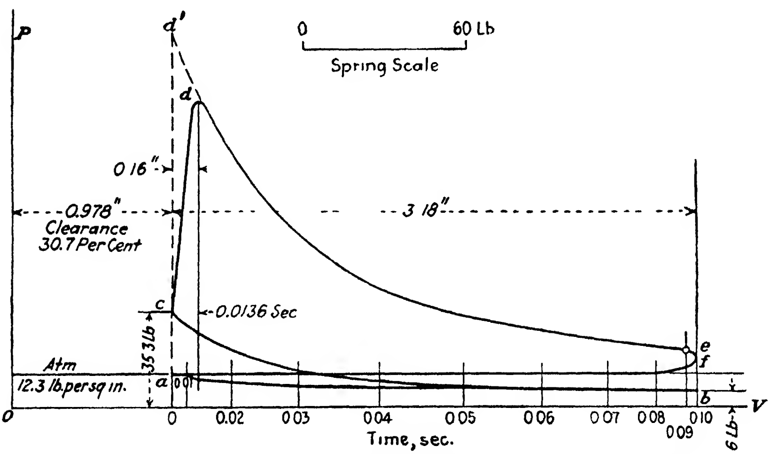


FIG. 70.—Actual indicator card from an Otto four-stroke engine.

upon the compression ratio. The efficiencies for various values of r are shown in Fig. 75. That the efficiency of an Otto engine increases with compression is borne out by the common experience that an automobile has more power when the "compression" is good than when it is weak due to leaky valves or piston rings or other causes.

78. Deviations of the Real from the Theoretical Cycle.—As was pointed out in Art. 5, the theoretical cycle must be modified in some of its details to make it practical. The ideal cycle of operations is likewise modified by imperfections and limitations of the practical machine. The deviations of the real cycle from the ideal are illustrated by Fig. 70, which shows an indicator card taken from an 8 by 11½-in. gasoline engine with 26-in. connecting rod running at 300 rpm.

1. The suction line *ab* lies below atmospheric pressure because of the inertia of the air and friction in passing through the inlet valve and passages. The engine from which the card of Fig. 70 was taken had a spring-loaded automatic inlet valve which made the pressure drop comparatively large.

2. The compression line *bc* is not an adiabatic but a polytropic whose exponent is about 1.3.

3. The explosion line *cd* is not vertical but slopes slightly because combustion requires a measurable time for its completion. For this particular card, the maximum pressure was not reached until the crank had turned through 13.6 per cent of its half revolution. Since 300 rpm equals 5 rps, the time for one crank revolution is 0.2 sec and for one stroke 0.1 sec. On this basis, the time scale has been laid off from the crank and connecting-rod diagram. Then combustion required 0.0136 sec if we can assume that the spark occurred exactly at dead center. The percentage of the stroke completed at *d* is $(100 \times 0.16)/3.18 = 5.03$ per cent.

4. The expansion line *de* is not an adiabatic but a polytropic whose exponent is about 1.3.

5. Release occurs slightly before lower dead center—in the engine of Fig. 70, about 0.01 sec or 18° before. It may be noted that the time required for the pressure to drop from that at *e* to atmospheric pressure was 0.03 sec instead of zero, as in the ideal cycle.

79. The Actual Working Substance.—In the ideal Otto cycle it was assumed that the working substance is air. In the real engine the working substance is not air but a mixture of air and the gases resulting from the combustion of the fuel. The composition of these combustion products was discussed in Chap. V. During the compression stroke, the substance in the cylinder is a mixture of fuel, air, and some spent gas which remained in the clearance space after the previous exhaust stroke.

In the theoretical cycle it was assumed that the specific heats of the working substance were constant. It has long been known that the specific heats of gases increase with increase in temperature, but the exact variation has not been conclusively determined for any of the common gases. Perhaps the most reliable data at the present time are those collected by Sweigert and Beardsley,[1] which have been plotted in Fig. 71. To obtain the change in energy of a gas from t_1 to t_2, it is necessary to integrate the area under the specific heat curve either mathematically or by approximate means.

[1] *Georgia State Eng. Exp. Sta., Bull.* 2, June, 1938.

Since 1 mole of a gas occupies a volume of about 378 cu ft at 60° and 30 in. Hg, the expansion per mole per degree equals $\frac{1}{520}$ of 378 = 0.727 cu ft. Then the external work done, measured in Btu, equals

$$\frac{p\Delta v}{77} = \frac{(30 \times 0.491 \times 144)(0.723)}{778} = 1.98 \text{ Btu}$$

That is, there should be a constant difference of 1.98 Btu between the molal specific heat at a constant pressure and that at constant

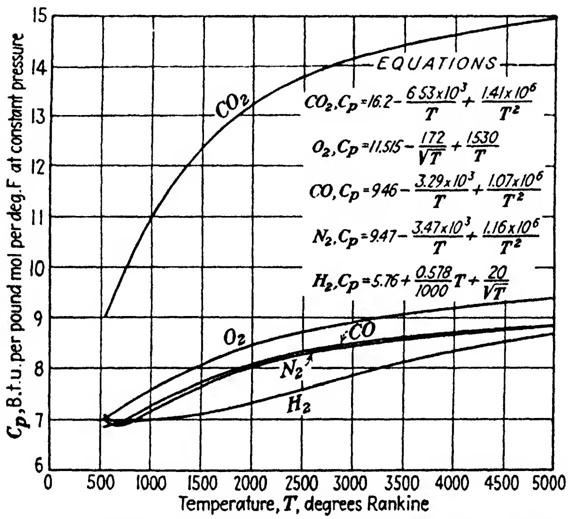


Fig. 71.—Molal specific heat at constant pressure of gases.

volume. This is, of course, the universal gas constant, Art. 67, in heat units.

THE DIESEL CYCLE

80. The Ideal Diesel Cycle.—An actual indicator card taken from a Diesel engine is shown in Fig. 72. In Fig. 73 is shown an ideal Diesel indicator card, for which the following assumptions are made:

1. The stroke *ab* is the same as for the actual card.
2. The clearance *fa* is the same as for the actual engine.
3. The suction and exhaust lines are coincident at atmospheric pressure.
4. The compression and expansion lines are adiabatics.
5. The combustion line *cd* is a constant-pressure line, while release *eb* is a constant-volume line.

6. The working substance is 1 lb air with constant specific heats.
7. The heat added along cd is the heat value of the fuel fed to the cylinder. The quantity of fuel is considered so small as not to dilute the working substance, air.
8. There are no losses due to radiation, friction, or other causes.

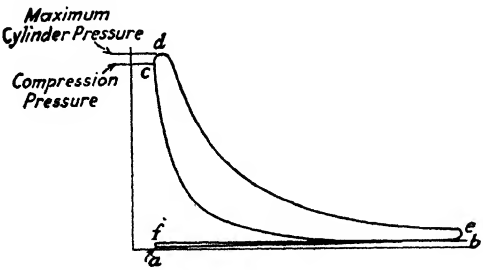


FIG. 72.—Actual indicator card from an air-injection Diesel engine.

81. Heat Added during Constant-pressure Combustion.—The heat added to the working substance along cd is the heat value of the fuel oil which is burned as it is sprayed into the cylinder and is found from the quantity of fuel consumed per

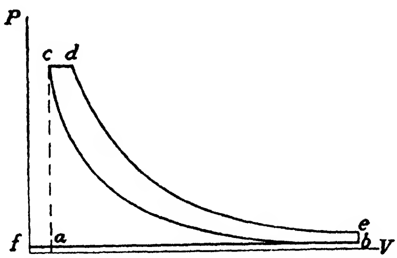


FIG. 73.—Idealized indicator card for a Diesel engine.

combustion stroke. This heat is accounted for in the cycle of Fig. 73 by the heat quantity, in Btu,

$$Q_1 = c_p(T_d - T_c) \tag{63}$$

82. Work Done during the Cycle.—The work done during the cycle may be determined in two different ways.

1. If we subtract the work area under cb from that under cde, we have, for the work of the ideal cycle, in foot-pounds,

$$W_i = \left[p_c(v_d - v_c) + \frac{p_d v_d - p_e v_e}{k - 1}\right] - \frac{p_c v_c - p_b v_b}{k - 1} \tag{64}$$

Now, letting $v_d/v_c = r_c$, the fuel cut-off ratio, and $v_b/v_c = r$, the air-compression ratio, and, remembering that $v_e = v_b$ and $p_d = p_c$ and substituting these in Eq. (64), we have

$$W_i = p_c(r_cv_c - v_c) + \frac{p_cr_cv_c - p_ev_e}{k-1} - \frac{p_cv_c - p_bv_b}{k-1}$$

which, expressed in temperatures and Btu, becomes

$$W_i = \frac{R}{J}T_c(r_c - 1) + \frac{R}{J(k-1)}(T_cr_c - T_e) - \frac{R}{J(k-1)}(T_c - T_b) \quad (65)$$

But since, as was shown in Arts. 70 and 67,

$$\frac{R}{J(k-1)} = c_v \text{ and } \frac{R}{J} = (c_p - c_v) = c_v(k-1)$$

we have, substituting in Eq. (65),

$$\begin{aligned} W_i &= c_v[(k-1)(r_c-1)T_c + T_cr_c - T_e - T_c + T_b] \\ &= c_v[k(r_c-1)T_c - (T_e - T_b)] \text{ Btu} \end{aligned} \quad (66)$$

2. We remember, from Eq. (51), that the work done in Btu $= Q_1 - Q_2$. Then Q_2 is the heat rejected at constant volume from e to b and

$$Q_2 = c_v(T_e - T_b)$$

The heat added during constant-pressure combustion is $Q_1 = c_p(T_d - T_c)$. Then the work done in heat units is

$$\begin{aligned} W_i &= Q_1 - Q_2 = c_p(T_d - T_c) - c_v(T_e - T_b) \\ &= c_v[k(T_d - T_c) - (T_e - T_b)] \end{aligned}$$

But since $T_d/T_c = v_d/v_c = r_c$, $T_d = T_cr_c$, and

$$W_i = c_v[k(r_c - 1)T_c - (T_e - T_b)] \quad (67)$$

as before.

83. Efficiency of the Theoretical Cycle.—Since efficiency equals output divided by input, we have as the efficiency of the ideal cycle

$$e_i = \frac{W_i}{Q_1} = \frac{c_v[k(r_c-1)T_c - (T_e - T_b)]}{c_p(T_d - T_c)}$$

But $c_v/c_p = 1/k$ and $T_d = r_c T_c$; hence

$$e_t = \frac{k(r_c - 1)T_c - (T_e - T_b)}{k(r_c - 1)T_c} = 1 - \frac{T_e - T_b}{k(r_c - 1)T_c} \tag{68}$$

Now, since de is an adiabatic, from Art. 71,

$$\frac{T_e}{T_d} = \left(\frac{v_d}{v_e}\right)^{k-1}$$

and

$$T_e = T_d\left(\frac{v_d}{v_e}\right)^{k-1} \tag{69}$$

Representing the compression ratio v_l/v_c by r, we have

$$v_b = v_e = rv_c$$

Substituting this value of v_e in Eq. (69) and remembering that $v_d = r_c v_c$,

$$T_e = T_d\left(\frac{v_d}{rv_c}\right)^{k-1} = T_d\left(\frac{r_c}{r}\right)^{k-1} \tag{70}$$

Again, we know, from Charles's law, that $T_d = T_c r_c$, and, substituting this in Eq. (70), we have

$$T_e = T_c r_c\left(\frac{r_c}{r}\right)^{k-1} \tag{71}$$

Further, since bc is adiabatic,

$$\frac{T_c}{T_b} = \left(\frac{v_b}{v_c}\right)^{k-1} \tag{72}$$

and

$$T_c = T_b r^{k-1} \tag{73}$$

which, substituted in Eq. (71), makes

$$T_e = T_b r^{k-1} r_c\left(\frac{r_c}{r}\right)^{k-1} = T_b r_c^k \tag{74}$$

Substituting these values of T_c and T_e in Eq. (68), we have

$$\begin{aligned} e_t &= 1 - \frac{T_b r_c^k - T_b}{k(r_c - 1)T_b r^{k-1}} \\ &= 1 - \frac{1}{r^{k-1}}\left[\frac{r_c^k - 1}{k(r_c - 1)}\right] \end{aligned} \tag{75}$$

It will be noted that this equation has the same form as Eq. (62) for the Otto cycle except for the last factor in the bracket. This indicates that the efficiency of the constant-pressure Diesel cycle increases as the cut-off ratio decreases. In other words, the theoretical efficiency is greatest at light loads. Since the Otto cycle cannot use a compression ratio so high as the Diesel cycle, it is reasonable to suppose that the latter would have the higher efficiency.

The deviations of the real Diesel from the ideal cycle are illustrated in comparing Fig. 73 with Fig. 72. In addition to the deviations as pointed out for the actual Otto cycle, the Diesel cycle has the curved and drooping combustion line, owing to the

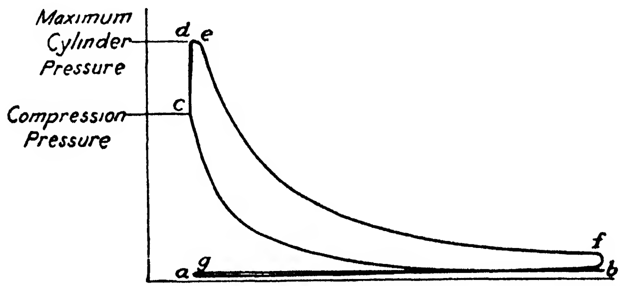


FIG. 74. Indicator card from an Ingersoll-Rand, type "PO," Diesel engine

fact that it is impossible to inject the fuel at just the proper rate to keep the pressure on the receding piston constant.

During the compression stroke the working substance is very nearly pure air because the cylinder is scavenged at the end of the exhaust.

84. Other Diesel Cycles.—If we could inject the fuel at just such a rate that the temperature in the cylinder would remain constant instead of the pressure's remaining constant, we should have isothermal combustion. The theoretical efficiency for this cycle is slightly greater than for constant pressure, but the capacity of the engine is less because the card area is smaller. In practice the actual combustion line is probably between isothermal and constant pressure.

Another cycle that has been used with much success is that shown in Fig. 74. In this cycle, the compression is carried to only 300 or 350 lb, and then the fuel is injected earlier and more quickly than in Fig. 72 so as to obtain as nearly constant volume combustion as possible. From c to d combustion is at constant volume, but there is a slight amount of combustion from d to e which is practically at constant pressure. Ideally, this cycle is a combination of the Otto and Diesel cycles and is variously called "dual combustion," "medium compression," or "semi-Diesel." As might be expected, the ideal efficiency of the dual combustion cycle lies between that of the Diesel and Otto cycles as shown in Fig. 75.

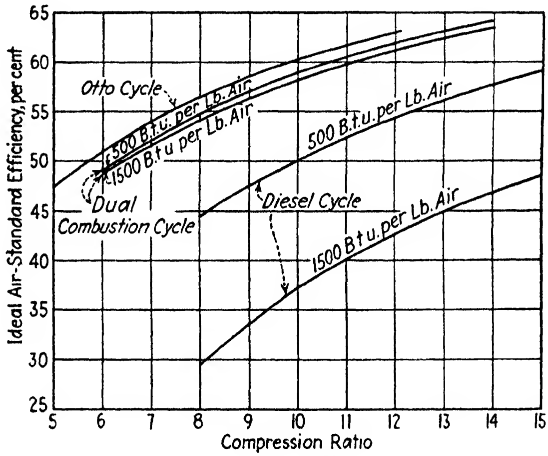


FIG. 75. Comparison of gas-engine cycle efficiencies.

85. Comparison of the Theoretical Cycles.—In Fig. 75 are plotted curves showing the variation in the theoretical efficiency of three gas-engine cycles with increasing compression. It should be especially noted that for a given ratio the Otto efficiency is always higher than that of the Diesel, but in practice the danger of preignition limits the compression in the Otto engine. However, the increasing use of high octane, Art. 39, fuels is making possible higher compression ratios each year. One 1938 car uses a ratio of 6.7 to 1. Figure 76 gives a historical summary of the characteristics of American automobile engine practice.[1]

The efficiencies for the dual combustion cycle were calculated on the assumption that half the heat input was at constant volume as in the Otto cycle and half at constant pressure as in the Diesel cycle.

[1] *Automotive Ind.*, annual statistical data.

In the air-standard cycles no account is taken of the presence of the fuel and the combustion products, of the change in specific heat of these substances during the cycle, or of combustion equilibrium temperatures. Goodenough and Baker have made a "real mixture" analysis[1] of these cycles on the basis of which the theoretically attainable efficiencies are considerably reduced.

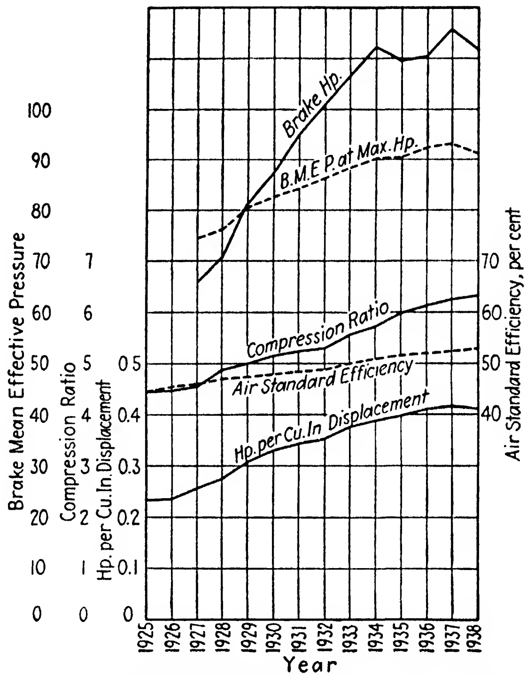


FIG. 76.—Average characteristics of American automobile engines.

Problems

1. What would be the theoretical horsepower of an ideal, single-cylinder, four-stroke, Otto-cycle gas engine having a 10-in. bore and a 15-in. stroke and running at 250 rpm? Assume 4.5:1 compression ratio; 14 lb per sq in. abs initial pressure; 60° initial temperature; and 1500 Btu supplied per pound of intake air.

2. Calculate the theoretical cylinder horsepower of a four-cylinder, four-stroke automobile engine having 3.75-in. bore and 4-in. stroke and an average piston speed of 1,000 ft per min. Assume a compression ratio of 5:1; gasoline to be C_8H_{18}; the theoretical quantity of air for combustion; and cycle idealized as in Art. 74.

[1] *Univ. Ill. Eng. Exp. Sta., Bull.* 160.

3. Calculate the theoretical cylinder dimensions of a single, tandem, double-acting, four-stroke, Otto gas engine to develop 500 shaft hp at 130 rpm. Assume mechanical efficiency of 90 per cent; gas composition to be CH_4, 25 per cent; H_2, 42 per cent; CO, 15 per cent; inert, 18 per cent; the theoretical quantity of air used for combustion; and cycle idealized as in Art. 74. Assume $P_b = 14.7$ and $P_c = 135$ lb per sq in. abs.

4. What would be the theoretical horsepower of an ideal, four-stroke cycle Diesel engine having a single cylinder with 10-in. bore and 15-in. stroke running at 250 rpm? Assume 15:1 compression ratio; 14 lb per sq in. abs initial pressure; 60° initial temperature; and 1500 Btu supplied per pound of intake air.

5. Calculate the theoretical cylinder horsepower of an ideal, three-cyclinder, four-stroke cycle Diesel engine having cylinder dimensions of 12 by 18 in. and running at 240 rpm. Assume a compression ratio of 15.1:1; fuel oil containing 84 per cent carbon and 16 per cent hydrogen; theoretical quantity of air used for combustion; cycle idealized as in Art. 80.

CHAPTER IX

ACTUAL PERFORMANCE OF INTERNAL-COMBUSTION ENGINES

86. Economy.—Perhaps the most common expression for the economy of a gas engine is "miles per gallon" of gasoline of an automotive engine. This expression has the obvious disadvantages (1) that not all miles require the same amount of energy and (2) that not all gallons of gasoline possess the same energy. A better way of stating the economy would be to give the *pounds of fuel per horsepower per hour.* For engines using the same kind of fuel this gives a good comparison, and even for engines using different fuels, it offers a simple basis for comparison of costs of energy. For example, a Diesel engine delivering 1 hp-hr output for 0.5 lb fuel oil is more economical than a gasoline engine that uses 1.0 lb gasoline per horsepower-hour.

Gas-engine economies are on the basis of the number of cubic feet of fuel gas under standard conditions of temperature and pressure per horsepower-hour.

Fuel economy may be stated in terms of indicated or brake horsepower.

Since the various gas-engine fuels are not comparable directly, a more satisfactory statement for economy would be in terms of Btu supplied per horsepower-hour. For this purpose, the higher heat value of the fuel is specified by the A.S.M.E., while in Europe the lower heat value is used. The justification for the use of the higher heat value is found (1) in the uncertainty as to just how much heat, in order to determine the lower value, should be deducted for that carried away by the moisture formed by the combustion of hydrogen, see Art. 42; and (2), in that the engine should be penalized, in comparison with other types of heat engines, if it cannot utilize the heat held by the moisture in the exhaust gases.

The *heat economy* is obtained by multiplying the fuel economy by the higher heating value of the fuel. Thus the heat economy of the Diesel engine using 0.5 lb fuel oil per horsepower-hour, if the heat value of the fuel oil is 19,000 Btu per lb, is

$$0.5 \times 19{,}000 = 9500 \text{ Btu per hp-hr}$$

87. Efficiency of Internal-combustion Engines.—If the heat economy of an engine be known, the efficiency of the engine as a machine for converting heat into mechanical energy is obtained by dividing the heat equivalent of 1 hp-hr by the heat economy per horsepower-hour. That is,

$$\text{Efficiency} = \frac{\text{output}}{\text{input}} = \frac{\text{heat equivalent of 1 hp-hr (Btu)}}{\text{heat economy per horsepower-hour (Btu)}}$$

Thus, in the example of the last article, the efficiency would be

$$\frac{2{,}543}{9{,}500} = 0.268 \text{ or } 26.8 \text{ per cent}$$

This efficiency is called the *heat efficiency* or *thermal efficiency* and may be based upon either indicated or brake horsepower. In the first case it is referred to as "indicated thermal efficiency" and in the second case as "brake thermal efficiency" or "over-all thermal efficiency."

It should be clearly understood that the efficiency referred to here is the actual efficiency as determined by measuring the fuel consumption and the engine output, while the efficiencies calculated in the last chapter were the theoretical efficiencies of the ideal cycles. The ratio of the actual efficiency to the ideal cycle efficiency is called the *engine efficiency* and is the measure of the ability of the engine to approach performance in accordance with its ideal cycle.

88. Performance of Typical Engines.—Table V gives the economies and efficiencies of several typical internal-combustion engines. The data given are for full load at normal speed.

89. Heat Losses.—The reason that the engine efficiency is not 100 per cent is that there are several losses of heat energy during the actual cycle. It is often of importance to be able to calculate each of these losses from the test data either as a check on the efficiency calculations or for other purposes in connection with the design or operation of the machine. These losses and the methods for calculating them are discussed in the following paragraphs.

The bases used will be 1 lb fuel for liquid fuels and 1 cu ft for gaseous fuels, since these are the units for which the heating values are usually determined.

1. *Heat Carried Away by the Dry Exhaust Gases.*—The exhaust gases leave the engine at temperatures varying from 500 to 1000° or even more and therefore carry away a large part of the heat of the fuel. The magnitude of this loss is

$$L_s = W_g c_{pm}(t_g - t_a) \tag{76}$$

in which L_s = heat loss per pound of fuel due to sensible heat of exhaust gas.

W_g = weight of exhaust gas per pound of fuel as determined by Orsat analysis or by measuring the air supply.

c_{pm} = the mean specific heat of the dry exhaust gases at constant pressure for the range $(t_g - t_a)$; see curves of Fig. 71.[1]

t_g = temperature of exhaust gases.

t_a = temperature of the atmosphere.

2. *Heat Carried Away by Moisture Formed by the Combustion of Hydrogen.*—The moisture in the exhaust is in the form of superheated steam and in this state possesses a large amount of heat per pound. Now,

if L_m = heat loss per pound of fuel due to superheated moisture in exhaust.

H = percentage of hydrogen in fuel.

we have, since the weight of water formed is nine times that of hydrogen,[2]

$$L_m = 9\frac{H}{100}[(1{,}058 + 0.46t_g - (t_a - 32)]$$

$$L_m = 9\frac{H}{100}[1{,}090 + 0.46t_g - t_a] \tag{77}$$

[1] For accurate work, the heat carried away by each gas should be computed separately by multiplying the weight of each component by its own mean specific heat for the given temperature range. For most work, however, it is accurate enough to consider the gas as all nitrogen and to assume c_{pm} to be 0.24.

[2] For explanation of the term in brackets see Art. 226. Also *Combustion,* April, 1937, p. 41.

3. *Heat Loss Due to Carbon Monoxide in the Exhaust.*—As pointed out in Art. 43, some carbon atoms may not find their full quota of oxygen. The heat loss due to this incomplete combustion of C is

$$L_{co} = \left(\frac{CO}{CO_2 + CO}\right)\frac{C}{100}(14{,}540 - 4{,}380) \tag{78}$$

in which L_{co} = heat loss per pound of fuel due to CO

C = percentage of carbon in fuel.

$\frac{CO}{CO_2 + CO}$ = fraction of the carbon ineffectively burned.

CO and CO_2 are volume percentages of these gases by Orsat analysis.

4. *Heat Loss Due to Cooling Water.*—The heat carried away by the jacket water per pound of fuel is

$$L_w = W_w(t_o - t_i) \tag{79}$$

in which W_w = weight of cooling water flowing through the jacket per pound of fuel.

t_i = inlet temperature of jacket water.

t_o = outlet temperature of jacket water.

5. *Radiation and Friction.*—The heat loss due to radiation from the cylinder and piston and to friction cannot be determined directly but must be found by difference, as shown in the following article.

90. Heat Balance.—The heat supplied to the engine by the fuel should be accounted for by the sum of the useful work plus the losses. The heat converted into useful work per pound of fuel equals the work in foot-pounds performed per pound of fuel, divided by 778.

Table V gives the heat balance for several typical engines. In each case the radiation and friction losses were found by subtracting the sum of the known losses and the useful work from the heating value of the fuel.

91. Variation in Engine Performance with Load.—It was shown, in Art. 59, that the mechanical efficiency varies with the engine output. As the load is increased on either the Otto or the Diesel engine, the pressure and temperature of the working substance become greater, and, therefore, since the temperature of

TABLE V.—PERFORMANCE DATA OF GAS ENGINES

Engine no.	Bhp load	Higher heat value, Btu per lb or cu ft	Fuel economy, lb or cu ft per bhp-hr	Heat balance		
				Useful shaft work, per cent	Jacket loss, per cent	Exhaust and radiation, per cent
1	109	19 632	0 441	29 49	25 08	45 47
2	482	19,568	0 414	29 0	32.0	39 0
3	600		0 422	32 7	33.1	34 2
4	2,927	18,500	0.473	29 08		
5	499	475 5	18 21	29 39		
6	3,400[1]	105	143[2]	28.6[3]	28.0	43 4
7	10,920[1]	18,233	0 521[2]	35 7[3]	26.0	29.2
8	1,728	1,254	7 77	26 0		

[1] Kilowatts. [2] Per kilowatt-hour [3] Useful electrical output

1. 17¾ by 19-in. Ingersoll-Rand PO single-cylinder, horizontal, four-stroke, 110-hp, 257 rpm Diesel.

2. 500-hp McIntosh & Seymore, four-cylinder, vertical, four-stroke, 164 rpm Diesel.

3. 17¾ by 22-in. Ingersoll-Rand PR five-cylinder, vertical, four-stroke, 600-hp, 240 rpm Diesel.

4. 28 by 40-in. Worthington, four-cylinder, vertical, double-acting, two-stroke, 2,900-hp Diesel.

5. 21½ by 36-in. Cooper, two-cylinder, horizontal, single-tandem, double-acting, four-stroke, 500-hp, manufactured-gas engine, 133 rpm.

6. 47 by 60-in., four-cylinder, horizontal, twin-tandem, double-acting, four-stroke, blast-furnace-gas engine, 88.3 rpm (heat balance based on indicated horsepower).

7. 33⅞ by 59-in. M.A.N., nine-cylinder, two-stroke, double-acting Diesel (friction blower and compressor use 9.1 per cent).

8. 17 by 24-in., twin, single-acting, four-stroke, 170-hp, natural-gas engine, 180 rpm, see Fig. 7.

the exhaust gases is higher, the heat carried away is greater. At higher loads, likewise, the heat carried away by the cooling water and lost in radiation is greater than at lighter loads. This variation in losses indicates that the economy and efficiency vary with the load on the engine. In general, the performance improves up to about full load and then begins to decline. Figures 77 and 78 show the variation in performance of two typical engines.

92. Variation in Engine Performance with Speed.—Some internal-combustion engines are designed to run at a definite speed which is maintained within narrow limits by the governor. Other engines, such as those for marine, automotive, and aviation

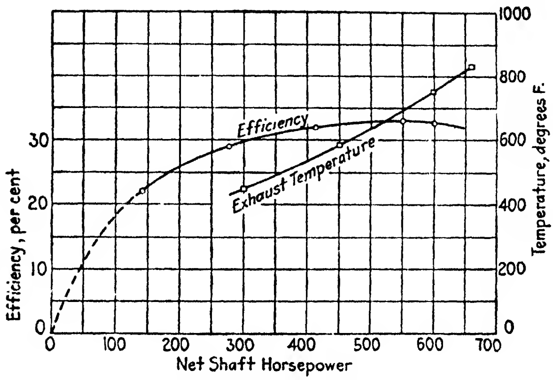


FIG. 77.—Thermal efficiency of an Ingersoll-Rand PR engine.

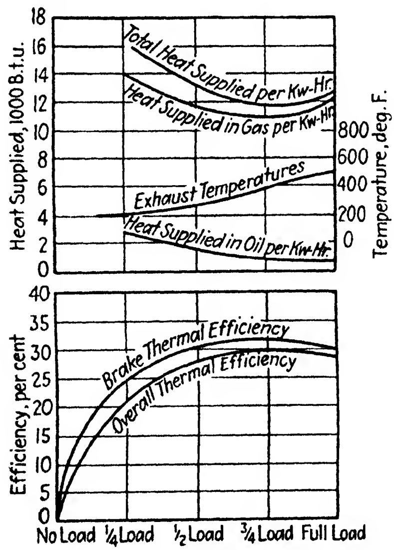
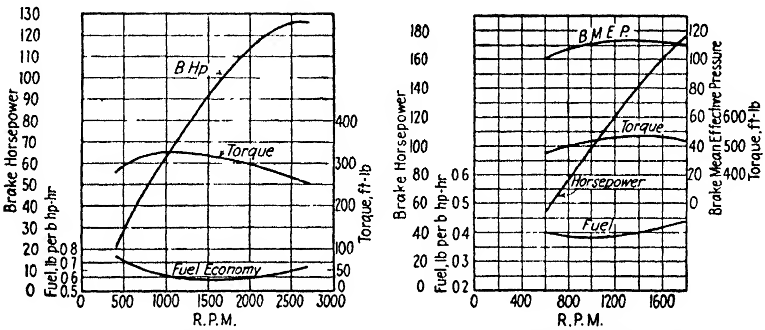


FIG. 78.—Performance curves of Nordberg dual-fuel Diesel (Fig. 14).

uses, are designed to operate at varying speeds. Since the heat lost by transmission to the cylinder walls is a function of the time required for the completion of each cycle, it is reasonable to expect that generally the efficiency of a variable-speed engine should increase with the speed. Figures 79, 80, and 81 show the variation in performance of three typical variable-speed engines with change in speed. Figure 80 gives the performance curves (*a*) and a typical indicator card (*b*) for the Allis-Chalmers, fuel-injection, spark-ignition engine of Fig. 18. A typical Diesel indicator card for a compression ratio of 16:1 is compared with the low-pressure, oil-engine card.



(*a*) Waukesha gasoline engine (Fig. 8). (*b*) Hercules Diesel engine (Fig. 15).

FIG. 79.—Performance characteristics of two automotive engines.

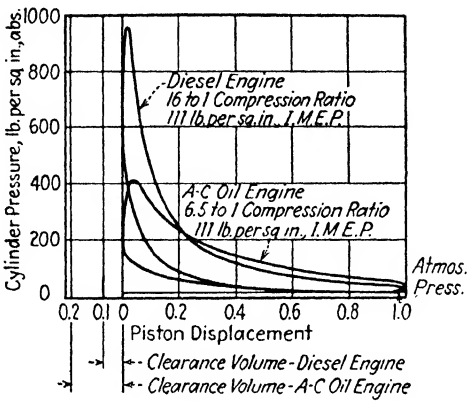
93. Rating of Internal-combustion Engines.—The horsepower capacity of an Otto engine or a Diesel engine is determined by the quantity of fuel that can be burned in a given time. The quantity of fuel, in turn, is limited directly by the volume of air drawn into the cylinder during the suction strokes. In other words, the displacement of the piston measures the horsepower capacity of the engine.

It was shown in Art. 57 that the indicated horsepower equals $\frac{1}{2}(p_m LAN) \div 33{,}000$ per cylinder for a four-stroke, single-acting engine. Now $A = \pi d^2/4$; also $2LN$ equals the average piston speed in feet per minute in which N = rpm. For rating purposes the S.A.E. assumes a piston speed of 1,000 ft per min and a mean effective pressure of 67 lb per sq in. for a gasoline engine. Substituting these values in the above equation and letting n represent the number of cylinders, we have

$$\text{S.A.E. rating} = \frac{1}{2}(67)\left(\frac{\pi d^2}{4}\right)\left(\frac{1{,}000}{2}\right)\frac{n}{33{,}000}$$

$$= \frac{d^2 n}{2.5} \qquad (80)$$

The disadvantages of the S.A.E. rating formula are that it does not take account of (1) quality of fuel; (2) temperature of



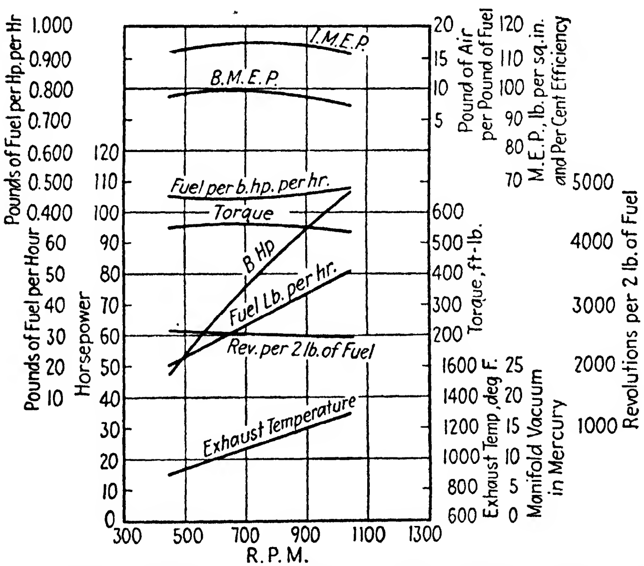


FIG. 80.—Indicator cards and performance curves of Allis-Chalmers Oil Engine (Fig. 18).

air supply; (3) barometric pressure; (4) compression ratio; (5) nature of cooling system; (6) effectiveness of valves and passages in admitting and discharging gases—all of which affect the mean effective pressure—and, also, that it assumes 1,000 ft per min piston speed regardless of the ratio of the bore to stroke or the maximum rpm of the engine. In spite of these disadvantages, the S.A.E. formula is used for determining the horsepower rating of automobiles for licensing purposes in many states. Since six- and eight-cylinder engines have a more uniform flow

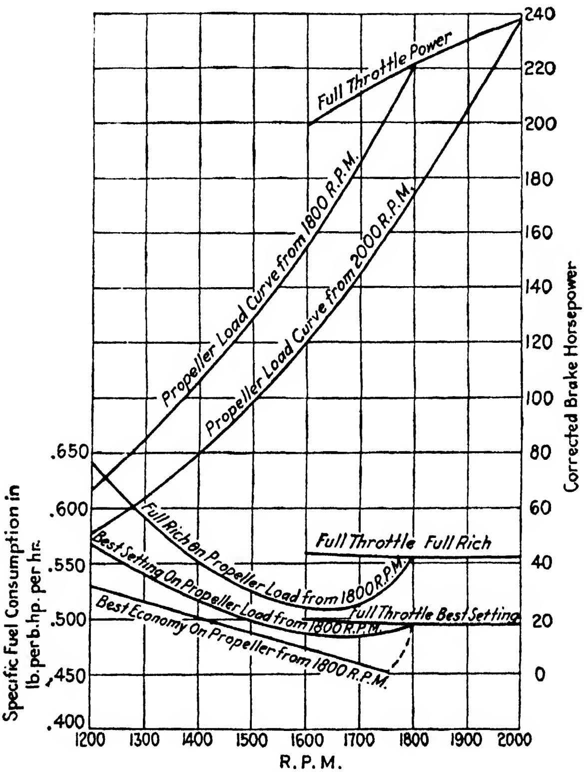


FIG. 81.—Performance curves of Wright Whirlwind engine (Fig. 11).

of power than four-cylinder ones, their speed is generally higher, and this gives them an advantage over the fours, in that the actual power is greater than the rating.

It is important, however, to note that the horsepower capacity of a given engine for a given set of conditions as to fuel, barometer temperature, etc., is very definitely limited by its piston displacement. For this reason the designer generally allows a margin of about 15 per cent additional cylinder volume over that required for the rated capacity in order to provide for some variations in these items and to provide a small overload capacity. While some prime movers, such as hydraulic turbines, steam engines, or steam turbines, may be made to carry overloads for short periods by admitting more than the normal economical rated quantity of water or steam, an internal-combustion engine has no appreciable overload capacity.

Otto and Diesel engines are ordinarily rated in terms of their brake or shaft horsepower. Variable-speed engines are rated at some assumed definite speed.

Problems

1. During a test run, a gas engine used each hour 481,000 cu ft gas, having a heating value of 105 Btu per cu ft. The engine drives a generator and the generator output was 3,400 kw. What was (*a*) the fuel economy; (*b*) the heat economy; (*c*) the thermal efficiency of the engine-generator unit?

2. A Diesel engine, during a test, used 330 lb fuel oil having 19,000 Btu per lb. The net brake load on the engine was 248 hp and the run 3 hr long. What was (*a*) the fuel economy; (*b*) the heat economy; (*c*) the thermal efficiency of the engine?

3. A gas engine used 6,880 cu ft gas per hour while carrying a load of 355 bhp. The gas has 495 Btu per cu ft. What was (*a*) the fuel economy; (*b*) the heat economy; (*c*) the thermal efficiency of the engine?

4. An automobile runs 15 miles on a gallon of gasoline when running at 30 miles per hr. At this speed, the shaft horsepower of the engine is 27.7. If gasoline be assumed to have a heating value of 20,000 Btu per lb and a specific gravity of 0.73, what is (*a*) the fuel economy; (*b*) the heat economy; and (*c*) the thermal efficiency of the engine?

5. The J-5 Whirlwind engine is guaranteed to give a brake horsepower-hour at rated load for not more than 0.6 lb gasoline. If the gasoline has a heating value of 20,000 Btu per lb, what is (*a*) the heat economy; (*b*) the thermal efficiency guaranteed by the builder?

6. A Diesel marine engine on a block test developed an average of 2,912 bhp and 4,095 ihp. During the 24-hr run, 32,200 lb fuel oil having a heating value of 18,500 Btu per lb was consumed. Compute (*a*) the mechanical efficiency; and for both indicated and brake horsepower, (*b*) the fuel economy, (*c*) the heat economy, and (*d*) the thermal efficiency.

7. A Diesel is supplied with 100 lb fuel oil having 19,600 Btu per lb and runs 24 min 5 sec when developing 590 hp at 235 rpm. The net brake load is 1,505.0 lb. Calculate (*a*) the brake constant; (*b*) the length of the brake arm; (*c*) the fuel economy; (*d*) the heat economy; (*e*) the thermal efficiency of the engine.

8. A series of tests on a Diesel engine direct connected to an alternator gave the following data:

Run number	1	2	3	4
Kilowatts load, average	340.6	168.9	253.3	86.1
Duration of test, hours	3.816	3.417	3.517	1.183
Fuel consumption, pounds	783	396	543	98.5
Generator efficiency, per cent	92.6	89.5	91.8	83.0

The fuel oil had a heating value of 19,500 Btu per lb. Calculate for each run (*a*) engine-shaft horsepower; (*b*) fuel economy; (*c*) heat economy; (*d*) thermal efficiency based on shaft horsepower. (*e*) Plot curves with shaft horsepower as abscissas and (1) fuel consumer per hour, (2) fuel consumed per shaft horsepower hour, and (3) thermal efficiency as ordinates.

9. A single cylinder, 17 by 19-in. four-stroke cycle, Diesel engine was subjected to a series of tests with a brake having an arm 7.86 ft. long. The following data were taken:

Run number	1	2	3	4	5
Net brake load, lbs	325	260	197	130	65
R.p.m	256	257	256	258	258
Oil per hr., lbs	51.4	38.15	28.9	19.96	13.0
Mean effective pressure	98.6	79.9	62.0	42.8	24.85

Calorific value of the fuel oil 19,632 Btu per lb. Calculate for each run (*a*) brake horsepower; (*b*) indicated horsepower; (*c*) mechanical efficiency; (*d*) indicated-horsepower fuel economy; (*e*) indicated-horsepower heat economy; (*f*) indicated-horsepower thermal efficiency. (*g*) Plot curves with brake horsepower as abscissas and (*c*), (*d*), (*f*), and total fuel per hour as ordinates.

10. During a series of tests on an automobile engine, the following data were taken:

Run number	1	2	3	4	5
R.p.m	1,200	1,298	1,500	1,600	1,700
Gasoline per hr., lbs	15.8	16.3	16.9	18.7	22.8
Brake horsepower	21.5	22.8	25.5	28.7	34.1

Assuming 19,500 as the heating value of the gasoline, calculate for each run (*a*) the fuel economy; (*b*) the heat economy; (*c*) the thermal efficiency. (*d*) Plot horsepower and economy as ordinates against speed as abscissas.

11. A $17\frac{3}{4}$ by 22-in., five-cylinder, four-stroke cycle, semi-Diesel engine was subjected to a series of brake tests, and the following data were taken:

Run number	1	2	3	4	5
Net brake load, lbs.	1,512.5	1,375	1,030	687.5	344
R.p.m.	239	239	240	242	246
Fuel oil used, lbs.	100	200	100	50	40
Duration of test	23′ 40″	52′ 20″	33′ 45″	22′ 58″	27′ 18″
Jacket water inlet, °F.	38	38	38	38	38.5
Jacket water outlet, °F.	113	106	96	93	81
Jacket water g.p.m.	41.3	41.3	39.1	33.1	33.6

Length of brake arm 8.7 ft. Fuel oil heating value 19,600 Btu per lb.

Calculate for each run (*a*) brake horsepower; (*b*) fuel economy; (*c*) heat economy; (*d*) thermal efficiency; (*e*) heat carried away by jacket water in Btu and per cent. (*f*) Plot against brake horsepower as abscissas (1) rpm, (2) thermal efficiency, (3) jacket-water loss, per cent.

12. A 600 rated horsepower Diesel engine gave the following data in a series of runs.

Run number	1	2	3	4
Brake horsepower	300	450	600	660
Fuel oil per b.hp-hr.	0.436	0.408	0.408	0.420
Exhaust temperature, °F.	455	585	745	830

Assuming the fuel oil to consist of carbon, 84 per cent, and hydrogen, 16 per cent, and that 10 per cent excess air is used in the engine at full-load operation, about what would be the heat carried away by the dry exhaust gases and by moisture due to combustion of hydrogen, at each load?

13. The following data were taken during a series of five brake tests on a 110-hp, semi-Diesel engine:

Run number	1	2	3	4	5
Net brake load, lbs.	258	234.5	176	117.2	58.6
R.p.m.	254.3	256.2	262.5	262.2	260.2
Fuel oil per hr., lbs.	48.0	42.4	31.9	23.6	15.2
Jacket water inlet, °F.	130.6	123.7	118.9	113.1	106.0
Jacket water outlet, °F.	159.7	152.5	140.0	131.0	120.0
Jacket water g.p.m.	16.15	14.63	15.95	14.61	14.28

Brake arm is 9.7 ft long. Heating value of the fuel oil was 19,630 Btu. Complete the tabulation by calculating for each run (*a*) fuel economy; (*b*) heat economy; (*c*) brake thermal efficiency; (*d*) heat carried away by jacket water in Btu and percentage. (*e*) Plot against brake horsepower as abscissas (1) total fuel per hour, (2) thermal efficiency, (3) percentage jacket loss.

14. A blast-furnace gas engine gave the following data during a test: (1) indicated horsepower 1,673, (2) cubic feet of fuel gas per hour 145,000, (3) heating value of fuel gas 105 Btu per cu ft, (4) temperature of exhaust gases 900°F., (5) quantity of exhaust gases 27,000 lb per hr, (6) jacket cooling water, 7,950 gal. per hr, (7) temperature rise of jacket water from 85 to 145°F.

Calculate the heat balance showing the Btu and percentage of the heat value of the gas converted into (1) cylinder work, (2) jacket loss, (3) exhaust gas loss, (4) radiation and unaccounted for.

Assume specific heat of exhaust gases to be 0.25 and room temperature 60°F.

CHAPTER X

THE STEAM-POWER PLANT

94. The Steam Engine.—The first part of this book has been devoted to the discussion of heat engines of the internal-combustion type. The remainder will deal with the two chief external-combustion prime movers in practical use, together with the

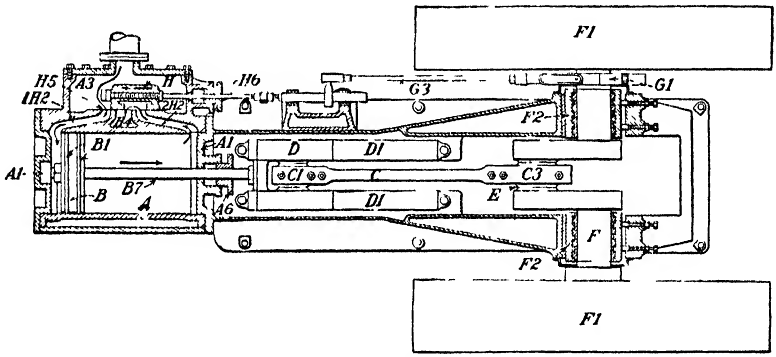


FIG. 82. Sectional view of simple, slide-valve steam engine

various auxiliary apparatus necessary for their operation. These two prime movers are the steam engine and the steam turbine.

The reciprocating steam engine was the earliest of the heat engines to be developed and for many years occupied the heat-power field nearly alone. In spite of the keen competition of the gas engine and the steam turbine, the reciprocating engine still retains the leadership in the matter of simplicity and reliability.

The steam engine consists of a cylinder in which a piston moves to and fro, the reciprocating motion of the piston being transformed into rotational motion by means of a crank and connecting rod. Figure 82 shows a "slide-valve" engine which is one of the earliest and simplest types.

The piston *B*, fitted with piston rings B1 to reduce steam leakage, moves in the cylinder *A*, communicating its motion to the piston rod *B*7 and to the crosshead *D* which moves in the guides *D*1. The gland *A*6 and stuffing box are provided to avoid steam leakage at the point where the piston rod $B7$ passes through the crank-end cylinder head $A1$.

The connecting rod C is connected to the crosshead by a crosshead pin at $C1$ and to the crankpin E at $C3$. The crankpin E forms part of the crankshaft F, which is mounted in substantial bearings $F2$ and carries the flywheels $F1$, which may be used for a belt drive to the driven machine.

Steam distribution is accomplished by means of the slide valve H, operated by the valve stem $H6$, which enters the valve chest $A3$ through a stuffing box and gland. The valve, in moving to and fro, covers and uncovers steam ports $1H2$ and $2H2$, distributing the high-pressure steam to the cylinder ends.

The motion of the slide valve is reciprocating, parallel to the motion of the piston, but considerably in advance of it.

The steam chest generally forms part of the steam-engine cylinder casting, thus permitting short steam passages between the steam chest and the steam cylinder.

As the steam passages $1H2$ and $2H2$ are alternately uncovered by the ends of the valve H, steam is permitted to pass from the steam chest $A3$ into the cylinder A. The resulting steam expansion pushes the piston B to and fro in the cylinder.

When steam passes into the cylinder A through passage $2H2$, the piston B is pushed to the left. The exhaust steam on the left of the piston B is forced out through passage $1H2$, the hollow slide valve H and the large central opening $H4$ communicating with the exhaust pipe (not shown).

When steam passes into the cylinder A through passage $1H2$, the piston B is pushed to the right. The exhaust steam on the right of the piston is forced out through passage $2H2$, the valve H, and the large central opening $H4$.

Improved types of the reciprocating steam engine are described in Chap. XIV, but for our present purpose the principle of operation of all of them is essentially the same as that of the slide-valve type. Steam engines are generally double acting, that is, steam is admitted alternately to both ends of the cylinder. For this reason the front end of the cylinder must be enclosed with a cylinder head $A1$ and provided with a stuffing box $A6$ through which the piston rod passes. Some engines are made with two or more cylinders through which the steam passes in succession, thus converting a greater proportion of the energy of the steam into useful work. Such engines are called "compound" engines. A single-cylinder engine such as is shown in Fig. 82 is called a "simple" engine.

95. The Steam Turbine.—Historically the principle of the steam turbine is much older than that of the reciprocating engine. A simple form was used as early as 120 B.C. by Hero of

Alexandria, and a second simple form was described by Branca in 1629.

The principle upon which the steam turbine operates is that of a stream of steam moving at high velocity and impinging upon blades on a rotating disk or cylinder. Figure 83 shows a simple form of turbine. A disk *C* is provided on its periphery with a series of blades *E* and is mounted on a shaft *D* supported by the bearings *FF*. A nozzle *B* allows steam to pass from the chamber at the left and directs it at high velocity onto the blades

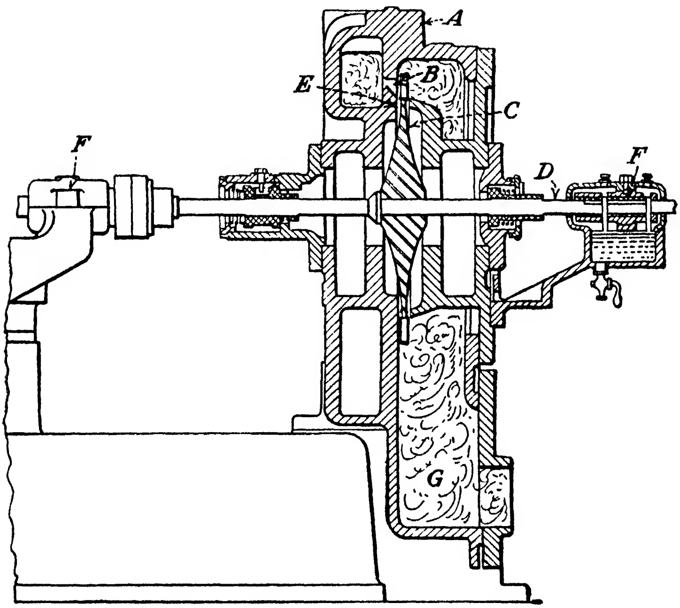


FIG. 83.—A simple, single-stage steam turbine.

E, causing the disk to turn and transmit power through the shaft *D*. After passing through the blades, the steam is exhausted into the chamber at the right of the disk and out of the opening *G*.

Several improved and more elaborate types of turbines are discussed in Chap. XV, but for our present purpose the essential principle, that of converting the energy of high-velocity steam into useful work, is the same in all of them. Most turbines have more than one disk carrying the moving blades in order to convert a greater proportion of the energy of the steam into useful work. Figure 83 shows a "single-stage" turbine.

Since the steam issues from the nozzle at high velocity, it is reasonable to suppose that the wheel would rotate at high speed, and this is the case. Two important distinctions between engines and turbines now are apparent: (1) The engine operates by virtue of the *pressure* of the steam upon the piston, while the turbine operates by virtue of the *velocity* of the steam acting upon the blades. (2) The steam engine is a *slow-speed* machine, while a turbine is a *high-speed* machine. Except in very small sizes, the speed of an engine is seldom higher than 250 rpm, while some of the very largest turbines operate at 1,800 rpm, and some of the smaller ones at several thousand revolutions per minute.

96. The Steam Plant As a Whole.—Neither steam engines nor turbines are complete in themselves, as are the internal-combus-

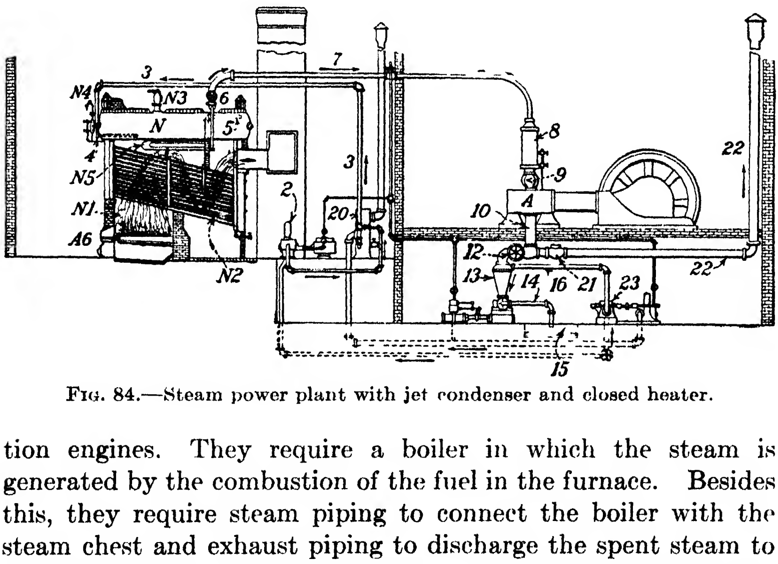


FIG. 84.—Steam power plant with jet condenser and closed heater.

tion engines. They require a boiler in which the steam is generated by the combustion of the fuel in the furnace. Besides this, they require steam piping to connect the boiler with the steam chest and exhaust piping to discharge the spent steam to the atmosphere or to the condenser. If the exhaust is to atmosphere, a supply of fresh water must be continuously pumped into the boiler by some form of boiler feeder. If a condenser is used, the condensed steam is returned to the boiler by the boiler feeder.

The arrangements of these pieces of equipment found in power plants are various. Figure 84 is a section of a steam-engine plant showing the essential features. The *boiler N* delivers steam through the pipe 7, the *separator* 8 (which removes any water carried along with the steam), and the throttle valve 9 to the *engine A*. The valve 12 may be closed so as to direct the exhaust steam through the relief valve 21 and the exhaust pipe 22 to the atmosphere, in which case the engine would be operated "noncondensing." If the valve 12 be opened, the exhaust will pass into the *jet condenser* 13 where it will be condensed on being sprayed with cold water from the supply pipe 1 by means of the *injection* pump 23. The condensate and injection water both pass out of the condenser into the sump 15 and are wasted.

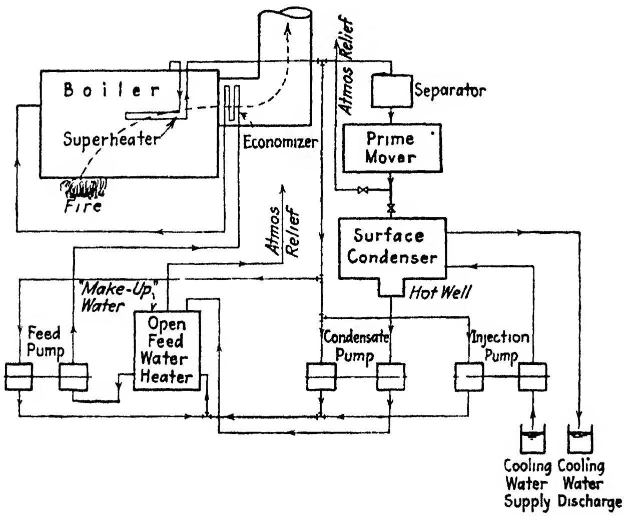


FIG. 85.—Flow diagram for water and steam in surface-condensing plant with open heater, economizer and superheater

The steam-driven reciprocating *pump* 2 draws cold water from the *supply pipe* and, after passing it through the *heater* 20, forces it into the boiler through the *feed line* 3. The *heater* 20 heats the feed water by means of the steam exhausted from the turbine driving the *injection pump* 23, from the engine end of the *removal pump* 14 on the condenser and from the engine end of the *boiler feed pump* 2.

The fuel is fed into the *furnace* at *N*1, and the hot gases find their way through the devious passes and out through the smoke pipe and up the *chimney*. In some plants an additional feed-water heater is placed in the smoke passage between the boiler and the chimney in order to salvage some of the heat left in the flue gases. The feed line 3 is detoured through this *economizer*, as it is called, on its way to the boiler.

The loop of pipe $N5$ is a *superheater* through which the steam may be passed after leaving the boiler in order to evaporate any moisture it may be carrying and to give some additional heat to the steam.

If it is desired to return the steam to the boiler after it is condensed, it is necessary to use a surface condenser in place of the jet condenser 13. In the *surface condenser*, the condensate and cooling water do not mix, but the cooling water passes through tubes, the outsides of which are exposed to the exhaust steam.

The heater 20, called a *closed heater*, is essentially the same as a surface condenser in construction. Some plants used feed-water heaters in which the waste steam comes into direct contact with the feed water. These are called *open heaters*.

If the engine A be replaced by a turbine, the prime mover becomes a high-speed instead of a slow-speed one, but, in all other respects, the plant might remain essentially the same.

Figure 85 shows a diagrammatic layout of a steam-power plant with a slightly different arrangement.

Each of the essential pieces of apparatus of the steam-power plant will be described in the chapters that follow. Before this is done, however, the properties of the working substance, steam, will be discussed in Chap. XI.

CHAPTER XI

STEAM

97. Effects of Adding Heat to a Liquid.—Suppose we have, Fig. 86, a vessel A having perfectly insulated sides and a perfectly conducting bottom through which heat may be transferred from the source C to 1 lb water inside. Let us, further, imagine a tight, frictionless piston resting upon the top of the water, which with the atmosphere exerts a total pressure of p lb per sq in. Suppose also that the temperature indicated by the thermometer t is 32°F.

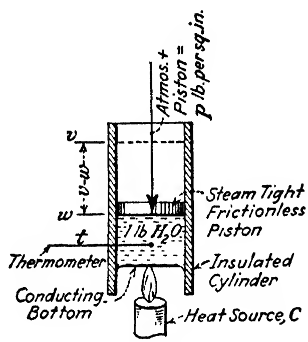


FIG. 86.

Then, as heat is given to the water, the following events take place:

1. The temperature of the water rises until the boiling point is reached. During this period, the quantity of heat supplied is absorbed in two ways: (a) The molecular activity of the water is increased; and (b) a small amount of external work is done. The volume of the liquid increases against the pressure p, and the *enthalpy* (see page 103) of the liquid water increases by an amount

$$h_f = e + \frac{p\Delta v}{J}$$

This increase from 32° is called the *enthalpy of saturated liquid*.

2. Evaporation of the water begins and continues until the water is entirely in the form of vapor, but during this period the temperature remains constant, being called the *saturation temperature* or temperature of evaporation. The heat added during evaporation is called *enthalpy of vaporization*. It consists of two parts, the *increase in internal energy*, used in changing the state from a liquid to a gas (sometimes spoken of as "disgrega-

tion" energy), and *the mechanical work,* done in expanding from water to steam against the constant pressure of p. In this case the mechanical work amounts to between 5 and 10 per cent of the enthalpy of vaporization, depending upon the pressure. The sum of the enthalpy of saturated liquid and the enthalpy of vaporization is called *enthalpy of saturated vapor.*

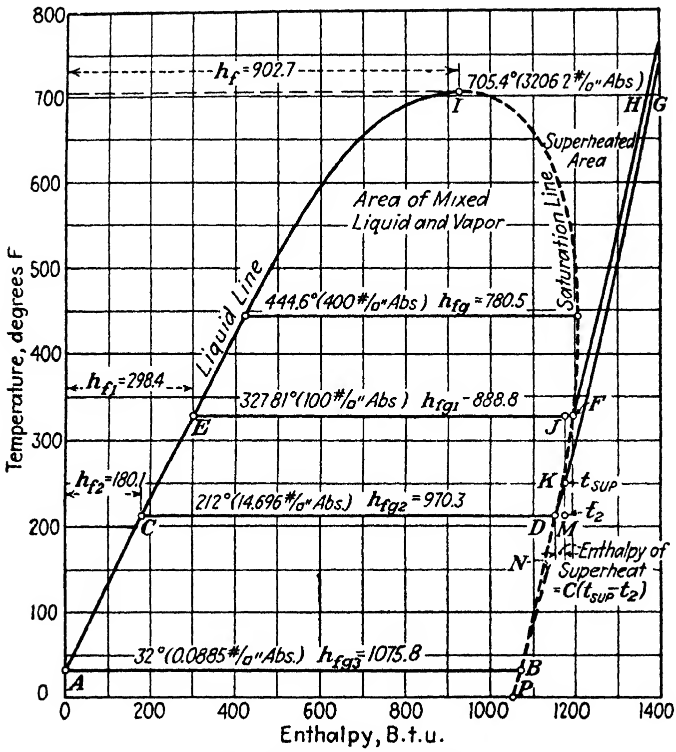


FIG. 87.—Temperature-enthalpy chart for steam.

3. When all of the water is evaporated, the temperature again begins to rise as more heat is added. (If the heating were carried to about 2500°F. and above, dissociation of the H_2O into H_2 and O_2 would take place. Since the engineering steam temperatures are below 1000°F., dissociation need not be considered.)

The heat added to the steam during this third period is called *enthalpy of superheat.* The rise in temperature above the "saturation temperature" of period 2 is called *superheat,* and the sum of the saturation temperature and the superheat is called the *total temperature*.

98. Graphic Representation of Evaporation.—These events are shown graphically in Fig. 87 in which Btu's added to the pound of water are plotted as abscissas against Fahrenheit temperature of the water as ordinates.

Beginning at point A with water at 32°, the temperature rises along curve ACE, which is nearly a straight line having a slope equal to the specific heat of water ($= 1$). Since the specific heat of water is not exactly constant and equal to unity, see Art. 60, the path curves slightly. If the pressure p is that of 1 standard atmosphere, 14.7 lb per sq in., the temperature will not rise above C, 212°F., but evaporation will begin and will follow line CD at constant temperature. When the entire pound of water is evaporated, D is reached, and, if heating is continued, the temperature will rise along line DKG. It will be noticed that the slope of the line DKG is about double that of line AC, the reason being that the specific heat c_p of superheated steam is about 0.5 for moderate temperatures and pressures, see Table VI, Art. 102.

If the final temperature is that of the state point K, we have, as the heat quantities added, for $p = 14.7$ lb per sq in.:

h_{f2} = the "enthalpy of saturated liquid" (at $t_2 = 212$°F.) = 180.1 Btu.

h_{fg2} (length of CD) = the enthalpy of vaporization (at $t_2 =$ 212°F.) = 970.3 Btu.

DM = the "enthalpy of superheat" at point K.

$(t_{\text{sup.}} - t_2)$ = degrees of superheat.

At a pressure of 100 lb per sq in., the path followed will be $AEFH$, and we have representing the heat quantities:

$$h_{f1} = \text{enthalpy of liquid} = 298.4 \text{ Btu}$$
$$h_{fg1} = \text{enthalpy of vaporization} = 888.8 \text{ Btu}$$

while the enthalpy of superheat is the horizontal projection of FH.

It will be observed that as the temperature (and with it the pressure) rises from 32° the enthalpy of vaporization diminishes continuously from 1075.8 Btu at 32° until it finally disappears at I, a temperature of 705.4°F. and a pressure of 3,206.2 lb per sq in.

This is called the "critical" temperature and pressure. Above this temperature and pressure there is no difference between the liquid and vapor states for water.

In an actual boiler there are tiny globules of water carried up with the bubbles of steam, which are held in suspension by the steam. In this case the heat added to 1 lb water during evaporation, for example, at 100 lb per sq in. (Fig. 87), would not be EF but some lesser value EJ; EJ/EF represents the proportion of dry steam in the mixture of water and steam; and JF/EF represents the proportion of moisture in the mixture. This proportion of dry steam in a water-and-steam mixture is called the *quality* of the steam and is denoted by x.

Steam that has not passed off the horizontal line, as EF, is called *saturated steam*. At point F it is *dry steam;* and, if it is at some point as J, it is called *wet-saturated steam* or simply *wet steam*. Saturated steam is usually in contact with some water at its own temperature. The temperature at which evaporation takes place for a given pressure is the *saturation temperature* for that pressure.

Steam anywhere along the line FH is "superheated" steam.

The line $ACEI$ is called the *saturated liquid line*, while the line $BDFI$, joining the ends of the evaporation lines, is called the *saturated vapor line*. These lines join at the critical point.

The area $ACEIFDB$ is the area of mixed liquid and vapor, while the area above and to the right of $IFDB$ is the superheated area. We may have water at any point, defined by its pressure, temperature, and enthalpy in either of these areas or on the liquid or saturation lines.

99. The Relation between Pressure and Temperature.—From Fig. 87 it is apparent that the absolute pressures do not rise at the same rate as do the temperatures but that pressures increase slowly between A and C and then very rapidly from C to I. Figure 88 is a curve plotted with absolute pressures against "saturation" temperatures.

100. The Relation between Pressure and Specific Volume.—As the pressure and temperature of saturated steam increase, the volume occupied by 1 lb of the vapor decreases according to the curve of Fig. 89.

101. Entropy of Steam.—In Art 72 the change in entropy of air as the temperature was changed at constant pressure was

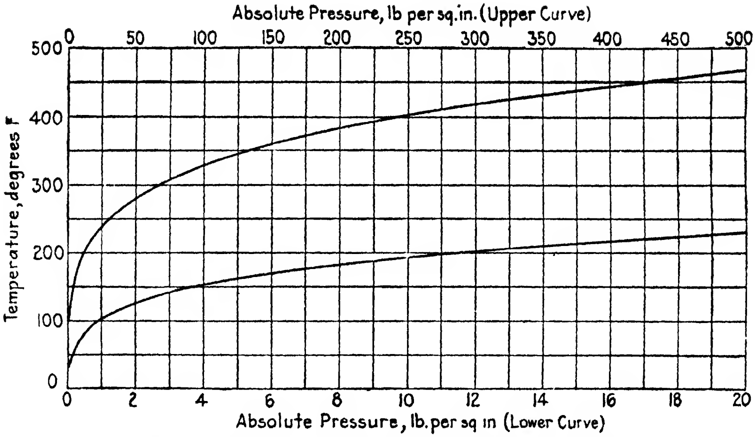


FIG. 88.—Relation between pressure and temperature of saturated steam.

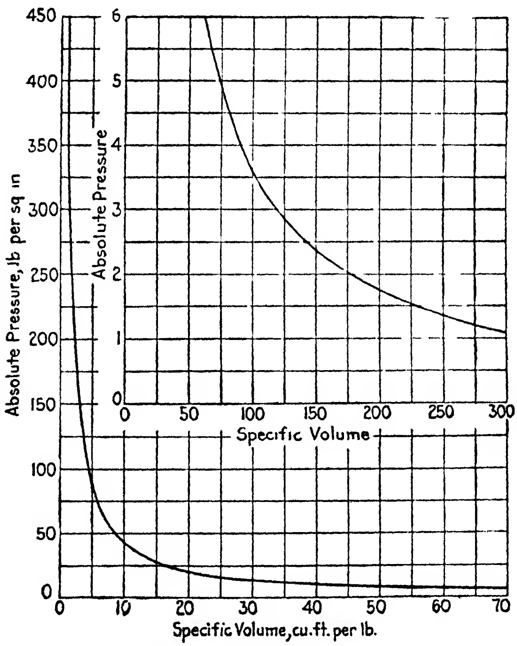


FIG. 89.—Relation between pressure and specific volume for saturated steam.

plotted on the T-S diagram (Fig. 68). We may plot a similar curve, Fig. 90, for the change in entropy of the pound of water of Figs. 86 and 87 as heat is added. We have as the general expression for entropy change from Art. 72:

$$s_2 - s_1 = \int_{T_1}^{T_2} \frac{dh}{T} = c_p \log_e \frac{T_2}{T_1} \tag{81}$$

1. *Entropy of the Liquid.*—If c_p is constant, we may compute the entropy change for any change of temperature. Since c_p

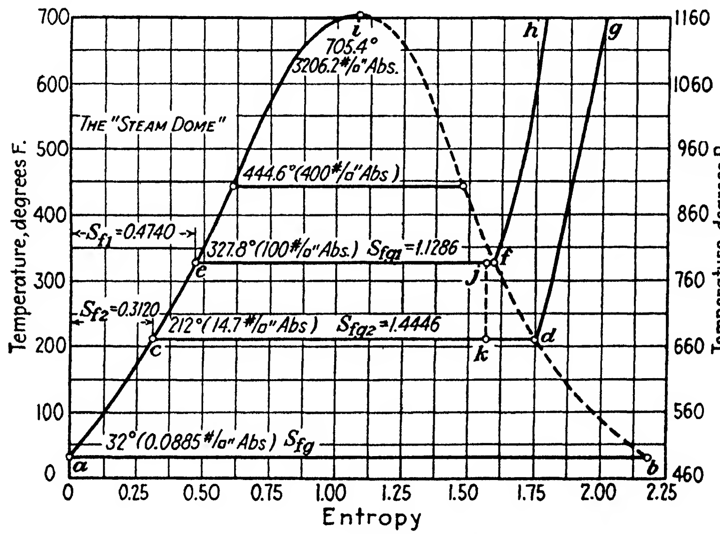


FIG. 90.—Temperature-entropy diagram for steam.

for water changes slightly, it will be necessary to calculate the change of entropy for small temperature changes over which c_p may be considered constant. Thus from 32°F., or 492°R., to 42°F., or 502°R., we have as the entropy change:

$$s_2 - s_1 = 1 \times \log_e {}^{502}\!/_{492} = 0.0202$$

In Fig. 90, beginning at a, 32°F., which temperature we shall consider our zero of entropy, we may proceed in steps to a temperature as high as we choose, and, plotting T against s, we obtain the curve *acei*. This curve is called the "saturated liquid" line on the T-S diagram.

2. *Entropy of Evaporation.*—If the pressure in the vessel of Fig. 86 is 14.7 lb abs, the heat added during evaporation, CD, is

970.1, and the temperature is constant at 212°F. (Fig. 87) or 672°R. Then, from Eq. (45*a*) and also by definition,

$$s_d - s_c = \frac{970.1}{672} = 1.445 \text{ units of entropy}$$

Plotting *cd* in Fig. 90 equal to 1.445, we have *cd* representing the entropy of evaporation at 212°F.

At a pressure of 100 lb per sq in. or 327.8°F. or 787.8°R., we have an entropy change from E to F (Fig. 87) equal to

$$\frac{888.8}{787.8} = 1.128 \text{ units of entropy}$$

Plotting *ef* in Fig. 90 equal to 1.128, we have *ef* representing the entropy of evaporation at 327.8°F.

By continuing to plot the entropy of evaporation to the right from the liquid line for several temperatures, we should have a locus *bdfi* of Fig. 90 which is called the "saturated vapor" line. The saturation and liquid lines join at the critical point and form what is called the "steam dome."

3. *Entropy of Superheat.*—In the same manner as for the liquid line, we may compute the entropy changes for the superheated state. The specific heat of superheated steam is much more variable than that of water. Then, for each particular saturation temperature (and pressure), we should have a particular superheat line on the T-S diagram, as *dg* for 14.7 lb and *fh* for 100 lb. We may let $s_{\text{sup.}}$ represent the entropy of superheat.

The total entropy (the abscissa) for any steam state is the sum of the entropies $s_f + s_{fg} + s_{\text{sup.}}$.

For steam that is not dry but has a quality x, as, for example, that represented by the state J, Fig. 87, the entropy of evaporation will be represented as xL/T, or *ej* of Fig. 90.

$$\frac{x(888.8)}{787.8} \text{ units of entropy}$$

It is to be remembered that in Fig. 87 temperature is plotted against enthalpy, while in Fig. 90 temperature is plotted against entropy above 32°. Also the area of Fig. 90 represents heat above 32°, since $TdS = dQ$, Art. 72.

102. Specific Heat of Superheated Steam.—As suggested in Art. 101, the specific heat at constant pressure, c_p, for superheated steam is variable. It varies with both pressure and the total temperature, as is shown by Table VI, which gives the mean specific heat from the saturation temperature to the temperature of the superheated steam. Table VI should be used for approximate calculations, only.

TABLE VI.—SPECIFIC HEAT OF SUPERHEATED STEAM[1]
Mean Values of c_p from Saturation Temperature to the Temperature of the Steam

Absolute pressure, lb. per sq. in.	Saturation temperature	Superheat, degrees Fahrenheit above saturation temperature						
		10	50	100	200	300	400	500
1	101.8	0.46	0.44					
14.7	212	0.50	0.48	0.48	0.47	0.47	0.47	
25	240.1	0.52	0.50	0.49	0.48	0.48	0.48	0.48
50	281.0	0.57	0.53	0.51	0.50	0.50	0.50	0.50
75	307.6	0.60	0.56	0.53	0.51	0.51	0.51	0.51
100	327.8	0.61	0.57	0.55	0.53	0.52	0.51	0.51
125	344.3	0.62	0.59	0.56	0.54	0.53	0.52	
150	358.4	0.64	0.61	0.58	0.55	0.54	0.53	
175	370.8	0.65	0.62	0.59	0.56	0.54	0.53	
200	381.8	0.66	0.63	0.60	0.57	0.55	0.54	
250	401.0	0.70	0.66	0.63	0.59	0.57	0.56	
300	417.3	0.75	0.69	0.65	0.60	0.58	0.57	
350	431.7	0.78	0.72	0.68	0.62	0.60	0.58	
400	444.6	0.85	0.75	0.70	0.64	0.61		
450	456.3	0.91	0.78	0.72	0.66	0.63		
500	467.0	0.95	0.80	0.74	0.68	0.64		
600	486.2	1.1	0.88	0.79	0.71	0.66		

[1] These values have been calculated from published results of the A.S.M.E. research on the properties of steam (*Mech. Eng.*, February, 1926).

103. The Steam Table.—Equations have been developed showing the interrelations of the various properties of steam discussed in the preceding paragraphs, but, since they are largely empirical, it is usually more satisfactory to have tables of the properties arranged for ready reference. These tables may be arranged with either temperatures in degrees Fahrenheit or absolute pressures in pounds per square inch as the independent variable. There are several tables which have been used in the past, among them being Peabody's, Goodenough's, Marks and Davis's, and Callender's, but, owing to the difficulty of the experimental work and to the fact that new and more nearly correct physical constants are continually being established, these tables do not exactly agree with each other.

It is now generally considered that the tables of "Thermodynamic Properties of Steam" by Keenan and Keyes[1] represent the best data available to American engineers. These tables are based upon a skeleton agreed upon by the International Steam Table Conferences, America's contribution resting upon researches carried on under the auspices of the American Society of Mechanical Engineers. This experimental work was divided into three major divisions: (1) the Joule-Thomson effect, (2) the pressure-volume relation, and (3) enthalpy. The three divisions of the work were carried on by Harvard University, the Massachusetts Institute of Technology, and the National Bureau of Standards, respectively. These researches excel all former work on the properties of steam both in elaborate provision to insure accuracy and in the ranges of pressures and temperatures explored.

Tables VII, VIII, and IX (see Appendix B), are abridged from the Keenan and Keyes tables.

104. The Enthalpy-entropy Diagram.—A very useful chart devised by Dr. Mollier, in 1904, is that shown in Appendix C in which the enthalpy of 1 lb steam is plotted against the entropy of the state for various conditions of quality and superheat at various pressures. Such a diagram, if drawn to a sufficiently large scale, is accurate enough for most engineering purposes and may be used instead of the steam table. Its use may be illustrated by the following examples:

Example A.—What is (*a*) the enthalpy and (*b*) the entropy of 1 lb steam at 100 lb per sq in. abs if (1) dry, (2) superheated 100°, or (3) its quality is 95 per cent?

SOLUTION.—1. On the diagram of Appendix C it will be noted that the line of *constant pressure* (running "northeast to southwest") marked 100, intersects the *saturation line* (running "northwest to southeast") at a point indicated by the vertical scale as slightly more than 1186 Btu and by the horizontal scale as slightly more than 1.6 units of entropy.

2. The 100-lb line intersects the 100° superheat line (roughly parallel to the saturation line) at a point indicated by the vertical scale as 1242 Btu and by the horizontal scale as 1.668 units of entropy.

3. Likewise the 100-lb line intersects the 5 per cent moisture line (also, roughly parallel to the saturation line) at 1142 Btu and 1.546 units of entropy.

[1] John Wiley & Sons, Inc., 1936.

Example B.—What would be the condition of 1 lb steam having enthalpy 1170 Btu at a pressure of (*a*) 200 lb per sq in. abs; (*b*) 12 lb per sq in. abs? (*c*) What is the temperature in each case?

SOLUTION.—(*a*) The 200-lb pressure line intersects the horizontal 1170-Btu line at a point 0.3 of the distance from the 3 to the 4 per cent moisture lines. Then the quality is 96.7 per cent.

(*b*) The 12-lb pressure line intersects the 1170-Btu line at a point half-way between the 40 and 60° superheat lines. The condition, therefore, is 50° superheat at 12 lb abs.

(*c*) Moist steam is at the saturation temperature corresponding to its pressure. Tables VI and VII indicate that the temperature of the 200-lb steam is 381.8°F. The saturation temperature of the 12-lb steam by Table VII is 202°, to which is to be added 50° superheat, making a total temperature of 252°.

105. Measurement of Steam Quality.—It was shown in Art. 98 that saturated steam may contain a small quantity of water in suspension in the form of tiny droplets or fog. The diagram of Fig. 87 suggests a simple method of measuring this quantity of moisture or its supplement, the quality x.

Suppose the point J represents the state point of steam in a steam pipe, that is, its pressure is 100 lb per sq in. abs, and its quality x_1 is EJ/EF. Then the enthalpy of each pound of the steam will be represented by the abscissa of the point J, and this heat quantity is

$$Q_1 = h_{f1} + x_1 h_{fg1} \tag{82}$$

Now, if the steam is caused to pass through a small orifice into a chamber in which the pressure is about atmospheric, the temperatures will drop as well as the pressure. If the chamber is insulated, so that no heat will be either received from an outside source or lost from the steam by radiation and if the velocities of the steam entering and leaving are so small that the kinetic energy is negligible, it may be assumed that this pressure and temperature drop will be at constant enthalpy and will be represented on the chart of Fig. 87 by the line JKM. The steam has now an abscissa equal to that of J or M, but its pressure is only 14.7 lb. The steam, therefore, has now an enthalpy of superheat represented by the horizontal distance DM, which is equivalent to state point K on the superheat curve DKG for steam at 14.7 lb pressure.

The enthalpy of the lower-pressure steam is now accounted for by the abscissa of the state point K, which is

$$Q_2 = h_{f2} + h_{fg2} + c_p(t_{sup.} - t_2) \tag{83}$$

in which $t_{sup.}$ is the actual temperature to which the steam falls inside the insulated chamber and t_2 is the saturation temperature corresponding to the pressure inside the chamber. c_p is the mean specific heat of the superheated steam from saturation to the temperature and pressure in the chamber.

Since JK is a constant-enthalpy line, Q_1 of Eq. (82) must equal Q_2 of Eq. (83), and we have

$$h_{f1} + x_1h_{fg1} = h_{f2} + h_{fg2} + c_p(t_{sup.} - t_2) \tag{84}$$

Then, to make this scheme usable, we need only to know the actual temperature $t_{sup.}$ in the chamber and the pressure p_2 in the chamber. By the aid of the steam table we may find $h_{f2} + h_{fg2}$ (which equals h_{g2}) and t_2. The quantities h_{f1} and h_{fg1} are, also, found from the steam table, since the initial pressure p_1, or temperature t_1, is known. We may now solve Eq. (84) for x_1, which is the only unknown in the equation. c_p at atmospheric pressure may be taken as 0.48 for the probable range of superheat (see Table VI).

It is apparent from the diagram that the maximum quantity of moisture that can be determined by this means is that for which the point J would fall directly above point D, that is, for which the enthalpy is 1150.4 Btu for $p_2 = 14.7$ lb, because at D there is no superheat and the equation is indeterminate. For the example shown in the diagram for $p_1 = 100$ lb per sq in. abs, steam at this pressure would have 298.4 Btu as enthalpy of the liquid, leaving only 852.0 Btu for x_1h_{fg1}. But h_{fg1} at this pressure is 888.8. Hence $x_1 = 852.1 \div 888.8 = 0.96$, or 96 per cent. The limiting quantity of moisture that could be measured at 100 lb abs pressure is, therefore, 4 per cent.

106. The Superheating Calorimeter.—The instrument used in this method for steam quality is called a *throttling* or *superheating calorimeter*, one form of which is shown in Fig. 91. Steam from the main is collected through a sampling nozzle which should be made in accordance with the A.S.M.E. Test Code[1] (see

[1] Power Test Codes, Series 1929, Instruments and Apparatus.

sampling tube at A in Fig. 92). The temperature of the steam from the main is measured by a thermometer in the well C. A $\frac{3}{32}$-in. orifice at D allows the steam to flow into the chamber E, where the pressure is that of the atmosphere and where the temperature is measured by the thermometer in the well F.

The entire instrument is carefully insulated to reduce radiation, the effect of which would be to lower the temperature t_{sup}

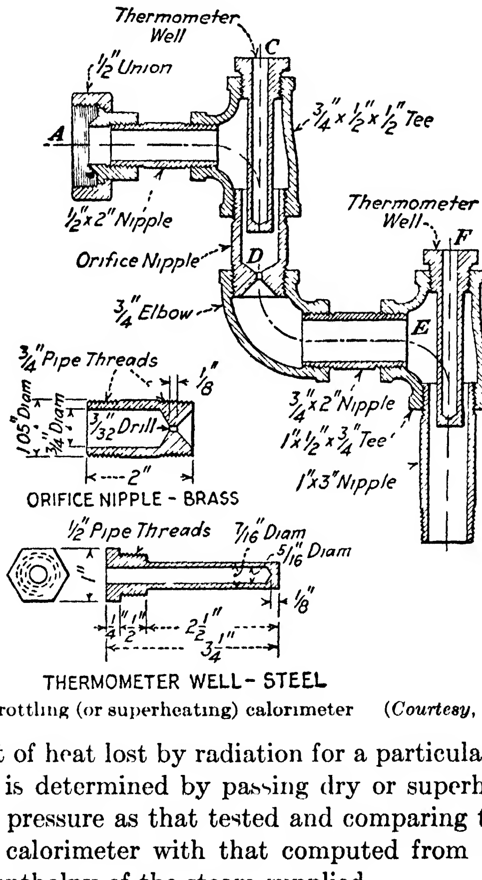


FIG. 91 —A throttling (or superheating) calorimeter (*Courtesy, Blaw-Knox Co.*)

The amount of heat lost by radiation for a particular instrument as installed is determined by passing dry or superheated steam of the same pressure as that tested and comparing the temperature in the calorimeter with that computed from Eq. (84) for the known enthalpy of the steam supplied.

In using the throttling calorimeter, the pressure or temperature of the steam in the main is read, and also temperature and absolute pressure inside the superheat chamber E are read. The chamber pressure is that indicated by the barometer at the time of making the test, except in those instruments that have a constricted outlet which builds the pressure up to slightly above atmospheric. In this case a mercury manometer is used, and its indication added to the barometer.

Then, with p_1 (or t_1), p_2, and $t_{sup.}$, we may select from the steam table h_{f1} and h_{fg1}, corresponding to p_1, and h_{g2} and t_2,

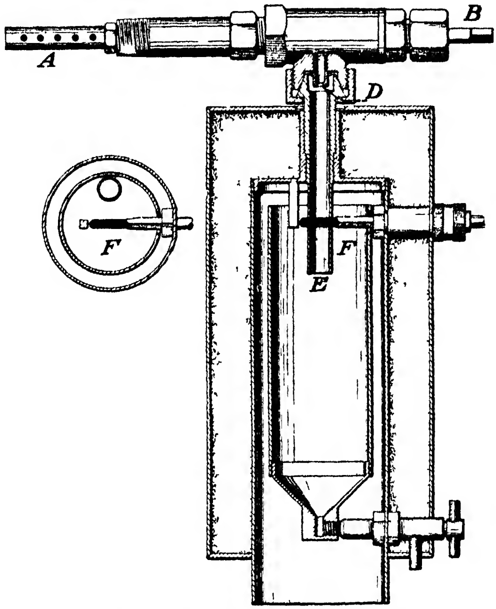


FIG. 92. Ellison U-path calorimeter

corresponding to p_2. With these values substituted in Eq. (84), we may readily solve for x_1.

Example A.—A throttling calorimeter test gave the following data: p_1 = 150 lb per sq in. abs; $t_{sup.}$ = 267°F., and p_2 = 14 lb per sq in. abs. What was the quality of the steam in the main?

SOLUTION 1.—From the steam table h_f = 330.5, h_{fg1} = 863.6, t_2 = 209.6°, and h_{g2} = 1,149.3 Btu. These values substituted in Eq. (84) give

$$330.5 + x_1(863.6) = 1{,}149.5 + 0.48(267 - 209.6)$$

and solving for x_1, we have

$$x_1 = 98 \text{ per cent}$$

and the moisture = 100 − 98 = 2 per cent.

SOLUTION 2.—The Mollier chart, Appendix C, offers a very convenient solution for the throttling calorimeter, as follows:

The final state point in the calorimeter is indicated by superheated steam at 14 lb abs. and 267 − 210 = 57° superheat.

The intersection of 57° superheat with the 14-lb line is at 1175 Btu. Now, since the pressure drop in the calorimeter was one at constant enthalpy, if we follow the 1175 Btu line to the left to the 150-lb line, we find the intersection to be at 2.1 per cent moisture, a result agreeing reasonably well with the 2 per cent of solution 1

A recent improved superheating calorimeter is shown in Fig. 92. Steam enters the instrument through the perforated sampling nipple A at the top and, after passing the angle valve B, expands into the inner chamber E through a small orifice D in the union connection. The flow of the steam through the calorimeter forms a U path, going down into the steam chamber E and then up and back down on the outside of the chamber. Any moisture still held by the steam as it enters the chamber is thrown out by the sudden change in direction of the steam flow and then evaporated. The thermometer is placed near the top of the chamber in a horizontal position.

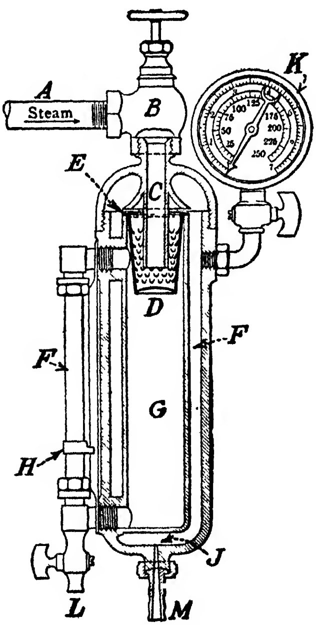


Fig. 93.—Separating calorimeter.

The outer passage around the chamber forms a steam jacket to prevent radiation, and the jacket steam is, in turn, protected against chilling by the insulating jacket on the sides and top, which is filled with lampblack and encased in a bright nickeled casing.

107. The Separating Calorimeter. A second type of calorimeter is the separating calorimeter shown in Fig. 93. The instrument is connected by the pipe A to the sampling nipple in the steam line and is controlled by the valve B. Steam enters through the passage C and on striking the cup D is deflected sharply upward and then outward through the narrow annular slit E into the outer jacket space F. Now the entrained moisture, being so much heavier than the steam, is thrown outward to the sides of the cup D and out of the slits in the side of the cup D into the chamber G, where it collects. The steam which has been freed of its moisture passes out of the nozzle J.

The quantity of moisture that has been separated by the instrument is determined by reading the scale beside the gage glass which is graduated to hundredths of pounds. The quantity of dry steam flowing through the orifice J is indicated by the pressure gage K which has a calibrated scale near the outer edge reading in pounds per 10 min.

Then the quality of the steam is determined by the following formula:

$$x = \frac{W}{W + w} \tag{85}$$

in which W = pounds of dry steam passing J per 10 min.
w = pounds of moisture collected in G per 10 min.

The calibration on the gage was made experimentally but may be checked approximately by the following formula, which is called "Napier's rule:"

$$W_s = \frac{PA}{70} \tag{86}$$

in which W_s = pounds of dry steam flowing through an orifice per second.
P = absolute pressure of the steam in pounds per square inch.
A = the area of the orifice in square inches.

This formula is independent of the pressure on the low-pressure side of the orifice if the low pressure is not more than 0.57 of the higher pressure. It holds only for dry, saturated steam over a limited range.

In practice, a more accurate method of determining the quality of steam with this instrument is as follows:

A hose is attached to the cock L and another to the steam outlet M, and each hose is led to a vessel of cold water standing on scales. The scale for weighing the collected moisture should read to hundredths of pounds; at a given signal, both scales are read simultaneously and, again, 10 min later. For check readings, the scales should be read at several 10-min intervals. Then the scale readings will give w and W as the weights of water and steam, respectively, which may be substituted in Eq. (85), as before. It may be necessary to place ice in the water in the vessel collecting the steam discharged from M to avoid its heating enough to lose weight by evaporation.

The separating calorimeter, like the superheating instrument, should be carefully insulated against loss of heat by radiation.

The separating calorimeter is, theoretically, not limited in the percentage of moisture it will handle, but, when the quality of the steam measured is low, it is sometimes well to attach a throttling calorimeter to the discharge M of the separating instrument in order to be sure that the separation is complete. In this case the quality of the steam measured is

$$x = \frac{x_T W_1}{W_1 + w} \tag{87}$$

in which W_1 = the weight of *apparently* dry steam passing the orifice J per 10 min.

x_T = the quality of these W_1 lb of apparently dry steam as measured by the throttling calorimeter.

Problems

1. Assuming the latent heat of fusion of ice to be 144 Btu and its specific heat to be 0.5, (*a*) extend the temperature-heat content chart (Fig. 87) to a point 25° below zero. (*b*) What is the state of water in the region below and to the right of the lines drawn in (*a*)?

2. What is the mean specific heat of water from 32° to the "critical" point? What is the specific heat *at* the critical point?

3. Using the chart of Fig. 87, what is the state of 1 lb steam (*a*) if its temperature is 444.6°F. and its enthalpy is 1100 Btu; (*b*) if its enthalpy is 1250 Btu?

4. Calculate the increase in entropy when 1 lb water is raised from 32°F. and converted into dry steam at 400 lb per sq in. abs.

5. (*a*) What is the normal boiling point at an altitude at which the normal barometer is 25 in. Hg? (*b*) The enthalpy of a pound of dry steam at this pressure? (*c*) The entropy?

6. Rankine gave $pv^{1\frac{7}{16}} = 475$ as the equation of the relation between the pressure and volume of 1 lb dry saturated steam. Test this equation with the steam table at 200, 20, and 5 lb.

7. A pound of "steam" at 150 lb per sq in. abs occupies a volume of 2.94 cu ft. (*a*) What is the state of the steam? (*b*) What is the state of the steam if it has a volume of 3.5 cu ft per lb. at this pressure?

8. A cylinder contains 2 cu ft steam at a pressure of 175 lb per sq in. abs and a temperature of 475°F. What is (*a*) the weight of the steam; (*b*) its enthalpy?

9. What would be the result of mixing 10 lb of superheated steam at 100 lb per sq in. and 428°F. with 10 lb water at 328°F. and the same pressure?

10. How many pounds of dry steam at 75 lb gage at sea level will be required just to melt 100 lb ice? Latent heat of fusion of ice is 144 Btu.

11. What will be the resulting condition of the water involved in adding 1 lb of dry steam at 125 lb abs to a mixture of 10 lb water and 10 lb ice?

12. Find the resulting condition of the water involved when 3 lb steam at 200 lb per sq in. and 200° superheat are injected into a vessel containing 12 lb water and 2 lb ice, the pressure in the vessel remaining at 14 lb per sq in. abs during the process.

13. One pound of dry steam at 100 lb per sq in. gage expands through an orifice into a vessel at atmospheric pressure of 14 lb. What is the temperature of the steam in the vessel?

14. Steam at 80 lb per sq in. gage and 98 per cent quality passed through a throttling calorimeter, the pressure in the calorimeter being atmospheric at 14.5 lb per sq in. What was the temperature in the calorimeter?

15. A calorimeter test gave the following data: pressure in the main, 60 lb per sq in. gage; barometer, 28.4 in.; pressure in calorimeter, 1.2 in. Hg above atmosphere; temperature in calorimeter, 264°F. What was the quality of the steam in the main?

16. Perfectly dry steam at 145 lb per sq in. abs is passed through a throttling calorimeter. The barometer is 27 in., and the pressure in the calorimeter is 0.4 in. Hg above atmosphere. (*a*) What should have been the temperature in the calorimeter? (*b*) If the temperature was actually 294°F., what was the percentage of moisture erroneously indicated by the instrument due to radiation?

17. A separating calorimeter test gave the following data during a 30-min run: water collected in the chamber, 0.27 lb.; rate of flow of dry steam leaving the orifice as indicated by the gage was 3.1 lb per 10 min. What was the quality of the steam tested?

18. A separating calorimeter and a throttling calorimeter are connected to the same steam pipe in which the pressure is 150 lb gage. The barometer is 30 in. Size of dry steam orifice of separating calorimeter is $\frac{5}{64}$ in. and water collects in the chamber at the rate of 0.45 lb per 10 min.

The temperature in the throttling calorimeter was 213°F., and the pressure was 15 lb per sq in. abs. (*a*) What was the quality of the steam in the pipe? (*b*) Explain what happened in the throttling calorimeter.

19. Steam at 100 lb. per sq in. abs passes through a separating calorimeter. During a 40-min run, 17.1 lb dry steam leave the instrument and 1.5 lb moisture are collected. What was (*a*) the quality of the steam; (*b*) the approximate size of the orifice, using Napier's rule?

20. What is the maximum percentage of moisture that can be measured by a throttling calorimeter at sea level in steam (*a*) at 275 lb per sq in. gage; (*b*) at 40 lb per sq in. gage?

21. Steam at 125 lb per sq in. abs. is passed through a throttling calorimeter, the calorimeter temperature being 334°F. and the pressure being 15 lb per sq in. abs. What was the condition of the steam tested?

22. A separating calorimeter is connected to a steam main in which the pressure is 145 lb per sq in. abs. During a 30-min run, 0.9 lb moisture is collected. The diameter of the orifice is 0.08 in.

A throttling calorimeter is connected to the discharge of the separating calorimeter, the pressure in the pipe between the two instruments being 60 lb per sq in. abs. The throttling calorimeter temperature and pressure are 261°F. and 16 lb per sq in. abs. Find the moisture indicated by (*a*) the throttling calorimeter; (*b*) the separating calorimeter; (*c*) find the initial moisture.

23. Calculate the approximate steam flow in pounds per hour through a 1-in. diameter orifice from 160 lb per sq in. gage to the atmosphere the pressure of which is 14 lb per sq in., using Napier's rule.

24. A flat valve disk 3 in. in diameter is lifted $\frac{1}{8}$ in. and allows 98 per cent quality steam at 155 lb per sq in. abs to flow into a chamber in which the pressure is 80 lb per sq in. abs. Estimate the weight of wet steam flowing into the chamber per hour, using Napier's rule.

CHAPTER XII

STEAM-ENGINE PERFORMANCE

108. The Action of Steam in the Engine Cylinder.—Figure 94 shows a slide-valve engine cylinder with a typical indicator card. Steam at throttle-valve pressure is admitted to the

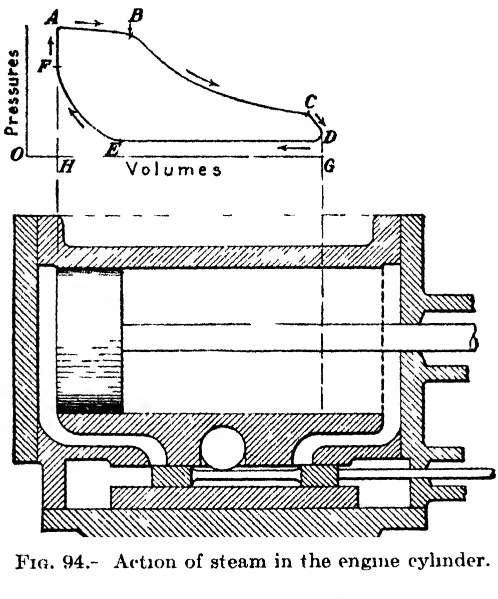


FIG. 94.- Action of steam in the engine cylinder.

cylinder at the point A, and we have full steam pressure acting upon the piston until the point B is reached, when the valve *cuts off* the steam supply. The steam now in the cylinder acts expansively upon the piston during the forward stroke until the point C is reached, when the exhaust port is opened and *release* occurs, the pressure dropping as the volume increases. From C to D the pressure drops quickly to nearly that in the exhaust pipe, and then, as the piston returns toward the left end of the cylinder, the spent steam is swept out of the cylinder along the line DE at exhaust pressure.

At E the exhaust is closed and the steam that is trapped in the left end of the cylinder is compressed along the line EF into the clearance space OH. This compression acts as a cushion and relieves the engine mechanism from a large part of the inertia forces acting upon the piston. By compressing some of the exhaust steam into the clearance volume prior to admitting fresh live steam, the quantity of steam used per stroke is materially reduced. Otherwise, live steam would be required to fill the clearance space, with little effect in producing work. This saving is much greater than the energy required for compression and so represents a real economy. It should be noted that the clearance space includes not only the space directly at the end of the cylinder but the steam port up to the valve as well.

The events of the diagram are named as follows:

F, point of admission.
F to B, admission
B, point of cut-off.
B to C, expansion.
C, point of release.
C to E, exhaust.
E, point of compression.
E to F, compression.

The ordinates of the upper part of Fig. 94 ($FABCD$) can be taken to represent steam pressures acting during the forward stroke of the piston. The ordinates of the lower part (DEF) represent the back pressure against the return stroke of the piston. The mean height between $FABCD$ and DEF will indicate the mean effective pressure which may be considered as acting upon this side of the piston once per revolution, for an effective working distance of one stroke. Since this is a double-acting engine, a similar action will occur in the opposite end of the cylinder, steam being admitted and expanded at the crank end while exhaust and compression are occurring in the head end.

Since the valve does not open and close instantly, the steam is cut off and released somewhat gradually, and this produces a rounded corner at B, C, D, and E due to friction, or "wire-drawing," as it is called Another result of wiredrawing is the slightly sloping admission AB.

109. The Conventional Steam-engine Indicator Diagram.—As was done in the cases of the internal-combustion engine cycles (Arts. 74 and 80), we may conventionalize the steam-engine indicator diagram of Fig. 94 by making the following assumptions:

1. The stroke of the conventional card is the same as for the real card.
2. There is no clearance. Really, there should be none, since the clearance steam represents a waste, but in practice it is impossible to build a steam engine without it.
3. Since there is no clearance, there can be no compression.
4. The steam admission and exhaust pressures are the same as for the real engine, and these lines are horizontal.
5. The valve acts instantly; and there are no rounded corners.
6. The expansion line is a rectangular hyperbola, the equation of which is $pv = c$. This is the curve most nearly like the one found on actual indicator diagrams and has the advantage of being easily dealt with mathematically.

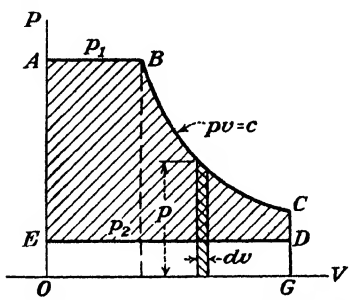


FIG. 95.—Conventional steam-engine indicator diagram.

Figure 95 shows such a conventional card. The shaded area, which represents the net work during the cycle, is represented by W_i = work under AB + work under BC − work under DE

$$= p_1v_b + \int_{v_b}^{v_c} p\,dv - p_2v_d \tag{88}$$

Now, since $pv = c$, $p = c/v$; also $v_c = v_d$; therefore

$$W_i = p_1v_b + c\int_{v_b}^{v_c} \frac{dv}{v} - p_2v_c$$

$$= p_1v_b + p_1v_b \log_e \frac{v_c}{v_b} - p_2v_c$$

$$= p_1v_b\left(1 + \log_e \frac{v_c}{v_b}\right) - p_2v_c \tag{89}$$

If we let $r = v_c/v_b$, the ratio of expansion, we have

$$W_i = p_1v_b(1 + \log_e r) - p_2v_c \tag{90}$$

The mean effective pressure acting on the piston during the cycle is found by dividing the work by the volume swept through. That is, for this diagram,

$$(\text{Mep})_i = \frac{W_i}{v_c} = \frac{p_1 v_b}{v_b}(1 + \log_e r) - p_2$$

$$= \frac{p_1}{r}(1 + \log_e r) - p_2 \tag{91}$$

110. Rated Indicated Horsepower of a Steam Engine.—If the theoretical mean effective pressure calculated from the conventional indicator card is expressed by Eq. (91), the horsepower would be, for a single-acting engine:

$$(\text{Ihp})_i = \frac{p_m LAN}{33{,}000} \tag{92}$$

in which p_m = the mean effective pressure of the conventional card.

N = rpm.

In practice, the actual horsepower that may be expected from an engine is less than that calculated from the conventional card by 10 to 30 per cent, owing principally to the fact that the actual engine has compression and has friction in the steam passages which rounds off the corners of the cards. The probable rated horsepower of an engine of given dimensions and rated speed may be found by applying a *card factor e* to the equation, which, when completed, becomes for a single-acting engine

$$(\text{Ihp})_R = \frac{e\left\{\frac{p_1}{r}(1 + \log_e r) - p_2\right\}LAN}{33{,}000} \tag{93}$$

in which $(\text{Ihp})_R$ = rated indicated horsepower estimated from the cylinder dimensions L and A and steam pressure p_1 and exhaust pressure p_2, as given.

N = rpm.

e = card factor for the particular kind of engine in question. This varies from 0.70 to 0.90.

For a double-acting engine, an additional equation similar to Eq. (93) would be used for the crank end, in which the area A would be corrected for the area of the piston rod.

The cut-off is usually given as occurring at a certain percentage or fraction of the stroke. In this case the ratio of expansion of the conventional engine is the reciprocal of the fraction of the stroke completed at cut-off. Thus, for a conventional engine cutting off at one-third stroke, the ratio of expansion would be 3.

111. Effect of Clearance in the Actual Engine.—The clearance volume has the effect of decreasing the apparent ratio of expansion, since the clearance and the piston displacement together constitute the space occupied by the steam. Clearance is expressed as a percentage of the piston displacement and varies from 1 to over 15 per cent. The real ratio of expansion is, in Fig. 94, $(OH + HG):(OH + AB)$ and, in terms of percentage clearance,

$$\frac{100 + C}{C.O + C}$$

in which C = percentage of clearance in terms of displacement
$C.O.$ = per cent of stroke or displacement completed at cut-off.

112. The Rankine Cycle.—The indicator card of the engine discussed in Art. 109 accounts for the work which would be done on the piston under the conditions prescribed. As far as

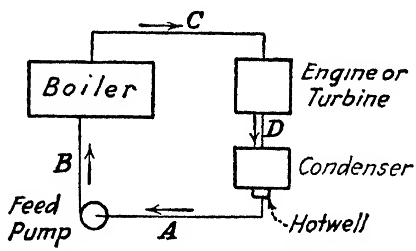


FIG. 96.—Essentials of a steam power plant Rankine cycle.

these conditions are concerned, the working substance might have been compressed air or any other gas; that is, the conventional indicator card deals only with the development of work in the engine cylinder and not with the complete transformation of heat energy into mechanical energy. To make an analysis of the steam engine comparable to that made for the internal-combustion engines in Chap. VIII, it is necessary to consider the other pieces of equipment that go to make up the entire external-combustion engine plant.

Figure 96 represents, diagrammatically, the essential pieces of the apparatus illustrated in Fig. 85 of the complete steam-power plant. The working substance, water, makes the continuous cycle from boiler to engine (or turbine) to condenser to feed pump and back to the boiler, as shown by the arrows. The pressure-volume relations of the water during the cycle are shown by the diagram of Fig. 97, in which, as usual, the ordinates represent pressure and abscissas represent volumes.

The boiler feed pump forces the condensate into the boiler along the line *ab*, the pressure being raised from that in the condenser hot well to that in the boiler. Now the boiler adds heat to the water and evaporates it at constant pressure from volume *fb* to volume *fc*. From *c* to *d* the steam is expanded adiabatically in the prime mover; *da* represents the decrease in volume which the steam undergoes in passing through the condenser and being condensed.

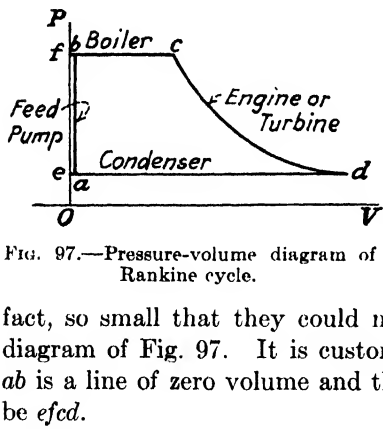


FIG. 97.—Pressure-volume diagram of Rankine cycle.

It is assumed in this idealized circuit that no heat is lost in any apparatus except that rejected to the condenser and that no heat is added except that acquired while the water is in the boiler. In practice, the water volumes *ea* and *fb* are so small as to be negligible, in fact, so small that they could not be seen on the scale of the diagram of Fig. 97. It is customary, therefore, to assume that *ab* is a line of zero volume and that the figure thus made would be *efcd*.

The series of events, *efcd*, represents the Rankine cycle and is a conventional measuring stick for steam-power plants. The Rankine cycle is not necessarily the most efficient steam cycle, as will be shown in Chap. XIII, but it does offer a conventional standard of reference. The fact that this is not a perfect standard should cause no more confusion than the fact that neither the Otto nor the Diesel cycle is the most efficient ideal internal-combustion cycle.

113. Efficiency of the Rankine Cycle.—The efficiency of the Rankine cycle is found in the same manner as that of the Otto cycle found in Arts. 73 to 77, that is,

$$e_R = \frac{Q_1 - Q_2}{Q_1} = \frac{\text{heat added} - \text{heat rejected}}{\text{heat added}}$$

A convenient basis for this calculation is 1 lb of the working substance, water. It is assumed that the amount of work done upon the water by the feed pump is negligible and that heat is neither added nor lost as the water passes through the pump. Then Q_f must equal Q_e.

Heat is added in the boiler, and the quantity thus added is

$$\begin{aligned} Q_1 &= \text{enthalpy at } c - \text{enthalpy at } e \\ &= h_c - h_e \end{aligned} \tag{94}$$

Heat is rejected to the condenser, and the quantity thus rejected is

$$\begin{aligned} Q_2 &= \text{enthalpy at } d - \text{enthalpy at } e \\ &= h_d - h_e \end{aligned} \tag{95}$$

Then

$$e_R = \frac{Q_1 - Q_2}{Q_1} = \frac{(h_c - h_e) - (h_d - h_e)}{h_c - h_e} \tag{96}$$

$$e_R = \frac{h_c - h_d}{h_c - h_e} \tag{97}$$

In using this expression for the efficiency of the Rankine cycle, it should be noted that

h_c = the enthalpy of the steam as it leaves the boiler. For saturated steam this is $h_{fc} + x_c h_{fgc}$, and for superheated steam, h_{gc} + superheat.

h_d = the enthalpy of the steam after an adiabatic expansion through the prime mover. Usually, the steam is moist at d and $Q_d = h_{fd} + x_d h_{fgd}$. The manner of calculating x_d is shown in the following article.

h_e = the enthalpy of the liquid of the condensate as it leaves the condenser and is equal to h_{fe} corresponding to the pressure in the condenser. This represents the heat salvaged from the exhaust from the engine or turbine.

The efficiency equation might have been calculated from the expression $e_R = W/Q_1$ if we know the equation for the steam adiabatic on the pressure-volume diagram. Unfortunately we do not know this with certainty for all conditions of steam at the point c, and so it is more convenient to use Eq. (97). We do, however, know that $W = h_c - h_d$.

114. Condition of Steam after an Adiabatic Expansion.—It was pointed out in Art. 72 that an adiabatic expansion is a constant-entropy expansion. Since this applies to a steam adiabatic as well as to a gas adiabatic, if we know the state of the steam at c, we may calculate the state at the point d at the lower pressure, which is known, by remembering that the entropy at d is the same as that at c. We have, then, by hypothesis,

$$s_c = s_d \tag{98}$$

in which s_c and s_d are the entropies at c and d, respectively. For moist steam, s becomes

$$s = s_f + xs_{fg}$$

in which s = entropy of the steam at quality x.

The calculation for the quality of the steam after adiabatic expansion is illustrated by the following example:

Example A.—One pound of dry steam at 240 lb per sq in. abs expands adiabatically to 10 lb per sq in. abs. What is the state of the steam after expansion?

Solution 1.—We may write, as an equality of the entropies before and after passing through the adiabatic drop,

$$s_{gc} = s_{fd} + x_d s_{fgd}$$

in which s_{gc} = entropy of the dry steam at pressure c.

s_{fd} = entropy of the liquid at pressure d.

s_{fgd} = entropy at evaporation at pressure d.

x_d = quality at d, the unknown quantity in this equation.

Finding these entropies for the conditions of the example, from the steam table, and substituting in the above equation, we have,

$$1.5298 = 0.2835 + x_d(1.5041)$$

and, solving for x_d,

$$x_d = \frac{1.5298 - 0.2835}{1.5041} = 0.828 \text{ or } 82.8 \text{ per cent}$$

and the moisture is $100 - 82.8 = 17.2$ per cent.

Solution 2.—The Mollier chart offers a simpler solution. Referring to the diagram of Appendix C, we find the crossing of the 240-lb pressure line with the saturation line to fall on the 1.53 entropy line. This crossing, also falls on the 1200 Btu enthalpy line and makes $h_c = 1200$ Btu.

Now, following the 1.53 entropy line down to its intersection with the 10-lb pressure line, we find this intersection to be on the 17 per cent moisture line and the enthalpy h_d to be 977 Btu. $h_f = (193 - 32) = 161$ Btu from the temperature corresponding to 10 lb pressure in Table VIII.

If it is desired now to calculate the efficiency of the cycle, we may substitute these values of h_c, h_d, and h_e in Eq. (97). Thus

$$e_R = \frac{1{,}200 - 977}{1{,}200 - 161} = \frac{223}{1{,}039} = 0.215 \text{ or } 21.5 \text{ per cent}$$

It should be noted that the Rankine-cycle efficiency does not include the efficiency of the boiler in absorbing heat from the fuel. This is treated in Chap. XIX.

115. Economy of Steam Prime Movers.—In Art. 86, the economy of an internal-combustion engine was stated in terms of the *fuel* consumption. For steam prime movers it is more convenient to state economy in terms of *steam* consumption per unit of output. This is called the *steam rate* of the machine. For reciprocating engines this becomes pounds of steam per indicated horsepower-hour or per brake horsepower-hour. For a steam turbine, economy is stated in terms of pounds of steam per shaft horsepower-hour, or per kilowatt-hour if the turbine drives a generator.

This statement of economy in terms of steam rate is not entirely satisfactory because of the difference in enthalpy of the steam used by different machines. A more definite statement is in terms of *Btu supplied per horsepower-hour* or per kilowatt-hour, which is called the *heat economy*. The heat economy is found by multiplying the steam rate by the heat chargeable to the prime mover (including the condenser), which equals $(Q_1 - h_{f2})$. Q_1 is the heat per pound of steam supplied, measured at the engine throttle, and h_{f2} is the enthalpy of saturated liquid of the exhaust or, in other words, the heat salvaged by the condenser, that is, the heat left in the condensate which is to be returned to the boiler.

116. Thermal Efficiency.—Since the heat economy of an engine represents the input for an output of 1 hp-hr or 1 kw-hr, the efficiency of the engine as a heat-power transforming machine is

$$e_a = \frac{2{,}543}{\text{heat economy, Btu/hp-hr}} = \frac{2{,}543}{s_{\text{hp hr}}\,(Q_1 - h_{f2})} \tag{99}$$

or

$$e_a = \frac{3{,}413}{\text{heat economy, Btu/hp-hr}} = \frac{3{,}413}{s_{\text{kw hr}}\,(Q_1 - h_{f2})}$$

in each e_a = actual heat efficiency or thermal efficiency.

v = steam rate per horsepower-hour or kilowatt-hour.

It is to be expected that e_a would fall somewhat short of e_R (the theoretical efficiency of the Rankine cycle), owing to imperfections in the actual plant; and the ratio e_a/e_R is called the *Rankine-cycle efficiency ratio* or, simply the *efficiency ratio* or *engine efficiency*. This ratio is the measure, in terms of the Rankine cycle, of the perfection of the heat-power machine. In most plants, this ratio is less than unity, but in some of the modern ones, the efficiency of the theoretical Rankine cycle is actually exceeded. It should be said, however, that the cycles on which these plants operate are much more elaborate than the Rankine cycle and that the prime movers are turbines instead of reciprocating engines.

117. Steam Rates of Typical Engines.—The steam consumption of reciprocating engines varies greatly and depends upon a number of factors that are treated more at length in the following

TABLE X.—HISTORICAL DEVELOPMENT OF THE STEAM ENGINE
Marks' "Handbook"

Engine	Ratio of expansion	Boiler pressure, lb. per sq in.	Type	Lb. steam per i.hp.-hr
Newcomen, 1700	No expansion	atmospheric	Vacuum	
Watt, 1763	4	7 to 25	Simple condensing	
Perkins, 1825	....	700 to 800	Simple	1.5 lb. coal
Cornish pumps, 1840	1½: 3½	45	Simple condensing	16½ to 24
Marine, 1850–1890	6	60 to 80	Simple or compound, condensing	19½
Marine, 1850–1890	14	150	Triple condensing	15
Marine, 1850–1890	15	210	Quadruple condensing	13½
Leavitt at Lawrence, 1872	16	120	Compound (Woolf) condensing, 12 r.p.m	16½*
Corliss at Pawtucket, 1878	16	120	Compound condensing	13¾*
Various power-plant engines	....	....	Triple condensing	12½
Rockwood and Greene	26	150	Compound condensing, 7: 1 cylinder ratio	12½*
Rice and Sargent	33	150	Compound condensing	12⁷⁄₁₀*
Van den Kerchove	32	130	Compound condensing	12*
Westinghouse	29	185	Compound condensing	12
Leavitt, Snow, Allis pumps	25: 33	175	Triple condensing	11 05*
Allis pump	....	85	Triple condensing	10.33
Stumpf, uniflow	....	140	Simple condensing	13.64
Van den Kerchove	....	130	Compound condensing	8.99(*A*)
Stumpf, uniflow	....	190	Simple condensing	9.06(*B*)
Locomobile	....	220	Compound condensing	8.25(*C*)
Schmidt	....	794	Condensing	5.12(*D*)

Mostly quoted from Denton, *Stevens Institute Indicator*, January, 1905.

* Engines jacketed. All engines used saturated steam except last four, which had total temperatures as follows: (*A*), 663°F.; (*B*), 667°F.; (*C*), 660°F.; (*D*), 815°F.

chapter. In general, however, the larger the engine, the higher the supply pressure, the lower the back pressure, and the larger the number of expansion stages the lower will be the steam rate. Table X gives the results of tests on various types of engines arranged so as to show the development of the steam engine historically.

118. Variable-load Economy.—It was pointed out, in Chap. IX, that the economy of an internal-combustion engine varies with the load. The same holds true for the steam engine.

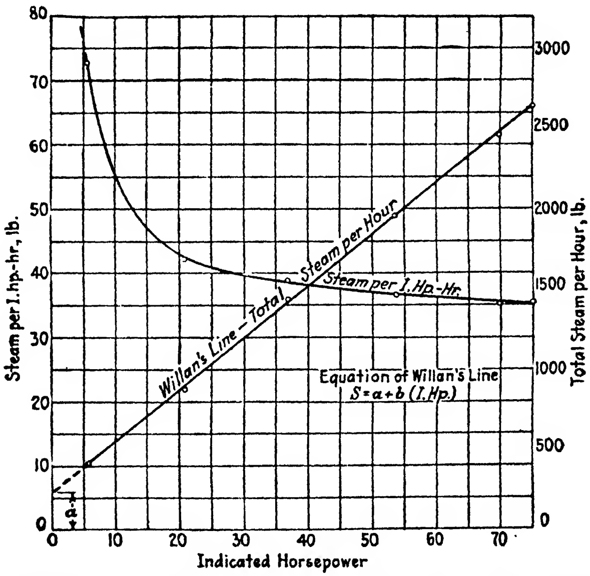


FIG. 98.—Variation of steam consumption with load (throttling-governed engines).

Figure 98 shows the variation in steam consumption of an engine having a *throttling governor* (see Art. 142). The straight line illustrates what is commonly called *Willan's law*, namely, that at a fixed cut-off and a variable initial pressure, the total steam consumption varies directly as the load, or, mathematically,

$$S = a + bh \tag{100}$$

in which S = total steam consumed per hour.

h = ihp.

a and b are constants. a is the intercept on the S-axis, but it should be pointed out that h cannot be zero.

The curve shows the steam consumption per indicated horsepower-hour.

Figure 99 shows the variable-load performance of an Ames uniflow engine having a cut-off governor. It will be noticed that the total steam-consumption line is not a straight line for this engine but curves slightly upward. *Willan's law does not hold for cut-off governors*

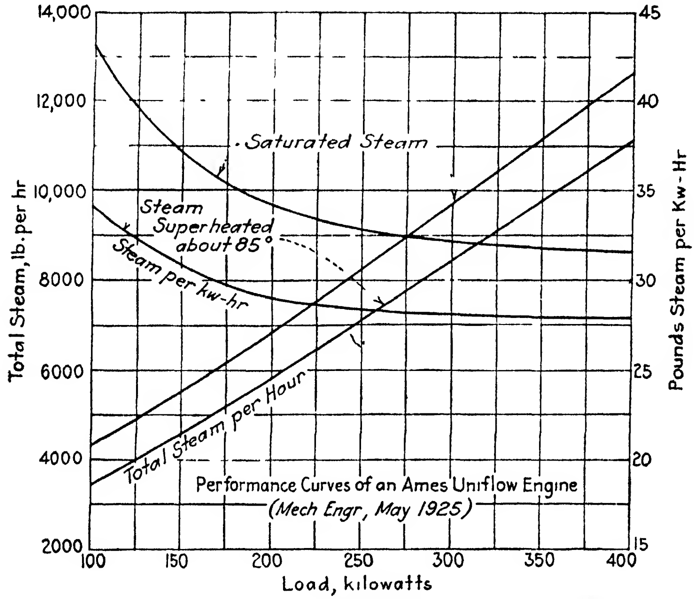


FIG. 99. Performance curves of an Ames uniflow engine.

119. Duty of Steam Pumps.—The economy of steam-driven pumps is usually expressed in terms of foot-pounds output of work done by the pump per 1,000,000 Btu supplied to the engine driving the pump. In effect, this is the reciprocal of heat economy, which is Btu per horsepower-hour. Duty is also sometimes stated in terms of the number of foot-pounds of work output of the pump per 1,000 lb steam supplied, but, since the engine is ordinarily charged with, roughly, 1000 Btu per lb of steam, these statements are approximately the same.

The 1925 A. S. M. E. test code says:

The duty in foot-pounds of work done per 1,000,000 Btu is found by multiplying the number of pounds of water pumped during the test by the average total dynamic head in feet, dividing the product by the number of Btu supplied, and multiplying the quotient by 1,000,000.

It should be noted that if the duty is stated as *foot-pounds of work per Btu*, we have a simple expression for efficiency by dividing this figure by 778.

Table XI shows what duty may be expected from various kinds of pumping engines.

TABLE XI.—DUTY OF PUMPING ENGINES

Type of pumping engine	Duty, in millions of ft-lb per 1,000,000 Btu
Small direct acting; noncondensing	10 to 20
Large direct acting; noncondensing	15 to 30
Large direct acting; compound condensing	20 to 60
Large flywheel; simple condensing	50 to 80
Small flywheel; compound condensing	60 to 100
Large flywheel; compound condensing	100 to 150
Large flywheel; triple expansion condensing	150 to 190

Since the duty of a pumping engine includes the mechanical losses in both the pump and the engine, the efficiency of the large flywheel pumping engines of this table, based on indicated horsepower, is remarkably high, 22 to 25 per cent being reported in several tests.

120. The Field of the Steam Engine.—The reciprocating steam engine may be characterized as a low-speed, variable-speed, high-torque, reversible engine and as such is particularly adapted for direct drives for low-speed machinery, for marine and locomotive engines, for hoisting engines, and for small isolated plants where simplicity and reliability are of prime importance. At the present time it is not generally practicable to build steam engines in large sizes because turbines can be built and installed more cheaply per horsepower, although in 1926 a 14,000-hp engine was built for steel-mill use.

Problems

1. A 16 by 36-in., simple, double-acting, steam engine runs at 100 rpm. Initial pressure is 125 lb gage and the back pressure 1 lb gage with a 29-in. barometer. Assuming a cut-off of 0.3 and a card factor of 85 per cent, calculate the rated cylinder horsepower of the engine.

2. A 12 by 18-in., simple, double-acting engine is designed to run at 200 rpm at atmospheric exhaust. With an 80 per cent card factor and cut-off at 0.35, what would be the cylinder horsepower with steam supplied at 60, 80, 100, 120, 140 lb per sq in. gage? Barometer 30 in.

3. The 12 by 18-in. engine of Prob. 2 is operated with the 100-lb steam. Calculate the cylinder horsepower with the engine running condensing with vacuums of 26, 22, 18, 14 in.

4. A simple, double-acting engine is to be designed to develop 500-cylinder hp with an average piston speed of 700 ft per min. Initial steam pressure 125 lb gage, cut-off at 25 per cent stroke. Ratio of stroke to bore 1.6. Card factor 85 per cent. Calculate the bore and stroke if the engine is to run at (*a*) atmospheric back pressure; (*b*) a vacuum of 26-in. barometer 30 in.

5. A simple, double-acting engine is to be designed for 100-cylinder hp at 100 rpm, 600 ft per min piston speed, cut-off at 25 per cent of stroke. Assuming a card factor of 80 per cent, 100 lb gage initial pressure, and atmospheric exhaust at 29-in. barometer, calculate the cylinder dimensions.

6. An 8 by 18-in. steam engine has a clearance of 6 per cent. (*a*) What is the actual ratio of expansion when the cut-off occurs at 25 per cent of stroke? (*b*) What is the volume of steam in the cylinder at cut-off?

7. A 12 by 16-in. engine has a clearance of 7.3 per cent and cuts off at 23 per cent of stroke. What is (*a*) the volume of steam in the cylinder at cut-off; (*b*) the real ratio of expansion; (*c*) the weight of wet steam in the cylinder at 100 lb per sq in. abs and 75 per cent quality?

8. Dry saturated steam at 100 lb per sq in. abs expands adiabatically to 10 lb abs. Calculate the quality of the steam after expansion and check by the Mollier chart.

9. Steam at 125 lb gage and 95 per cent quality expands adiabatically to atmospheric pressure when the barometer is 28.7 in. Calculate the quality after expansion and check by the Mollier chart.

10. Steam at 250 lb per sq in. abs and 100° superheat is expanded at constant entropy to 15 lb abs. Calculate the quality after expansion and check by the Mollier chart.

11. Superheated steam at 385 lb per sq in. abs and 600°F. is expanded adiabatically to 75 lb abs. What is the final condition of the steam? Check by means of Mollier's chart.

12. A boiler delivers steam at 143 lb gage and 98.5 per cent quality. Barometer, 29.6 in. This steam is expanded at constant entropy into a vacuum of 19 in. What is the quality of steam at the end of the expansion?

13. A steam engine uses dry steam at 165 lb per sq in. abs and exhausts at 15 lb abs. Calculate the efficiency of the Rankine cycle for the same conditions.

14. A steam turbine using superheated steam at 220 lb gage and 190° superheat exhausts at a vacuum of 28.5 in. when the barometer is 29.9 in. Calculate the efficiency of the corresponding Rankine cycle.

15. What would be the efficiency of a Rankine cycle operating between 1,200 lb per sq in. abs. and 700° total temperature and 1 lb abs?

16. A steam engine uses 22 lb steam per indicated horsepower per hour. Initial steam condition 120 lb per sq in. gage and 50° superheat. Barometer is 28 in., and the vacuum is 24 in. Calculate (*a*) the actual thermal efficiency; (*b*) the Rankine-cycle efficiency; (*c*) the efficiency ratio.

17. A steam engine used 33.3 lb steam per indicated horsepower-hour. Initial steam condition, 140 lb per sq in. abs and 99 per cent quality. Back pressure, 17 lb abs. Calculate (*a*) the actual thermal efficiency; (*b*) the Rankine-cycle efficiency; (*c*) the efficiency ratio.

18. A direct-connected engine-generator set generates 495 kw and uses 11,140 lb steam per hour. Initial condition of steam, 160 lb gage and dry. Barometer, 29.95 in.; vacuum, 26.1 in.; efficiency of generator, 92.4 per cent; and mechanical efficiency of engine, 90.3 per cent. Calculate (*a*) the actual thermal efficiency of the engine based on indicated horsepower; (*b*) the Rankine-cycle efficiency; (*c*) the efficiency ratio.

19. During a test of a double-acting, simple, 12.5 by 15-in. steam engine, the following data were taken: area of head-end card, 0.88 sq. in.; crank-end card, 0.83 sq in.; diameter of piston rod, 2 in.; length of cards, 3.11.; scale of spring, 79.5 lb. A Prony brake having an arm 60.25 in. long gave a tare weight of 55 lb and a gross load of 231 lb. Steam pressure was 100 lb per sq in. abs, and the back pressure, 4 lb abs. The calorimeter test on the supply steam showed 230°F. in the calorimeter and atmospheric pressure at 29.1 in. barometer. The engine ran at 260 rpm and used 3,150 lb steam per hour. Calculate (*a*) the indicated horsepower; (*b*) the brake horsepower; (*c*) the mechanical efficiency; (*d*) the quality of the supply steam; (*e*) the actual thermal efficiency based on indicated horsepower; (*f*) the Rankine-cycle efficiency; (*g*) the efficiency ratio.

20. A 24 by 30-in. simple engine running at 175 rpm drives a generator which, during a test, delivered 310 kw to the switchboard. Generator efficiency 91 per cent. Steam pressure, 166 lb per sq in. abs; 38.1 lb steam per kilowatt-hour. Vacuum into which engine exhausts, 26 in., barometer 30 in.; head-end, indicator-card area, 1.62 sq in., crank end, 1.66 sq in.; length of cards, 3.16 in.; scale of spring, 80 lb; piston rod, 2.5 in. diameter. Throttling calorimeter pressure, 2.5 in. above atmosphere. Calorimeter temperature, 289°F. Calculate (*a*) the indicated horsepower; (*b*) the shaft horsepower; (*c*) the mechanical efficiency; (*d*) quality of steam supplied; (*e*) actual thermal efficiency; (*f*) the Rankine-cycle efficiency; (*g*) the efficiency ratio.

21. A manufacturer guaranteed an engine to produce an indicated horsepower-hour on 20 lb dry steam at 100 lb gage and exhausting into a 24-in. vacuum at sea level. Under test it took 21 lb steam at 93 lb gage and 25-in. vacuum. The engine is to carry a load of 140 ihp for 3,000 hr per year using steam which costs 20 cts. per 1,000 lb. Assuming interest at 6 per cent and depreciation at 5 per cent, how much should be deducted from the purchase price?

22. An ingenious engineer bought exhaust steam at 1 lb gage for $100 per year to run his low-pressure, 20 by 30-in., 90 rpm engine exhausting into a 26-in. vacuum referred to a 30-in. barometer, cut-off at 30 per cent, card factor 75 per cent. How much did his power cost him per horsepower per year?

23. A 9 by 15-in. engine has a clearance of 10 per cent. Cut-off at 25 per cent; speed 200 rpm. It uses 1,128 lb steam per hour at 100 lb per sq in.

gage; barometer, 14.7 lb. At a point on the compression curve at 12 per cent of stroke before dead center the pressure is 5 lb gage. Find the quality of the steam at cut-off, assuming $x = 100$ per cent at the point on compression curve.

24. A pumping engine delivers 20,000,000 gal water against a total head of 140 ft every 24 hr. It uses 5,700 lb dry steam per hour at 132 lb per sq in. abs and exhausts into a condenser in which the pressure is 1.5 lb per sq in. abs. What is (*a*) the "duty" per 1,000,000 Btu; (*b*) the steam rate per water horsepower-hour; (*c*) the over-all thermal efficiency; (*d*) the Rankine-cycle efficiency?

25. A pumping engine delivers 19,400,000 gal water per 24 hr against a discharge gage pressure of 59.5 lb per sq in. The suction gage, which is 5 ft below the discharge gage, shows a vacuum of 16 in. The indicated horsepower being developed in the cylinder is 498. The steam rate is 15 lb per ihp-hr. Steam-supply pressure 115 lb per sq in. abs, and the quality 99.6 per cent. The engine exhausts at a pressure of 0.95 lb abs. Calculate (*a*) the duty per 1,000,000 Btu; (*b*) the over-all thermal efficiency; (*c*) the thermal efficiency based on indicated horsepower; (*d*) the Rankine-cycle efficiency; (*e*) the efficiency ratio.

26. During a test of a 200-kw, d-c, direct-connected, engine-generator set, the following data were taken:

Load	1	2	3	4	5
Volts	240	239	240	240	238
Amperes	1,010	812	705	500	300
Total steam per hour	9,900	8,400	7,600	6,100	4,650

Steam pressure at throttle, 125 lb gage; quality, 99 per cent; barometer, 29.5 in.; vacuum, 24.3 in. average.

At the factory, the generator showed the following efficiencies:

Load, kilowatts	200	150	100	50
Efficiency, per cent	92.5	88.6	85.1	75.4

A friction-horsepower test of the engine alone gave 17 hp loss at full load and 15 at quarter load.

(*a*) Plot curves with kilowatts as abscissas and generator efficiencies and engine friction horsepower as ordinates. Compute for each load (*b*) kilowatts output; (*c*) shaft horsepower; (*d*) indicated horsepower; (*e*) steam rate per indicated horsepower-hour; (*f*) actual thermal efficiency based on indicated horsepower. (*g*) Plot on same sheet with curves of (*a*) total steam (Willan's line), steam per indicated horsepower-hour, and thermal efficiency against load.

27. A 14 by 16-in., high-speed engine rated at 173 ihp is guaranteed to give the following steam consumption per indicated horsepower-hour, at 150 lb gage and exhausting at 10 lb, gage, back pressure:

Load	173	130	87
Steam per indicated horsepower hour	25.6	25.0	27.0

Normal barometer is 25 in.

Calculate (*a*) the guaranteed heat efficiencies; (*b*) the Rankine-cycle efficiency. (*c*) Plot steam rates and efficiencies against load.

28. A 13 by 15-in. uniflow engine is guaranteed to give the following steam rates per indicated horsepower-hour:

Load, i.hp.	30	60	90	120
140 pounds gage	21.5	19.3	19.2	19.7
150 pounds gage	21.3	19.1	19.0	19.4

Exhaust pressure atmospheric at a barometer of 25 in.

Calculate (*a*) the guaranteed thermal efficiencies; (*b*) the Rankine-cycle efficiency. (*c*) Plot steam rates and efficiencies against load.

CHAPTER XIII

IMPROVING STEAM-ENGINE PERFORMANCE

121. Losses in Steam Engines.—The energy losses in reciprocating steam engines are three.

1. *The Exhaust.*—It will be remembered that the latent heat of low-pressure steam is greater than that of high-pressure steam. This means that steam after expanding in an engine cylinder, even though it may be wet, still contains a large quantity of heat that is either thrown away to atmosphere or given up in the condenser. If, for example, an engine exhausts at atmospheric pressure and the exhaust steam contains 15 per cent moisture, the heat thrown away would be $180 + 0.85 (970.3) = 1005$ Btu per lb of water. If the enthalpy of the steam entering the engine was, say, 1200 Btu, the percentage loss in the exhaust would be $(1{,}005 \div 1{,}200) \times 100 = 83.8$ per cent.

In practice, this loss varies from 70 per cent in good engines to 90 per cent in poor ones. In some plants this exhaust heat can be salvaged for heating purposes, but, even in these cases, the saving thus made cannot be credited to the engine but to the plant as a whole.

It would be highly desirable from the point of view of thermal efficiency if it were possible to exhaust the steam as water direct from the engine after converting the latent heat into work. So far, this has not appeared possible of accomplishment. In general, however, engines which have the wettest exhaust should be the most efficient if the wetness is caused by the conversion of heat into work, and those which have the driest exhaust should be the least efficient. Five factors that tend to make the exhaust loss high are (1) leakage past piston and valves, (2) incomplete expansion, (3) high back pressure, (4) wiredrawing, and (5) cylinder condensation and reevaporation. Each of these is dealt with in the articles that follow.

2. *Radiation and Convection from the Cylinder.*—This loss is really very small in comparison with the exhaust loss, amount-

ing to 1 to 5 per cent of the heat supplied. Radiation is reduced by insulating, or "lagging," the cylinder with magnesia, mineral wool, or other high-temperature insulation.

3. *Mechanical Friction.*—The mechanical efficiency of steam engines may vary from 80 per cent in poor engines to 96 per cent in good ones, leaving mechanical friction for poor engines about 20 per cent and for good engines 4 per cent of the indicated horsepower. Now, since the indicated thermal efficiency varies in these poor and good engines from 5 to 25 per cent, respectively, the friction loss in terms of the heat supplied is about 5 per cent of 20 per cent = 1 per cent, or 25 per cent of 4 per cent = 1 per cent of the heat supplied.

That is to say, the friction loss and radiation loss are relatively very small, but the exhaust loss is very large. Any reduction, therefore, that can be made in the exhaust loss will be well rewarded. Practically all improvements in steam engines in the direction of improving efficiency have had as their objective the reduction of the heat carried away by the exhaust. The five factors mentioned above which tend to keep the exhaust dry and the remedies that have been devised for their reduction will now be discussed.

122. Leakage of Valves and Piston.—Leakage past the piston and valves allows the steam to expand to the lower pressure in the exhaust pipe without doing useful work. The amount of this leakage in new engines is usually very small, but tests quoted by Dean and Wood[1] indicate that after from 1 to 5 years of service the steam rates of various engines may be from 1 to 45 per cent higher than for the same engines when new.

Leaky valves may be caused by warping of the valves to high temperature or by wear of valves or seats.

123. Incomplete Expansion.—Theoretically, it would be desirable if the steam could be expanded completely down to the pressure in the exhaust pipe, as is assumed to be done in the Rankine cycle. In practice, however, this is not feasible, for the reasons (1) that the work done by the thin toe of the *P-V* diagram is very small and the increase in size and cost of the engine would not be justified; and (2) that, as the load is varied, the expansion might be complete at light loads but not at heavier loads when cut-off is delayed.

[1] *Trans., A.S.M.E.*, 30: 6.

In extreme cases, we may have no expansion at all, as, for example, in the direct-acting pump, which admits steam for the full stroke, or in a Corliss engine when laboring under a heavy load with the valves "hooked up" so as not to cut-off; in these cases the efficiency is notoriously poor. In general, the "drop" at release should be as small as possible, but, if expansion is carried too far, as at light loads, we may actually have a rise at release instead of a drop, thus causing a loop of negative work. For simple engines the most economical ratios of expansion are between 3 and 5, and for multistage engines between 6 and 36. Too great a ratio of expansion in a single cylinder leads to excessive cylinder condensation (see Art. 126).

124. Back Pressure.—Figure 100 shows the pressure-volume diagram for the adiabatic expansion of 1 lb steam from 200

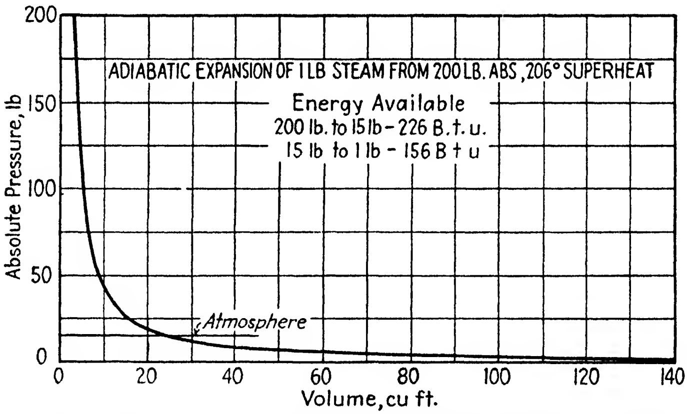


FIG. 100. Pressure-volume relation for adiabatic expansion of steam.

lb abs and 206° superheat to about 2 lb abs. The area under the curve down to atmosphere represents 226 Btu or 176,000 ft-lb, while the area below atmosphere down to 1 lb abs represents 156 Btu or 121,000 ft-lb. That is, the work available below atmospheric pressure is a very large part of the total available work, and the lower the back pressure can be made the more economical will be the engine. Unnecessary back pressure, therefore, causes a large avoidable loss.

The back pressure is reduced by condensing the exhaust steam. The problem and machinery of condensing will be discussed in Chap. XVI.

Barrus[1] shows that the simple condensing four-valve engine may save as much as 26 per cent over noncondensing engines of the same type, whereas compound condensing four-valve engines may save 35 per cent over four-valve, noncondensing compounds. Engines using superheat are 20 to 30 per cent more efficient when operating condensing than when noncondensing

With reciprocating engines the practical limit of vacuum referred to 30-in. barometer is 26 or 28 in., because the volume of steam to be handled becomes very large at lower back pressures and the exhaust passages inordinately large

125. Wiredrawing.—It was pointed out in Art. 108 that wiredrawing rounds the corners of the indicator card and thereby reduces the capacity of the engine somewhat. Another and more serious result of wiredrawing is the reduction of the efficiency of the engine. Wiredrawing is a throttling process in which the steam passes a constriction and thereby suffers a reduction in pressure and temperature but at the same time keeps its enthalpy unchanged. In other words, the process is similar to the action in the throttling calorimeter except to a much smaller degree Figure 101 illustrates the effect of throttling. Suppose dry saturated steam at A, 160 lb abs, is throttled to C, 120 lb abs, this throttling being at constant enthalpy, 1195 Btu, but with an increase in entropy from 1.563 to 1.594. An ideal engine exhausting at 2 lb abs would have energy available along the adiabatic line CD from 1195 to 925 Btu, that is, 270 Btu. Without throttling, the ideal engine would have available $1195 - 907 = 288$ Btu, a difference of 18 Btu, the percentage loss due to throttling being $(18/288) \times 100 = 6.3$ per cent. Wiredrawing thus reduces the *availability* of the energy by reducing the pressure and temperature. Since the enthalpy is constant in throttling and the temperature is reduced, the entropy must increase and an increase in entropy for the same enthalpy always means a reduction in availability.

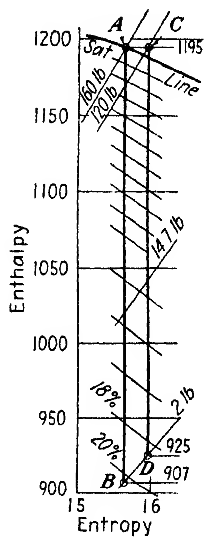


FIG. 101.—Enthalpy-entropy diagram illustrating effect of throttling.

[1] "Engine Tests."

126. Condensation and Reevaporation.—When steam is expanded in the engine cylinder, its temperature drops, and this drop, in turn, lowers the temperature of the cylinder walls. The greater the ratio of expansion the lower will be the temperature at the end of expansion and, consequently, the cooler will the cylinder walls become during the exhaust stroke. Now, as fresh steam is admitted to the cylinder at the beginning of a new power stroke, some of it is condensed by the cool cylinder walls and the walls have their temperature raised to near that of the entering steam. As the piston uncovers more cool wall, the condensation continues until cut-off.

When expansion begins, the temperature of the steam begins to drop, but the drop is not quite so rapid as it would be if there were no hot water in suspension in the steam and on the walls and if the cylinders did not give their heat back to the steam. By the time the end of the expansion stroke is reached, the walls are actually giving heat back to the steam and even evaporating some of the moisture. The drying action continues during the exhaust stroke, and by the time of the next admission the walls are probably dry and relatively cool.

The effect, then, of this alternate heating and cooling of the walls is to set up heat waves traveling outward through the thickness of the metal, the amplitude of the waves diminishing very rapidly as they proceed through the metal. In most engines the wave amplitude is probably very nearly zero within a few hundredths of an inch from the inner surface, but, thin as this film of metal is, it acts as a heat sponge to store and release rather large quantities of heat. This absorption of heat from the steam at the beginning of the cycle and then the returning of it to the steam being exhausted, after the steam has done its work, obviously represents a heat waste.

The remedies that have been used to reduce the loss due to initial condensation and reevaporation have been many and various, the most common of which are of sufficient importance to be discussed here at some length. These remedies are (1) jacketing, (2) compounding, (3) superheating, (4) reheating between stages, (5) the uniflow principle.

127. Jacketing.—The heat waves inside the cylinder walls might be made of less effect if the *outside* of the walls were kept so hot that the amplitude of the wave would be reduced. Steam jackets around the cylinder have been used in the attempt to keep the inside of the walls hot, but it is doubtful whether jackets can be made to reduce initial condensation materially except for slow-speed engines and high ratios of expansion. The outside surface of the jacket is, moreover, hotter than the cylinder itself would be, and the external area of the jacket is greater than that of the cylinder; therefore, the loss by *radiation* must be greater than for an unjacketed cylinder.

The steam used by the jacket must, of course, be charged against the engine, and this steam must be live steam. Attempts have been made to jacket the cylinder with exhaust steam, but this has a tendency to *cool* the cylinder rather than *heat* it.

In nearly all tests on jacketed engines, the jacket shows some net gain, although there is great diversity in the amount of gain reported. From 0 to 30 per cent saving has been reported, but the majority of tests show a saving of less than 10 per cent, and some tests at low ratios of expansion show a negative saving.

Some engines have only the cylinder heads jacketed, since this is the part that is exposed longest to both hot and cold steam. Uniflow engines generally have jacketed cylinder heads.

128. Compounding.—It was shown in Art. 123 that high ratios of expansion tend toward high economy, but high ratios of expansion also tend toward an increase of cylinder condensation, because the temperature range in the cylinder is large and the cylinder volume and surface are also larger. One of the very early improvements of the steam engine was an arrangement whereby the expansion should take place in two or more cylinders or stages in order that the temperature range should be smaller in each stage.

Compounding not only reduces the temperature range per cylinder but also reduces the high-pressure (H.P.) clearance volume to a much smaller percentage of the total final volume of the steam than if the expansion had all occurred in one large cylinder. It also reduces the surface area of this clearance space. High-pressure clearance steam and leakage steam, moreover, are used in the succeeding stages and so represent less loss than in the simple engine.

The apparent ratio of expansion or "number of expansions" of a compound engine is the ratio of expansion in the high-pressure cylinder multiplied by the ratio of the low-pressure (L.P.) cylinder displacement to that of the high-pressure cylinder.

Consider Fig. 102, in which *A* represents a cylinder with a piston having a 5-sq ft area and a 4-ft stroke. The clearance volume, including the ports, is represented by the space at the ends, which we assume to be 5 per cent. If this cylinder is operated as a simple engine with an apparent ratio of expansion of 10, cut-off would occur at one-tenth stroke, and the actual volume of steam in the cylinder at cut-off would be 15 per cent of the displacement. The real ratio of expansion would then be $^{105}/_{15}$ = 7 instead of 10. The clearance volume would be 5 per cent of 20 cu ft, or 1 cu ft, and the surface area of the clearance space would be something over 5 sq ft.

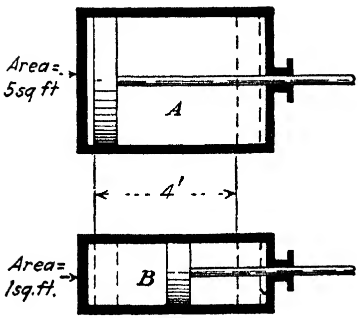


Fig 102

If the engine were compounded by adding the high-pressure cylinder *B* having a piston area of 1 sq ft and the same stroke as *A* the clearance of which is likewise 5 per cent, the actual high-pressure clearance volume would be only 5 per cent of 4 cu ft, or $^1/_5$ cu ft, and the surface area of the clearance space something over 1 sq ft. Then, if the cut-off in the high-pressure cylinder is at half stroke, the apparent number of expansions will be $2 \times 5 = 10$, as for the simple engine. The real ratio of expansion, however, is now

$$\frac{20 + 1}{2 + \frac{1}{5}} = \frac{21}{2.2} = 9.55$$

This improvement in ratio of expansion means more nearly complete expansion.

The gains by compounding are partly neutralized, however, by the fact that the radiating area of the machine is increased by that of the high-pressure cylinder *B* and the steam passages between cylinders. Radiation is only about 1 to 5 per cent. For pressures under 100 lb and ratios of expansion of less than 5, the simple engine is more economical, but for pressures above 100 lb and expansion from 5 to 15 it pays to use compound cylinders. For ratios above 15 it is generally best to use triple expansion, that is, three stages of expansion. Noncondensing compound engines are not considered desirable, especially since the advent of the uniflow engine.

129. Types of Compound Engines.—One way to arrange a compound engine is with the cylinders side by side and with the cranks placed on the shaft at right angles to each other; that is, when one piston is at the end of its stroke, the other is about at the middle, as shown in Fig. 103. With the cranks at 90 deg, it is

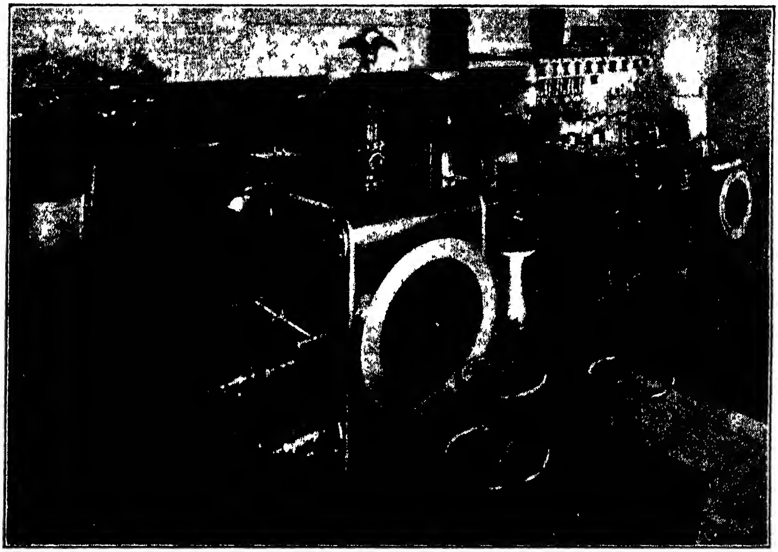


FIG. 103.—Three-hundred-hp Chuse cross-compound, nonreleasing Corliss engine

necessary to have a *receiver* between the high-pressure exhaust and the low-pressure inlet in order to allow the high-pressure cylinder to exhaust after cut-off in the low-pressure cylinder and to allow the low-pressure cylinder to receive steam when the high-pressure cylinder is not exhausting. This arrangement of cylinders is called cross compound. It has the advantage of giving a power impulse every 90 deg of crank angle but has the disadvantage of requiring the extra connecting rod and crank and of requiring much floor space.

The action of the steam in the cylinders of a cross-compound engine with a receiver is illustrated by the indicator diagrams of Fig. 104(*a*). High-pressure exhaust along *DE* is into the receiver against a rising pressure, until, at midstroke, admission occurs in the low-pressure cylinder at *G*. Then high-pressure exhaust continues into both the receiver and low-pressure cylinder until high-pressure compression at *F*. Low-pressure cut-off from the receiver occurs at *J*, and the receiver pressure is left at about the level of *JD* where the high-pressure exhaust found it.

The proportion of work done by each cylinder is adjusted by means of the low-pressure cut-off. To increase the work of the low-pressure cylinder, make the low-pressure cut-off come *earlier*. This will leave the receiver pressure higher and, in turn, cause the line *DEF* to be higher, which reduces the work in the high-pressure cylinder and increases that in the low-

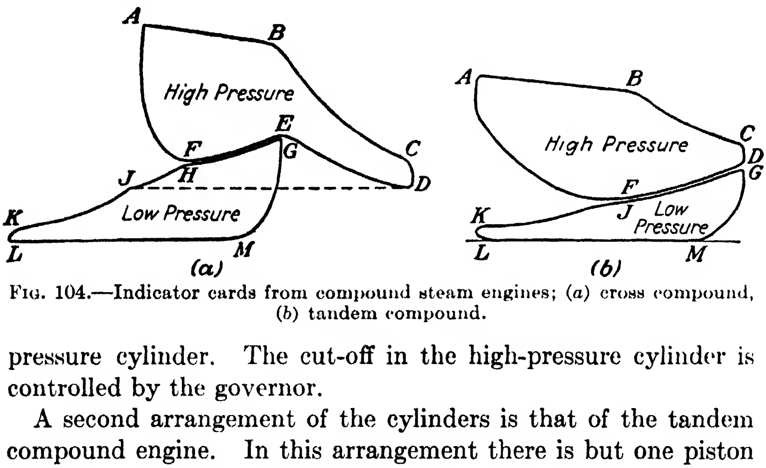


FIG. 104.—Indicator cards from compound steam engines; (*a*) cross compound, (*b*) tandem compound.

pressure cylinder. The cut-off in the high-pressure cylinder is controlled by the governor.

A second arrangement of the cylinders is that of the tandem compound engine. In this arrangement there is but one piston rod, one connecting rod, and one crank. The tandem compound has the advantages of simplicity and small floor space and the fact that no receiver is necessary. It has the disadvantage, however, that it delivers only two power impulses each revolution and may stop on dead center.

Figure 104(*b*) shows a pair of indicator cards taken from a tandem compound engine with no receiver other than the pipe connecting the two cylinders. With such an engine it is not necessary to have an admission valve on the low-pressure cylinder because the high-pressure exhaust valve controls low-pressure admission of steam and the high-pressure compression *F* is simultaneous with the low-pressure cut-off *J*.

130. Work Done by Compound Engines.—The work done by compound engines per cycle is the sum of the work of the two cylinders indicated separately. The rated indicated horsepower is obtained in the same manner as for the simple engine, except that the apparent over-all ratio of expansion, or number of expansions as discussed in Art. 110, is used.

An equivalent single indicator card is sometimes constructed by shortening the high-pressure card by the ratio of cylinder volumes and lowering the low-pressure card by the ratio of

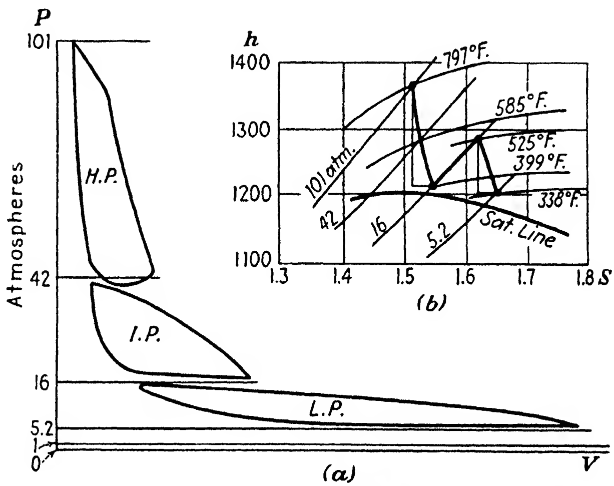


FIG. 105.—Equivalent combined indicator cards (*a*) and Mollier diagram (*b*) for a triple-expansion engine. (*V. D. I. Zeitschrift*, V79, *p.* 487.)

indicator springs. Figure 105 shows such an equivalent indicator card for the triple-expansion engine discussed in Art. 134.

131. Superheating.—It is a well-known fact that heat absorption by wet surfaces is greater than by dry surfaces. If the supply steam is superheated so that the cylinder walls remain dry during admission, the absorption of heat by the walls will be much reduced, and consequently the heat given back to the steam by the walls during exhaust will be correspondingly reduced. It could be shown that the efficiency of a Rankine cycle using superheated steam is very little better than for saturated steam, but the reduction in initial condensation is ample justification for its use. Results of published tests with superheated steam show that the economy of an engine is improved almost directly in proportion to the amount of superheat. Figure 99 (Chap XII) shows the improvement of a uniflow engine using steam superheated about 80°. Figure 106 shows the improvement in steam rate of a 500-hp, triple-expansion, marine engine with moderate superheat. Superheat to 600° total temperature may cause a saving of 20 per cent in simple engines, 15 per cent in compound, and 8 per cent in triple-expansion engines. Naturally, the saving would be greatest in those engines which have the greatest loss due to initial condensation when no superheat is used.

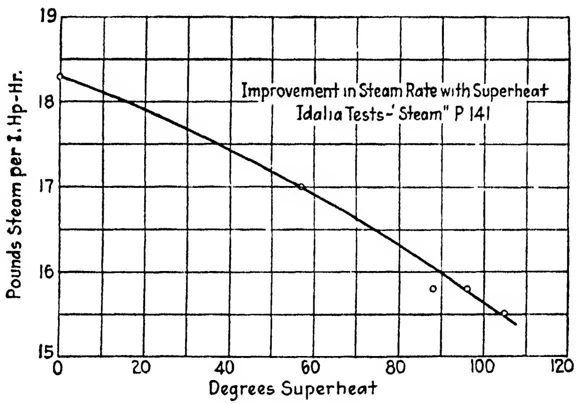


FIG. 106 Reduction in steam rate with superheat

The use of superheated steam has a tendency to cause warping of valves and cylinders and to make lubrication difficult. A total temperature of 600° is about the limit for advantageous use in reciprocating engines, those having poppet valves being able to handle hotter steam than those with either Corliss or slide valves.

132. Reheating between Stages.—The receivers of compound engines are sometimes heated with steam coils through which high-pressure and -temperature steam passes. The advantage of this reheating of the steam is exactly similar to that obtained by superheating for the high-pressure cylinder or for simple engines. Marks[1] has shown that the economy of compound engines may be improved 6 to 8 per cent by superheating in the receiver 30 to 100°. To be satisfactory, superheating receivers

[1] *Trans., A.S.M.E.*, vol. 25.

must be provided with separators to remove the moisture from the working steam before reheating; otherwise, the purpose of reheating is defeated. Figure 107 shows a sectional view of a reheating receiver.

As in the case of jacketed cylinders, the heat used in the reheaters must be charged against the performance of the engine. Reheating and superheating increase the capacity of the engine as well as its economy.

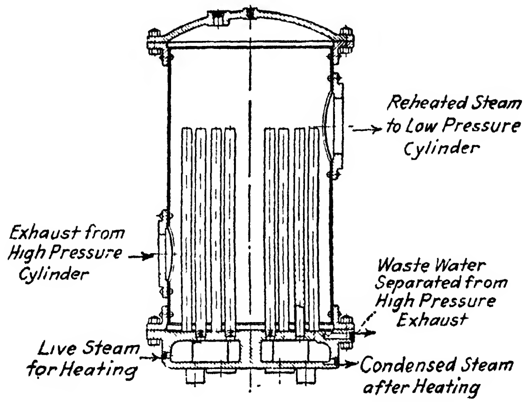


FIG. 107.—Nordberg reheating receiver.

133. The Uniflow Principle.—Figure 108 illustrates a type of cylinder which has for its primary objective the reduction of cylinder condensation. In the engines so far described, the live steam and exhaust steam pass alternately in opposite directions over the clearance surfaces. In the uniflow engine, the steam is admitted through the short passages between the inlet valve *B* and the cylinder. The length of the piston *C* is about 0.9 of the stroke, so that, when 0.9 of the stroke is completed, the ring of exhaust ports *D* in the middle of the cylinder is uncovered and the steam is released and exhausted through *E*. In this way, the steam enters at the end of the cylinder and moves toward the center, never flowing in the reverse direction; hence the name "uniflow." When release occurs, the steam flows directly over the surface of the cylinder walls and the face of the piston, and any moisture present is swept out with the exhaust.

Figure 109 shows indicator diagrams from a typical uniflow engine. Test curves were shown in Fig. 99, Chap. XII. In these diagrams, cut-off occurred at about 15 per cent of stroke and release at about 90 per cent. Compression began at 10 per cent of stroke and at the end of the stroke reached about 105 lb abs. The clearance was 17 per cent.

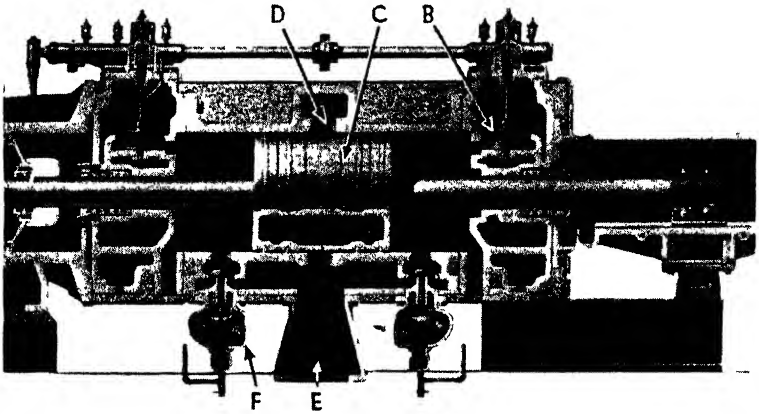


FIG. 108.—Skinner Universal Unaflow cylinder with floating piston.

In uniflow engines it is aimed to have the clearance volume such that compression will be carried to approximately the pressure of the live steam. When this is attained, no live steam will be used in bringing the pressure in the clearance volume up to admission pressure. For condensing operation, with very low

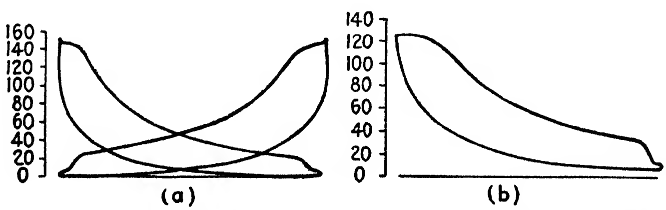


FIG. 109.—Indicator diagrams from a uniflow engine. (*a*) Throttle gage, 153 lb; back pressure, 0 lb; mep, 45.1 lb. (*b*) Throttle gage, 125 lb; back pressure, 6.7 lb; mep, 42.9 lb.

exhaust pressures, this means that the clearance volume must be very small, usually between 1 and 2 per cent, depending upon the ratio of steam pressure to exhaust pressure. For noncondensing operation it is necessary either to provide some method of delaying compression or else to provide for increasing the clearance volume. Various methods are used by the different manufacturers. The cylinder of Fig. 108 has valves *F* near the ends which are held open, thus allowing exhaust to continue for about two-thirds of the compression stroke. Other manufacturers provide a valve that relieves the compression if it is carried above the throttle pressure.

All modern uniflow engines have jacketed heads through which the supply steam passes on its way to the cylinder. In

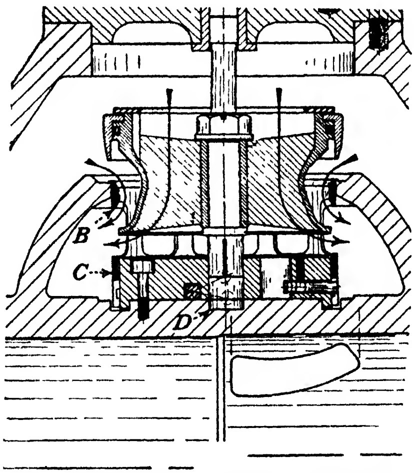


FIG. 110.—Skinner variable cut-off valve.

general, very little is to be gained by compounding, although some compound uniflows have been built.

One advantage of the uniflow engine is its great overload capacity due to the fact that cut-off may be delayed to 60 or 65 per cent of stroke. A variable cut-off valve developed for Skinner variable-speed engines is shown in Fig. 110. Above is the upper seat of the regular expansion compensating valve. *B* and *C* are the cut-off sleeves, on a spider that can be raised and lowered shown in its lowest position. As the spider with its sleeves is raised by a lever at *D*, cut-off is made earlier when the sleeves *B* and *C* meet the double-ported valve, see Art. 137. A recent 14,000-hp uniflow engine[1] built for blooming-mill service

[1] *Power*, 63: 986.

has a short-time capacity of 30,000 hp Figure 111 shows this engine during erection at the factory.

The history of the uniflow engine is of interest.[1] The first patent on the uniflow principle was taken out by Jacob Perkins in England in 1827, and in the same year a second patent followed on a compound uniflow. Perkins built two commercial pumping engines, one of 16 and the other of 10 hp, and operated them with steam at 700 to 800 lb per sq in. at a time when Watt placed 25 lb gage as the practical limit of pressure. A reported

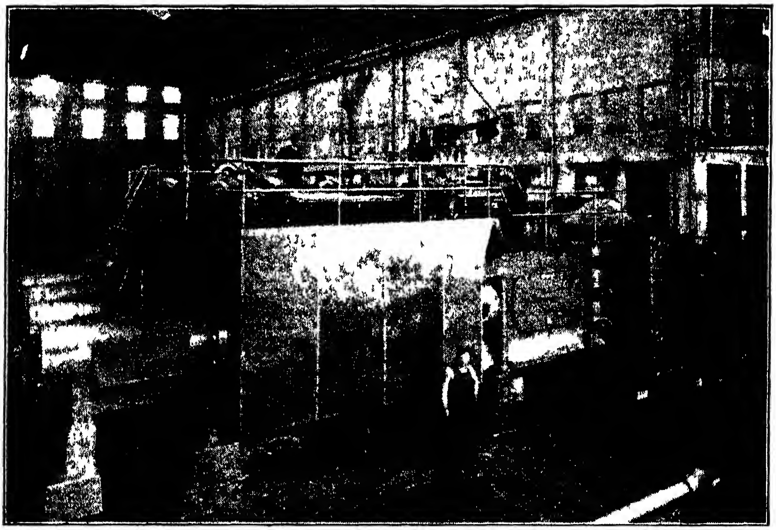


FIG. 111.—Nordberg uniflow engine (four-cylinder), 14,000 hp

test on one of these engines showed 1 bhp-hr with a coal consumption of 1.5 lb. An American patent was issued in 1872 on the uniflow principle, and in 1885 an English patent was issued to T. J. Todd, but the real development of the idea began with the work of Professor Stumpf of Charlottenburg, Germany, in 1908.

134. Size and Speed of Steam Engines.—There are two reasons why a large engine is apt to be more economical than a small one. (1) The workmanship is more apt to be of high grade on a large and important engine, although this is not necessarily so. The cost per horsepower for a large engine is less than for a small engine, and the manufacturer can afford greater refinement

[1] *Ibid*., May 6, 1924; p 718

in workmanship and design on the large engines. The conditions under which the large engine operates and the care it receives are usually better than for the small engine. (2) The cylinder wall surface is smaller in proportion to the volume in a large engine than in a small one because the volume varies as the cube and the area as the square of the dimensions.

Efficiency is more directly a function of piston speed because a high piston speed means a shorter exposure of a charge of steam to the metal of the cylinder walls and hence less heat transfer in this way.

While it is true that cylinder condensation is reduced by high rotative speed, because time is required for transfer of heat to the cylinder walls, the efficiency is not generally improved. This is partly because large engines cannot generally be run over 100 rpm and this leaves the high speeds to smaller engines. For high speeds, moreover, the designer is limited in the choice of valve gears, and the steam distribution may be imperfect. The best performance records are held by engines normally running at speeds below 100 rpm.

Tests with low-speed engines, however, do show that the efficiency varies directly with the speed.

In 1931, the Philip Carey Co. installed in its Lockland, Ohio, plant,[1] two 6,000-hp German-built engines which combine in a single machine several of the schemes mentioned in the foregoing paragraphs for improving performance. (1) The throttle pressure is 1,500 lb with a temperature of 780°F. (2) Triple expansion is used, the high pressure being single acting. (3) The steam is reheated before passing to the low-pressure cylinder. (4) The engine is heavily insulated. (5) Instead of exhausting at low pressure, the exhaust steam is used for process work in the plant. Performance data on this engine are given in Prob. 17, page 204, and indicator cards are shown in Fig. 105.

Problems

1. A 16 by 24-in. engine runs at 175 rpm. If the initial pressure is 130 lb per sq in. abs. and the exhaust is at 15 lb abs, assuming a card factor of 0.8, what would be the rated indicated horsepower for cut-offs at 0.2, 0.3, 0.4, and 0.5?

2. A 16 by 24-in. engine runs at 175 rpm. Assuming a card factor of 0.8 and cut-off at 25 per cent, what would be the rated indicated horsepower

[1] *Power*, April, 1933, p. 169.

for initial pressures of 80, 100, 120, and 140 lb per sq in. abs and 15 lb abs back pressure?

3. A 16 by 24-in. engine runs at 175 rpm. Assuming a card factor of 0.8 and a cut-off of 25 per cent, what would be the rated indicated horsepower for an initial pressure of 130 lb per sq in. abs and back pressures of 4, 10, 18, and 25 lb abs?

4. A 20 by 30-in., simple, noncondensing Corliss engine runs at 100 rpm. Its latest point of cut-off is at 40 per cent of stroke. Initial steam pressure 150 lb gage. By what percentage will the horsepower of the engine be increased on being removed from sea level to an altitude where the normal barometer is 23 in.?

5. A steam engine has an efficiency ratio of 88 per cent at full load. Initial steam conditions, 125 lb per sq in. abs; and 99.5 per cent quality. Back pressure, 16 lb abs. Calculate (*a*) the full-load steam rate per indicated horsepower-hour; (*b*) the heat consumption per indicated horsepower-hour. (*c*) If the exhaust is used for heating, calculate the heat consumption per indicated horsepower-hour chargeable to the engine.

6. Which engine uses the least heat per indicated horsepower-hour, a simple, noncondensing engine using 26 lb dry steam per horsepower-hour; initial pressure, 100 lb per sq in. abs; or a compound, condensing engine using 12 lb steam per horsepower-hour; initial pressure, 290 lb abs, and 350° superheat; back pressure 1 lb abs?

7. A noncondensing engine uses 22.5 lb steam per indicated horsepower-hour. Initial pressure, 145 lb gage and 60° superheat; exhaust at 2 lb gage; barometer, 30 in.

It is planned to operate this engine condensing at 26-in. vacuum, 125 lb gage, and 100° superheat. If the Rankine-cycle efficiency ratio is decreased 5 per cent by the reduction in initial pressure, increased 5 per cent by the increased superheat and increased 24 per cent by the reduction in back pressure, what will be the new steam rate?

8. An engine is supplied with steam at 150 lb per sq in. abs and 100° superheat, but on entering the engine the steam is throttled at constant enthalpy to 140 lb abs. The back pressure is 4 lb abs. What is the reduction in the available heat due to throttling and the reduction in the Rankine-cycle efficiency?

9. A 9 by 12-in. engine running at 228 rpm uses 712 lb steam per hour. Initial pressure 114 lb per square in. absolute and exhaust at 14 lb abs. Clearance 9.8 per cent. At a point on the indicator card 85 per cent of the compression stroke, the pressure on the compression line is 35 lb abs and quality assumed to be 100 per cent. (*a*) What is the weight of steam in the cylinder at cut-off? (*b*) If cut-off occurred at 22 per cent of stroke, what was the quality of the steam at cut-off?

10. An 18 and 28 by 30-in. compound engine running at 145 rpm cuts off at 25 per cent of stroke in the high-pressure cylinder. Initial steam pressure is 150 lb gage, exhaust vacuum 26 in., barometer 30 in. With a card factor of 0.80, what would be the rated indicated horsepower of the engine?

11. A 12 and 25 by 18-in. cross-compound engine runs at 200 rpm on steam at 130 lb gage and 98 per cent quality. Barometer 29.5 in. Exhaust vacuum 21.3 in. Card factor 0.85 with cut-off at 25 per cent of stroke (*a*) Calculate the rated indicated horsepower. The high-pressure cylinder mean effective pressures are 55.1 lb on head end and 58.9 lb on crank end. Low-pressure cylinder mean effective pressures are 12.3 lb on the head end and 13.6 lb on the crank end. Piston rods have diameter of 2.5 in. (*b*) Calculate actual indicated horsepower.

12. A 24.5 and 41.5 and 72 by 48-in. marine engine running at 88 rpm develops 2,800 hp with steam at 200 lb gage and a 25-in. vacuum. Cut-off in the high-pressure cylinder occurs at 76 per cent of stroke. (*a*) Calculate the card factor. (*b*) What is the total theoretical ratio of expansion?

13. A compound engine,[1] direct connected to a d-c generator, has a 17 by 30-in., high-pressure cylinder and a 32 by 40-in., low-pressure cylinder, with effective piston areas of 207.3 and 780.5 sq in., respectively. During a test the following data were taken: duration of test, 3 hr; generator load, 1,007 kw; initial steam pressure, 354 lb gage, temperature, 581°F.; receiver pressure, 70.1 lb gage, temperature after reheating, 390°F.; exhaust pressure, 2 lb gage, temperature, 219.5°F.; total condensate, including jacket and reheater condensate, 59,613 lb; barometer, 29.4 in.; nominal cut-off in high-pressure cylinder, 53 per cent, in low-pressure cylinder, 39.3 per cent; mean effective pressures: high-pressure cylinder, 203.4 lb; low-pressure cylinder, 42.1 lb; rpm, 119; indicated horsepower at no load, 57.2.

Calculate (*a*) high-pressure indicated horsepower; (*b*) low-pressure indicated horsepower; (*c*) high-pressure card factor; (*d*) low-pressure card factor; (*e*) total indicated horsepower; (*f*) steam rate per indicated horsepower-hour; (*g*) Btu per indicated horsepower-hour; (*h*) thermal efficiency referred to indicated horsepower; (*i*) Rankine-cycle efficiency; (*j*) engine-efficiency ratio, referred to Rankine cycle.

14. The clearance of the high-pressure cylinder of Prob. 13 is 4.3 per cent, and that of the low-pressure cylinder is 3.6 per cent. What was the

Load, approximately	1/4	1/2	3/4	4/4
Lower Superheat Tests				
Steam pressure gage	163	165	161	164
Superheat, °F	112	107	111	109
Vacuum, in. Hg	24.2	24.3	24.2	24.2
Lb steam per ihp-hr	11.67	11.82	12.10	12.30
Higher Superheat Tests				
Steam pres. gage	166	163	159	161
Superheat, °F	154	159	164	158
Vacuum, in. Hg	24.2	24.3	24.4	24.2
Lb steam per ihp-hr	9.24	10.06	10.72	11.22

[1] See *Power*, Aug. 23, 1927, p. 291.

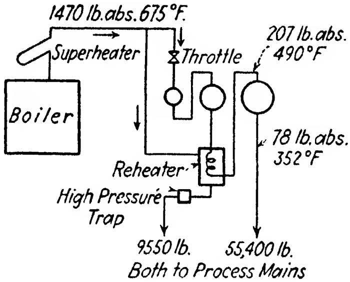
real ratio of expansion in (*a*) the high-pressure cylinder; (*b*) the low-pressure cylinder? (*c*) What was the total ratio of expansion?

15. The Delray plant of the Detroit Edison Co. increased the temperature of their 390 lb per sq abs steam from 700 to 1000°F. Exhaust pressure remained at 1 in. Hg abs. The heat consumption was decreased from 11,650 to 10,730 Btu per kw-hr. Calculate the Rankine cycle efficiency ratio before and after the change and check with the Mollier chart.

16. A Skinner marine uniflow engine test is reported as tabulated on page 203 (*Marine Eng. Ship.*, June, 1934):

Calculate for each test the thermal efficiency and determine the improvement due to increased superheat.

17. An actual test of the Philip Carey triple expansion engine of Art. 134 yielded the following data: (1) indicated horsepower of high-pressure and



low-pressure cylinders, 2,711; (2) indicated horsepower of low-pressure cylinders, 1,257; (3) shaft horsepower, 3,500; (4) kilowatt at switchboard, 2,450; (5) throttle steam, 55,400 lb per hr at 1,470 lb abs and 675°F.; (6) interheater reheated the inlet steam to the low-pressure cylinder at 207 lb abs to 490°F.; (7) the trap on the interheater discharged 9,550 lb water; (8) engine exhausted at 78 lb abs and 352°F. Calculate (*a*) the thermal efficiency of the engine alone; (*b*) mechanical efficiency of engine; (*c*) the efficiency of the generator; (*d*) plot the heat quantities for 1 lb throttle steam on the *H-S* diagram and trace on the problem sheet.

CHAPTER XIV

STEAM-ENGINE MECHANISMS

135. The D-slide-valve Engine.—The valve action of the simple slide-valve engine as described in Art. 94 is illustrated by Fig. 112.

Let *o* represent the center of the engine shaft; *oa*, the throw of the eccentric; and *ae*, the eccentric rod. As the engine shaft turns, the valve will move to the right of its mid position the distance *ob* and an equal distance to the left *oa*. The

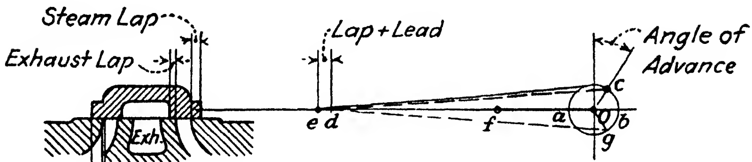


FIG. 112.—Relative positions of eccentric and crank.

line *of* represents the throw of the engine crank and in its present position is on head-end dead center. In order to have the steam ready for the piston as it is about to leave its head-end position, it is necessary to have the valve open some amount, as that marked "lead" in the diagram. This means that the valve must have been moved to the right a distance equal to the "lap" plus the lead; in order to accomplish this, the eccentric must be turned on the shaft through the angle *aoc*, and *e* will have moved to *d*, the crank and piston remaining at head-end dead center. That is, the eccentric is set somewhat more than 90 deg *ahead* of the crank. The purpose of the steam lap is to cut off the steam before the piston completes its stroke and, thus, to cause expansion. The purpose of the exhaust lap is to close the exhaust before the piston completes the exhaust stroke and, thus, to cause compression.

The simple slide-valve engine possesses several inherent defects the most important of which are as follows:

1. Since the slide valve has full throttle pressure on one side and exhaust pressure on a large part of the other side, there is a very large unbalanced pressure which causes much friction and rapid wear.

2. The opening and closing of the valve not being instantaneous causes the corners of the indicator card to be rounded, thus reducing the capacity of the engine.

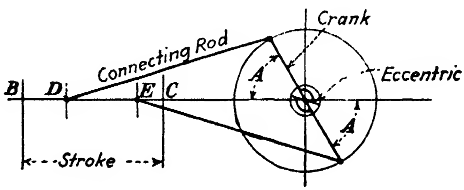


FIG. 113.—Effect of angularity of the connecting rod.

3. The long steam passages or *ports* from the valve to the cylinder constitute a large clearance volume which is wasteful of steam, see Art. 111, and furnishes extra surface for dissipation of heat from the cylinder.

4. *Unequal steam distribution* to the two ends of the cylinder due to the *angularity of the connecting rod.* Figure 113 shows that the distance BD, the distance that the crosshead moves from its head-end, dead-center position for a given crank angle A, is considerably larger than the distance CE corresponding to the same crank angle A from the crank-end, dead-center position. During these same two movements, however, the slide valve would have moved through nearly the same distance at both ends, because the eccentric rod is very long in proportion to the throw of the eccentric. This would cause the piston to be farther away from the end of the stroke at the head end than at the crank end for a given valve position.

5. The same steam ports, being used for both inlet and exhaust steam, increase the loss due to initial condensation; see Art. 126.

6. When used with high-temperature steam, slide valves will warp and become leaky.

To remedy these defects, various improved engine mechanisms have been devised which are described in the following articles.

136. Balanced Valves.—To overcome the unbalanced pressure on the back of the D-shaped slide valve, two improved types of valves have come into use, the *balanced valve* and the *piston valve*.

The balanced valve has a *pressure plate* at its back which carries the weight of the steam pressure while the valve slides underneath it. In Fig. 114 is shown such a balanced valve, and the low-pressure cylinder of Fig. 115 also has a balanced valve. The

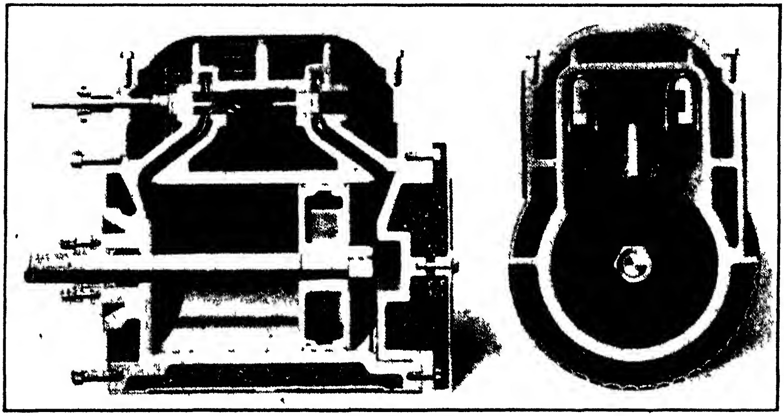


Fig. 114.—Elliott double-ported balanced valve

valve-pressure plate of Fig. 114 has a stiff, flat spring between the pressure plate and the steam-chest cover. This allows the valve to be lifted off its seat in case water should be trapped in the cylinder and thus prevents damage to the engine. The position of the pressure plate requires very nice adjustment to prevent leakage and, at the same time, to carry the pressure. In some of these balanced valves the area protected by the plate is somewhat smaller than that resting against the seat in order to have the valve positively seated by a small unbalanced pressure. In some engines, adjustable wedges are provided which may be manipulated from the outside while the engine is in operation.

If we could imagine the D-valve to be rolled up into cylindrical form with the valve stem as the axis, we should have what is called a *piston* valve. The high-pressure and intermediate cylinders of the marine engine of Fig. 115 have such valves. It will be seen that the pressure, being exerted radially in all directions, is perfectly balanced, and the wear is only that due to the packing rings on the valve.

An interesting feature of the piston valves of Fig. 115 is that they are *inside admission*, that is, the steam is admitted from the middle and exhausted at the end of the valve. Inside-admission piston valves are commonly used on locomotive and marine engines.

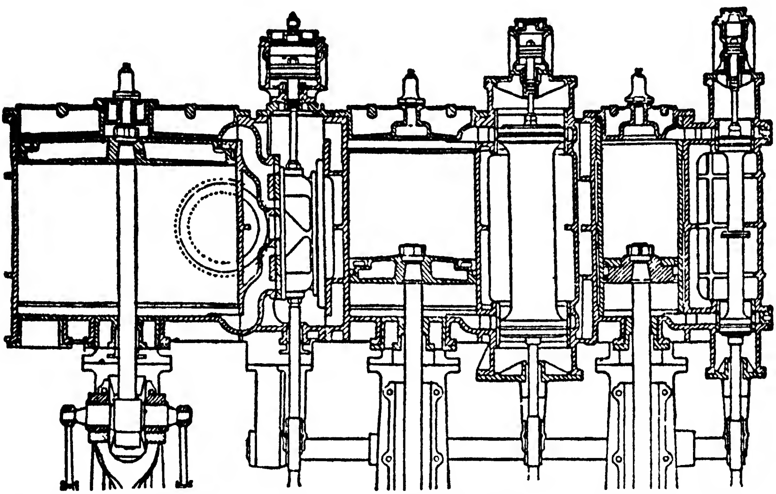


Fig. 115.—Triple-expansion engine built for the Emergency Fleet Corporation, 2,800 hp.

137. Double-ported Valves.—In order to reduce wiredrawing, slide valves are sometimes made with two steam passages which come into use just as the valve is opening or closing. The balanced valve of Fig. 114 is a double-ported valve, the two arrows showing the two steam and exhaust paths. The poppet valves of the Skinner uniflow engine of Figs. 108 and 110 are hollow and double ported. Piston valves are sometimes made hollow and provided with two ports for the same purpose.

Piston valves possess a second advantage in that they allow the steam ports to be made very short, as will be seen in Fig. 115.

138. Corliss-valve Gear.—*Corliss valves*, named after the inventor, are oscillating cylindrical valves placed at right angles to the axis of the cylinder, placed either in the cylinder heads or above and below the cylinder at the ends. Figure 116 shows a typical Corliss engine having two steam valves $1H$ and $2H$ at the top and two exhaust valves $1H1$ and $2H1$ at the bottom, all of which are operated by one eccentric. The advantages of this engine mechanism are (1) the valves may be adjusted independently of each other and, thus, obtain the same steam distribution to both ends of the cylinder; (2) the valves being very close to the

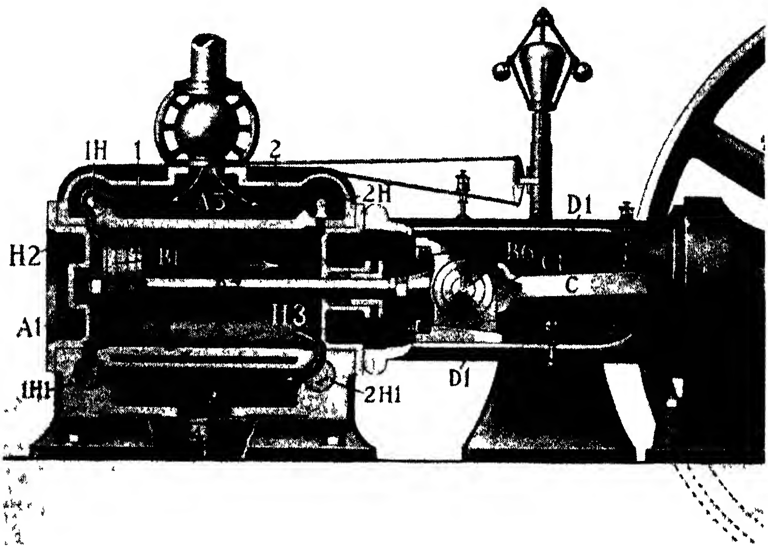


FIG. 116.—Corliss engine with cylinder and valves shown in section. (*Courtesy of Vacuum Oil Co.*)

cylinder, the steam ports are very short and the clearance can be made very small; and (3) the exhaust does not sweep over and cool the steam inlet valves.

Figure 117 shows a diagram of the releasing type of Corliss valve mechanism. The wrist plate is mounted on the side of the cylinder block and is oscillated continuously through the angle *aob* by means of the eccentric on the crankshaft of the engine, the motion being transmitted by the eccentric rod, rocker arm, and reach rod.

Figure 117 shows the mechanism for but one steam valve and one exhaust valve, although all four valves are operated from the same wrist plate. The times of opening and of closing the exhaust valve are fixed in the design of the engine and are not subject to variation during operation. Hence this valve is actuated by the simple exhaust link, connecting a pin on the wrist plate with the exhaust valve arm, as shown in the lower right corner of Fig. 117.

For operating the steam valve a more complicated mechanism shown at the upper right corner of Fig. 117 is required. Mounted on the valve stem *V* are three arms. The outer arm *E* is keyed

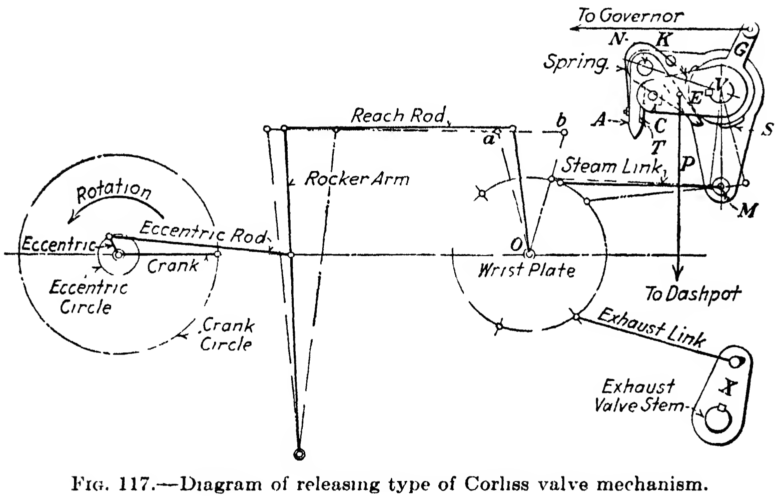


Fig. 117.—Diagram of releasing type of Corliss valve mechanism.

to the stem at its outer end so that the upward motion of *E* opens the valve and the downward motion closes it. A bell crank having the two arms *M* and *N* rides freely upon the valve stem and is oscillated by the stem link connecting *M* with the wrist plate. Intermediate between *E* and the bell crank *MN* is the cam plate *G*, also riding freely on the valve stem. In the same plane with *G* there is the latch *A* carried by the arm *N* of the bell crank and the catch block *C* carried by the valve arm *E*. A spring holds the inner end of the latch *A* against the cam plate *G*.

When the motion of the wrist plate depresses the arm *N*, catch *A* engages the catch block *C*, and, as the arm *N* rises, it carries upward with it the arm E, thus opening the steam valve for admission of steam. As E continues to rise, the inner arm of catch A strikes the knockoff cam K on the cam plate G. This causes A to be disengaged from C and allows the valve to close. Arm E is connected to a dashpot by the rod P. When E is lifted, P is pulled up, and a partial vacuum is created in the dashpot This vacuum closes the valve when the valve is unlatched

The position of the cam plate G is controlled by the governor In this way the position of the knockoff cam K, and therefore

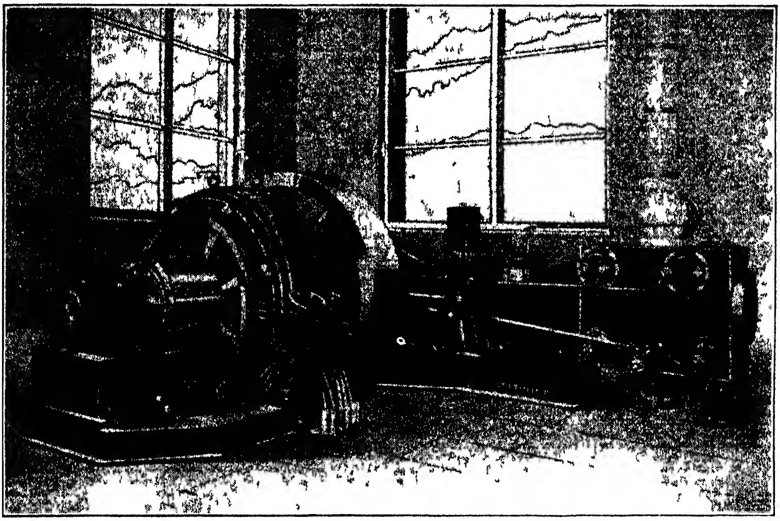


FIG 118.—Erie-Ball nonreleasing Corliss engine

the cut-off, is controlled by the speed of the engine. The other cam S on the cam plate is a *safety* cam. If the governor belt should break, the governor balls will drop and cam S will be rotated clockwise until it strikes the hook A, thus preventing the latch from engaging the catch block and opening the valve

In the position of the mechanism shown in Fig. 117, the piston is at head-end dead center, and the steam valve should be slightly open The maximum opening of the valve will occur when the eccentric is horizontal, and this is also the latest possible moment for cut-off to occur. This means that, with the proportions of the mechanism as shown, the cut-off cannot occur later than about 0 3 of the stroke.

139. Nonreleasing Corliss Gear.—One disadvantage of the Corliss mechanism described above is that the releasing mechanism will not operate satisfactorily at speeds much above 150 rpm. To obtain higher capacities and economies resulting from higher speeds, several manufacturers build engines having Corliss valves which are not detached from the driving mechanism during any part of the stroke.

Figure 118 shows the valve-gear side of a Ball nonreleasing Corliss engine having separate eccentrics for steam and exhaust valves. In the design of these engines the mechanism is arranged so as to cause the valves to open and close sharply and to remain stationary at those portions of the stroke when there is the greatest unbalanced pressure upon them. This movement is achieved by a system of levers in the gear cases on the valve bonnets.

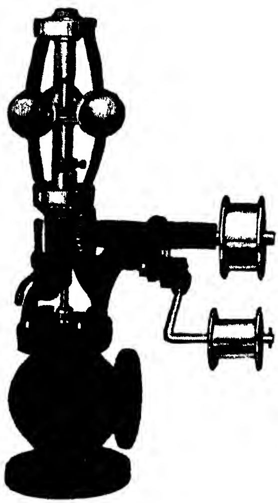


Fig. 119.—Pickering throttling governor.

In addition to having sharp movements, the valves for nonreleasing engines are made double or triple ported.

The compound engine of Fig. 103 is a nonreleasing Corliss engine.

140. Governing Mechanism.—In order to keep the speed of an engine constant, it is necessary to balance the steam supply to the load. There are two general methods of governing steam engines, by *throttling* governors and by *cut-off* governors.

Figure 119 shows a typical throttling governor. The upper pulley at the right is belted to the engine shaft and drives the flyball spindle through bevel gears. As the balls fly out, the stem in the center closes the balanced angle valve at the bottom. The tension on the belt holds up the idle pulley (shown below at the right), so that the governor may act on the valve, but, in case the belt should break or fly off, the idle pulley would drop and close off the steam instantly.

The action of a throttling governor is to lower the steam supply pressure but to leave all the events of the cycle undisturbed.

141. Cut-off Governors.—In connection with the Corliss engine, it was pointed out that the position of the cut-off was adjusted to the load by means of the governor, which shifted the position of the knockoff cam to suit. The governor mechanisms on releasing Corliss engines are of the *flyball type* and are driven by a belt from the engine shaft, see Fig. 116. The knockoff cam lever is provided with a second cam which is a safety cam for automatically stopping the engine in case of failure of the driving belt.

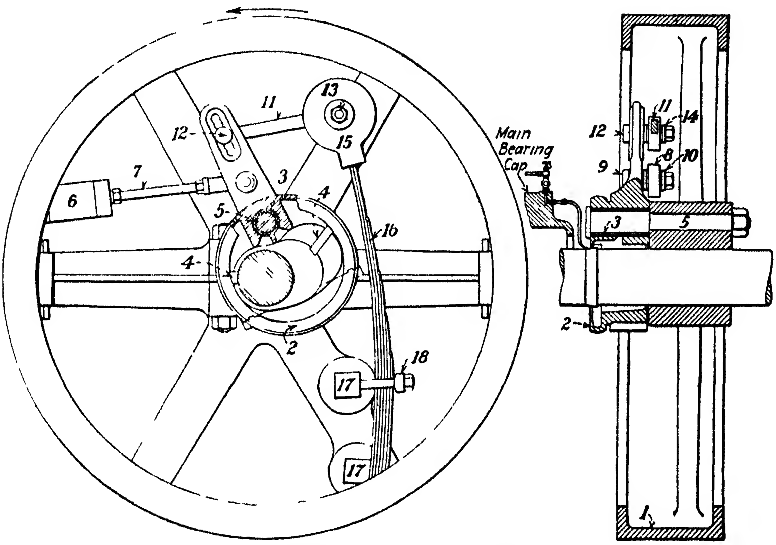


FIG. 120. Erie-Ball centrifugal shaft governor

A second type of cut-off governor is the *shaft governor*. Such governors consist of heavy weights attached to the flywheel in such a manner that increasing speed shifts the position of the weight, and this, in turn, changes either the throw of the eccentric, the angle of advance, or both. There are two general principles acting in all shaft governors, namely, *inertia* and *centrifugal force*, the proportion of the two principles depending upon the design.

Figure 120 illustrates a so-called *centrifugal* type of shaft governor. The weight 15 carried at the end of the lead spring 16 is thrown outward by centrifugal force against the tension of the spring and, in so doing, pulls the eccentric arm by means of the small link 11. The eccentric arm is pivoted about the large pin 5 just above the center of the flywheel and at its lower end carries the eccentric 2. As the weight 15 is thrown out, the eccentric is thrown toward the center of rotation, and the eccentricity and, consequently, the valve travel are reduced, thus cutting off the steam earlier in the stroke.

Figure 121 illustrates what is ordinarily called an *inertia* shaft governor, which differs from the governor of Fig. 120 in having the mass of the governor weight spread out over a long arm which is pivoted about a pin on the flywheel. The weight arm carries the eccentric, and the motion of the arm is restrained by a coil spring. Sudden changes in speed of the flywheel will cause the heavy governor arm to change its position relative to the flywheel because of its inertia and thus change the cut-off. The arm has a considerable unbalanced weight and so has, also, a centrifugal action, as does the governor of Fig. 120.

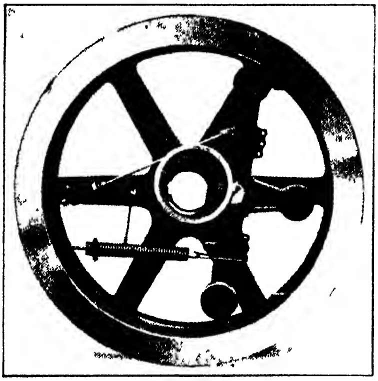


FIG. 121.—Elliott inertia shaft governor.

Nonreleasing Corliss engines are regularly equipped with shaft governors, see Fig. 118.

142. The Flywheel.—The function of the flywheel is to store up energy by speeding up slightly during the admission part of the cycle and to return this surplus energy to the shaft for useful work by slowing down slightly during the expansion or compression parts of the cycle. The amount of rotating mass necessary to hold the cyclic variations in speed down to a given value will depend upon the nature of the indicator cards under the worst operating conditions, that is, when the cut-off occurs very early and the surplus energy must be stored up during a very small part of the cycle.

Curve A of Fig. 122 shows the tangential crank-effort diagram for one working stroke of a double-acting steam engine. This curve is obtained by taking the diagram for a working stroke and deducting from it the back pressures during exhaust and compression in the other end of the cylinder. This diagram of the net forces on the piston is further corrected for the inertia of the reciprocating masses and for friction. From this, the resultant tangential forces acting on the crankpin are determined and laid off on a base line BC representing 180 deg of crankpin travel. If DE is the line of mean torque, representing a uniform load of value BD, the area HKJ will represent energy which must be stored up in the flywheel during the high-effort part of the cycle, while the equivalent areas BDH and JEC will

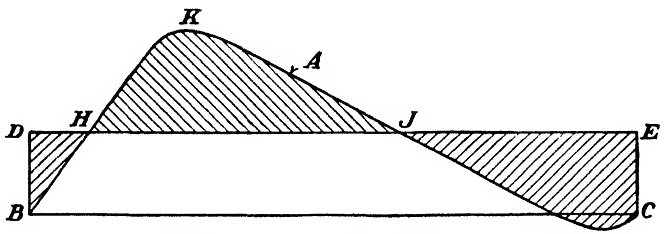


FIG. 122. Tangential crank-effort diagram for steam engine.

represent the energy which must be supplied by the flywheel in maintaining rotation during the low-effort periods.

Now, if V_h = the highest velocity of the crankpin in feet per second

V_l = the lowest, and

V_m = the mean velocity = $(V_h + V_l)/2$, nearly,

we may let $E = (V_h - V_l)/V_m$ = coefficient of unsteadiness E varies from 0.05 for pumping engines to 0.01 for d-c generators and 0.005 for alternators.

Let W equal the work area HKJ. Then the weight (w_c) that must be concentrated in a ring having a radius equal to that of the crank would be

$$w_c = \frac{W}{\dfrac{(V_h^2 - V_l^2)}{2g}}$$

But the velocity of the rim varies with the radius, and, if the radius of the flywheel rim is R times the crank, the weight of the rim would be

$$w_r = \frac{2gW}{R^2(V_h^2 - V_l^2)} = \frac{2Wg}{R^2(V_h + V_l)(V_h - V_l)} = \frac{Wg}{R^2V_m(V_h - V_l)}$$

But $(V_h - V_l) = EV_m$, and then

$$w_r = (Wg/R^2V_m^2E).$$

143. Reversing Mechanisms.—Engines that are used for hoisting, rolling-mill, marine, locomotive, or tractor service must have some means for reversing the direction of rotation. The earliest mechanism and one of the most practical for this purpose is the *Stephenson-link* motion invented by Robert Stephenson. Figure 123 shows a Stephenson link.

Two eccentrics A and B are keyed to the shaft symmetrically with the crank, that is, so that A is ahead of the crank in the direction of the arrow by the angle AOC and B is ahead of the

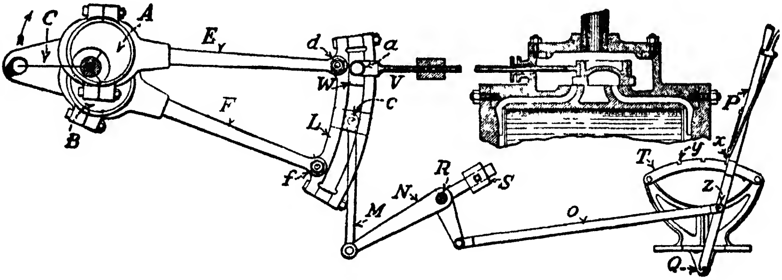
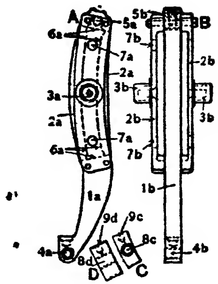


FIG. 123.—Stephenson-link reversing gear.

crank in the opposite direction by the equal angle BOC. The two eccentric rods E and F are pivoted to the two ends of slotted link L and the valve stem is pivoted to a block W that fits into the slot in the link L.

When the mechanism is in the position shown in the figure, the eccentric rod E is in line with the valve stem, and the action is exactly the same as if the eccentric B were not on the engine, that is, the eccentric A drives the valve, and the rotation is in the direction of the arrow.

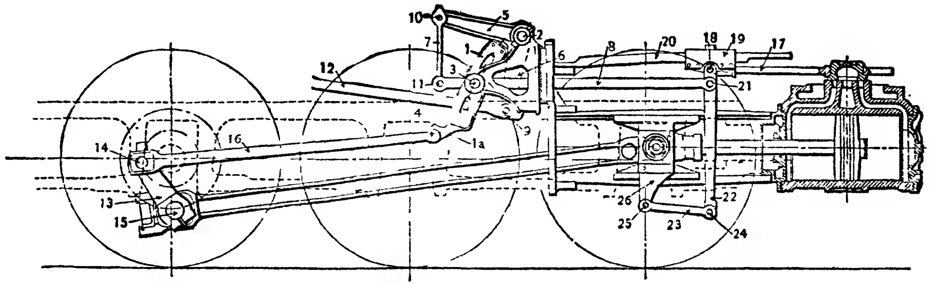
If the reversing lever P is thrown over to notch y in the sector, the shaft R will be turned so as to raise lever N, which, in turn, acts through the connecting bar M and raises the link L so as to place eccentric rod F in line with the valve stem. The eccentric B will control the valve, and, being ahead of the crank in the direction opposite to the arrow, the engine will rotate in that direction.



THE WALSCHAERTS LINK.
(Detail parts.)

Details of Link.

A, Side View of Link.
B, Edge View of Link.
C, Side View of Link Block.
D, Edge View of Link Block.
1a, *1b*, Link Piece.
2a, *2b*, Link Bracket Pieces.
3a, *3b*, Link Fulcrum Pin, or Link Trunnion.
4a, *4b*, Link-Foot Pin Hole.
5a, *5b*, Top Oil Well.
6a, Outlines of Link Slot.
7a, *7b*, Port Holes in Bracket Pieces.
8c, *8d*, Link-Block Pin Hole.
9c, *9d*, Link-Block Oil Well.



Names of Parts.

Side Elevation.

1, Link; 1*a*, Link Foot.
2, Reversing Shaft; 3, Link-Fulcrum Pin, or Trunnion.
4, Link-Foot Pin.
5, Lifting Arm.
6, Reversing Arm.
7, Radius-Rod Hanger.
8, Radius Rod.
9, Reach-Rod Pin.
10, Lifting-Arm Pin.
11, Radius-Rod-Hanger Pin.
12, Reach Rod.
13, Eccentric Crank.
14, Eccentric Pin.
15, Main Crankpin.
16, Eccentric Rod.
17, Valve Stem.
18, Valve-Stem Crosshead Pin.
19, Valve-Stem Crosshead.
20, Valve-Stem Crosshead Guide.
21, Radius-Rod Connecting Pin.
22, Lap-and-Lead Lever, or Combination Lever.
23, Lap-and-Lead Lever Connector, or Crosshead Link.
24, Front Connector Pin.
25, Back Connector Pin.
26, Crosshead Arm.

FIG. 124.—The Walschaerts locomotive valve gear.

If, however, the lever be moved to the middle notch of the sector, the valve stem will be opposite the center of the link *L* and will be moved by both eccentrics, so that the motion of the valve is equal to the steam lap plus the lead, or, in other words, the valve is just barely opened at dead center and immediately closed again. If the valve stem is part way between *d* and *c*, the engine will still move in the direction of the arrow, but with a shorter valve travel.

A serious objection to the use of the Stephenson link on locomotives is that the two eccentrics are beneath the truck in a position to gather grit and to wear rapidly. Another objection is that the gear, being in a horizontal position, is heavy to shift An improved gear invented in 1848 by Egide Walschaerts, master mechanic of the Belgian State Railways, was introduced by several American railroads in 1904 and is now used extensively

The principle of the Walschaerts valve gear is illustrated by the drawing of Fig. 124. Clamped to the main crankpin 15 is an eccentric crank 13 carrying the eccentric pin 14, which is just 90° ahead of the crankpin 15. Eccentric rod 16 connects 14 with the pin 4 on the foot of link 1, which is pivoted at its middle on fulcrum pin 3. The radius rod 8 carries a pin which engages a block in the slot of link 1. The end 11 of the radius rod may be raised or lowered by the bell-crank lever 5, the radius-rod hanger 7, and the reach rod 12, which goes to the cab. Radius rod 8 is connected to the *lap-and-lead* lever 22, the upper end of which is connected to the valve stem, the lower end being linked to the crosshead arm 26.

If the radius rod is at the top of the link 1, the motion of the valve will be opposite to that resulting from having it at the bottom of the link. The lap-and-lead lever moves the valve a distance equal to the lap plus the lead, the valve receiving further motion according to the position of the sliding block in the slotted link 1.

By placing the gear in intermediate positions, the valve travel and, consequently, the cut-off may be varied.

144. The Case for the Steam Engine.—Before proceeding further it might be well to size up the present status of the reciprocating steam engine among prime movers. In the first place the steam engine is a relatively simple machine, well understood and easily kept in order by plant operators. In the second place there are two fields in which the power characteristics of the steam engine are unexcelled.

The first of these is the field of variable-speed and high-torque drives such as is required for rolling mills, shovels, hoists, locomotives, and tractors. It is possible that the steam engine may some day again find its place among automobile engines, a place for which its characteristics are ideally suited. For these purposes the steam engine can deliver any desired torque up to the limit of the machine, at any speed starting from rest. The simple reversibilty of the steam engine is also an advantage here. Its only prime-mover rival in this field is the internal-combustion engine which possesses some advantages that in certain cases outweigh its low torque at low speeds.

The second field is that of by-product power in which the engine acts as a reducing valve for steam for industrial processes, or for building heating. Its prime-mover rival here is the back-pressure and bleeder types of steam turbine, see Art. 160, which are used for electric-power generation, or the small mechanical drive turbines for direct connection to mechanical units. (These are not ordinarily called prime movers although they are rightfully so considered here.)

There is still a very important place in industry for the steam engine but, measured by installed horsepower, this place is relatively small. It has, however, almost completely vanished as a prime mover for public-utility electric power.

Problems

1. A D-slide valve has a travel of 3 in. and an eccentric rod 48 in. long. The engine stroke is 18 in. and the connecting rod 52 in. long. Angle of advance is 32 deg. (*a*) Determine graphically the piston positions and valve positions for crank positions 15 deg apart during a complete revolution (*b*) Plot valve and piston positions against crank positions as abscissas.

2. If the valve of Prob. 1 has a steam lap of 0.72 in. and an exhaust lap of 0.15 in. on both ends, determine (*a*) distance the steam port is open when the piston is at each dead center; (*b*) the crank position when cut-off occurs; (*c*) crank positions for compression.

3. A D-slide valve is 16 in. long and 12 in. wide. (*a*) With a steam pressure of 125 lb gage and an average coefficient of friction of 0.25, what would be the average pull on the valve stem? (*b*) If the valve travel is 6 in. and the engine runs at 250 rpm, what would be the power lost in friction in the valve?

4. At a particular instant, the piston of a 16 by 30-in. engine is moving at a velocity of 1,000 ft per min, and the valve, which is 16 in. long, is open ⅛ in. What is the steam velocity through the valve opening?

5. A Corliss engine has a connecting rod five times as long as its crank, and the angle of advance of the eccentric is 38 deg. (*a*) What is the latest position in the forward stroke (from the head end) that cut-off may be made to occur? (*b*) On the return stroke?

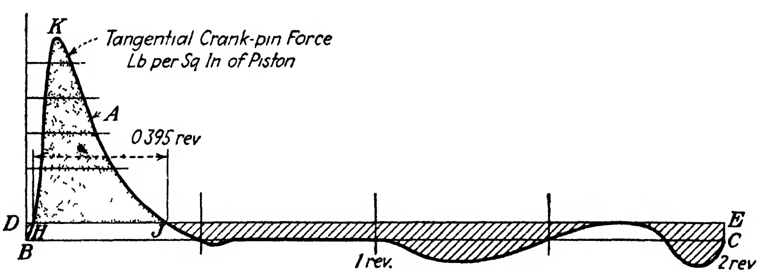
6. A double-beat poppet valve 4 in. in diameter has a lift of ¼ in. What is the approximate area of the port opening?

7. A double-acting engine having a 3-ft stroke runs at 100 rpm. The area *HKJ*, Fig. 122, equals 0.44 sq in., and the diagram was drawn so that 1 in. vertical represents 24,000 lb tangential force at the crankpin and 1 in. horizontal represents 1½-ft crankpin travel. If the ratio of flywheel-rim radius to crank radius equals 4 and the coefficient of unsteadiness is 0.025, calculate the weight of the rim.

8. Figure out (*a*) which would be the direction of rotation of the engine of Fig. 124 with the radius rod at the top of the link; (*b*) the direction of rotation of the engine of Fig. 123 with the gear in the position shown in the figure.

9. An 8 by 11½-in., single-acting, four-stroke, single-cylinder Reeves gas engine has two flywheels which may be considered as a single ring weighing 1,883 lb with all its weight concentrated at a radius of 1.696 ft Normal speed is 300 rpm.

The diagram below was developed by the method described on page 215 from the gas engine indicator diagram reproduced on page 122. The dotted area just equals the cross-hatched area, the rectangular area *BDEC* being the average constant flow of energy. Obviously energy must be stored in the flywheels during the part of a revolution *HJ* in an amount represented by dotted area *HKJ*. Ordinates are pounds per square inch of



piston area acting tangentially on the crankpin. Mean ordinate above *HJ* which is 0.395 revolution is 22.5 lb per sq in. Ordinate *DB* is 4.95 lb per sq in. Calculate (*a*) the indicated horsepower output; (*b*) the coefficient of unsteadiness of the engine.

CHAPTER XV

THE STEAM TURBINE

145. Flow of Fluids through Orifices.—Consider a standpipe S, Fig. 125, having an orifice O at some distance h_2 above its base (considered as a datum of reference). The water level is kept at

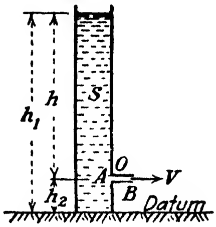


FIG. 125.

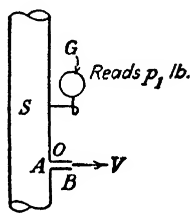


FIG. 126.

a constant height h_1. Then the theoretical spouting velocity of the jet from the orifice will be

$$V = \sqrt{2gh} = \sqrt{2g \times (h_1 - h_2)} \tag{101}$$

from which

$$(h_1 - h_2) = \frac{V^2}{2g} \tag{102}$$

The energy possessed by 1 lb water at A due to the head above it equals h or $(h_1 - h_2)$ ft-lb, and that due to the elevation above the datum equals h_2, or

$$E_A = (h_1 - h_2) + h_2 = h_1 \tag{103}$$

(The energy of every pound of water in the standpipe is h_1, regardless of its location.)

Also the energy, due to its velocity, possessed by the same pound on arriving at B equals

$$\frac{mV^2}{2} = \frac{V^2}{2g}$$

to which must be added the energy h_2, due to its position above the datum. That is, the total energy at B is

$$E_B = \frac{V^2}{2g} + h_2 \tag{104}$$

Neglecting friction, $E_A = E_B$ and, combining Eqs. (103) and (104),

$$h_1 = \frac{V^2}{2g} + h_2$$

$$(h_1 - h_2) = \frac{V^2}{2g} \tag{105}$$

as in Eq. (102); that is, the *change in potential energy* of the pound of water in passing from A to B equals the kinetic energy at B, and is responsible for the velocity acquired by the jet. Then, solving Eq. (105) for V, we have

$$V = \sqrt{2g(h_1 - h_2)} = 8.02\sqrt{h_1 - h_2}$$

Following this illustration as an analogy, consider a steam pipe S, Fig. 126, having an orifice at O and carrying steam at a constant pressure p_1 and enthalpy of h_1 Btu per lb above water at 32°. The energy of each pound of steam in the pipe is h_1, regardless of its location.

The steam blows through the orifice and leaves with a velocity V.

At B, just outside the orifice, 1 lb of the moving steam possesses the kinetic energy

$$\frac{mV^2}{2} = \frac{V^2}{2g} \text{ ft-lb} \tag{106}$$

plus the potential heat energy h_2 due to its temperature, pressure, and quality after passing the orifice; that is, the total energy in foot-pounds at B equals

$$E_B = \frac{V^2}{2g} + Jh_2 \tag{107}$$

If we assume that the passage through the orifice is adiabatic, we have

$$E_A = E_B$$

or

$$Jh_1 = \frac{V^2}{2g} + Jh_2$$

and

$$J(h_1 - h_2) = \frac{V^2}{2g} \tag{108}$$

that is, the *change in potential heat energy* of the pound of steam in passing through the orifice equals the kinetic energy at B and is responsible for the velocity acquired by the jet. Then, solving for V, we have

$$V = \sqrt{2gJ(h_1 - h_2)} = 8.02\sqrt{J(h_1 - h_2)} \tag{109}$$

We know that the flow through the orifice is theoretically adiabatic, and, if the initial condition of the steam at p_1 and the pressure p_2 are known, we may obtain from the steam table the enthalpy h_1 and compute the enthalpy h_2 after the constant entropy drop to p_2, as was done for the Rankine cycle, Art. 112.

Example.—Suppose the initial pressure p_1 be 350 per sq in. abs, the superheat be 300°, and the final pressure p_2 be 1 lb per sq in. abs. What will be the theoretical jet velocity after an adiabatic passage through the orifice?

SOLUTION.—The steam temperature is 431.7 + 300 = 731.7°F.

From Table IX, Appendix B,

$$h_1 = 1383 \text{ Btu}$$

and

$$s_1 = 1.6708$$

Also at 1 lb abs, from Table VIII, Appendix B,

$$s_{f2} = 0.1326$$

and

$$s_{fg2} = 1.8442$$

Now for an adiabatic (constant-entropy) drop

$$s_1 = s_{f2} + x_2 s_{fg2}$$

or

$$1.6708 = 0.1326 + x_2(1.8442)$$

$$x_2 = \frac{1.6708 - 0.1326}{1.8442} = \frac{1.5382}{1.8442} = 0.834$$

Then

$$h_2 = (h_{f2} + x_2 h_{fg2}) = 69.7 + 0.834(1{,}035.3) = 933 \text{ Btu}$$

and, from Eq. (109),

$$V = 8.02\sqrt{778(1{,}383 - 933)}$$
$$= 8.02\sqrt{349{,}500} = 4{,}740 \text{ ft per sec}$$

146. The Impulse Principle.—If the steam jet of Art. 145 is allowed to impinge upon the blades of a wheel, it will cause the wheel to turn. If these blades are shaped as in Fig. 127, the direction of the jet will be completely reversed. Then, if the blade has a velocity away from the jet equal to half the velocity of the jet, the steam will leave the blade with 0 absolute velocity. The steam velocity *relative* to the blade will be 2,370 ft per sec both entering and leaving it, but the *absolute* velocity of the steam on leaving the blade will be zero. This means that

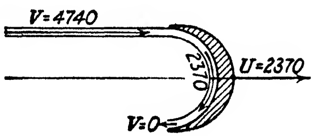


FIG. 127.

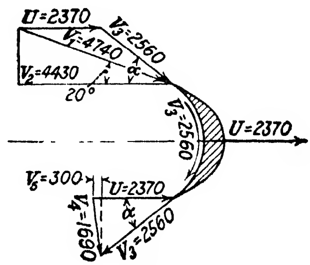


FIG. 128.

the jet will have given all its kinetic energy to the blade. Theoretically, therefore, the blade should move at half the jet velocity for maximum absorption of energy from the jet.

In practice it is not feasible for the jet to lie in the plane of rotation of the blades, but it must be placed beside the wheel at some small angle (about 20 deg) as shown in Fig. 128. The calculation of the energy absorbed by the wheel then becomes more complicated.

The velocity-vector diagram of Fig. 128 shows the relationships of the various velocities. The jet angle is assumed to be 20 deg and the blade is symmetrical about its center plane. Let V_1 represent the velocity of the jet and U the velocity of the blade, arbitrarily taken as half of the velocity of the jet. If we take graphically the vector difference of these two velocities, we shall obtain V_3. This component V_3 has a magnitude equal to 2,560 relative to the blade and is at an angle α about 38.5 deg with U; that is, the angle of the blade tip must be 38.5 deg with the center plane in order that the steam may enter the blade without shock.

The steam moves along the surface of the blade at a velocity of 2,560 ft per sec relative to the blade surface and, when it arrives at the leaving edge, leaves the blade at a velocity of 2,560 ft per sec. But the blade is receding at the velocity of 2,370, and, if we combine vectorially the 2,560 jet velocity and the 2,370 blade velocity, we obtain V_4, equal to 1,690 ft per sec, as the *absolute* velocity of the steam at exit and in the direction shown in the diagram.

From the diagram it will be noted that V_2, the component of V_1 in the direction of rotation, is 4,430 ft per sec and that, when the jet leaves the wheel, it still has a component V_5 of 300 ft per sec in the direction of rotation This means that the *change* in absolute velocity in the direction of rotation is $(4{,}430 - 300) =$ 4,130 ft per sec From mechanics we know that force = mass $\times$ acceleration, or, multiplying both terms by time t, we know that

$$F \times t = m \times at = m\Delta V$$

or

$$\text{impulse} = \text{change of momentum}$$

Whence

$$F = \frac{m}{t}\Delta V$$

Or, in words, a force F acting for a time t will produce in a mass m a velocity change ΔV in the direction of F. Conversely, a mass m per second, undergoing a velocity change ΔV will exert a force F, where ΔV is the vector change of velocity in the direction of F.

The change of velocity of the steam in the direction of rotation of the blade is 4,130 ft per sec. Then, if 1 lb steam passes through the blading in each second with this reduction of its velocity in the direction of rotation, we shall have the relation

$$F = \frac{1}{g} \times 4{,}130 = \frac{4{,}130}{32.16} = 128.5 \text{ lb}$$

The work done per second by each pound of steam is

$$FU = 128.5 \times 2{,}370 = 305{,}000 \text{ ft-lb}$$

and the horsepower for each pound of steam per second is 305,000/550 = 554.0 hp.

The kinetic energy of the jet per pound of steam we found to be

$$J(h_1 - h_2) = \frac{V^2}{2g} = 349{,}500 \text{ ft-lb}$$

The theoretical efficiency of conversion of the 349,500 ft-lb of available work is 305,000/349,500 = 87.3 per cent.

Another way to arrive at the same net work is as follows: The residual velocity in the steam leaving the blade was 1,690 ft per sec, which represents a loss, due to kinetic energy left in the steam, of

$$\frac{V^2}{2g} = \frac{1{,}690^2}{64.32} = 44{,}500 \text{ ft-lb}$$

Then the useful work is 349,500 − 44,500 = 305,000 ft-lb, as before.

This example indicates that one way to improve the efficiency of the machine would be to reduce the residual velocity of the steam. The methods of doing this are described in later articles.

147. Flow of Steam through Nozzles.— Although the velocity of steam issuing from an orifice seems very great (nearly a mile a second) for the example of Art 146, the passage through the orifice is not instantaneous and the drop in pressure requires a measurable time. This means that if we could measure the pressure of the steam along the jet as it issues, we should find that it falls progressively to the lowest pressure p_2 and that the velocity increases progressively to the highest V. In the case of the water jet the material is not expansible and the area of the stream is inversely proportional to the velocity at each point and makes a tapering stream. It has been found that, if the inner edges of the orifice are rounded so as to approximate this reduction in area of the stream, the flow will be facilitated.

In the case of steam, however, the enthalpy, quality and specific volume change progressively, as well as do the pressure and velocity. This means that the cross-sectional area of the jet must change, and, if we were to inclose the jet in a tube or nozzle, the shape of the nozzle must follow some law. What this law is can best be pointed out by means of an example.

TABLE XII.—CHANGE IN PROPERTIES OF 1 LB OF STEAM DURING AN ADIABATIC EXPANSION
From 350 Lb Abs Pressure and 300° Superheat

1. Pressure, lb per sq in. abs	2. Enthalpy, Btu	3. Enthalpy decrease, Btu	4. Velocity, ft per sec	5. Superheat, °F.	6. Quality, x	7. Specific volume, cu ft	8. Ratio of 7 : 4, sq ft	9. Diam. of circular nozzle, in
350	1,383	0	0	300	.	1 95	Infinite	Infinite
300	1,364	19	1,000	274		2 21	0.00221	0 636
260	1,347	36	1,360	251		2.47	0.00182	0.577
240	1,337	46	1,515	238	.	2.63	0.00173	0.563
220	1,327	56	1,675	222		2.81	0.00168	0 555
200	1,316	67	1,830	207		3 03	0.00166	0.551
180	1,304	79	1,990	191	. .	3.28	0.00165	0.550
160	1,292	91	2,135	171	. .	3.59	0.00168	0.556
140	1,277	106	2,300	151	. . .	3.98	0.00173	0.564
120	1,262	121	2,470	130	. . .	4.48	0.00182	0.578
100	1,245	138	2,640	105	. . . .	5.15	0.00195	0 598
80	1,224	159	2,830	75	. .	6.11	0.00216	0 630
60	1,199	184	3,035	39	. .	7.66	0.00252	0.680
40	1,165	218	3,295		0.996	10,45	0 00317	0 763
25	1,130	253	3,560		0.968	15.80	0.00444	0.902
14	1,089	294	3,830	.	0.938	26.3	0.00687	1.120
6	1,035	348	4,170		0.900	55 9	0.01340	1.569
1	934	450	4,740	.	0 834	278.7	0 0588	3 282

Suppose the steam of the example of Art. 146 is considered as expanding adiabatically in steps from 350 lb abs and 300° superheat. Let the enthalpy change, velocity, quality or superheat, specific volume at each acquired state, and the ratio of the specific volume to velocity for the successive steps be calculated and tabulated as in Table XII. This calculation is greatly facilitated by the use of the Mollier chart or Ellenwood's charts.

The velocities (column 4), specific volumes (column 7), and the ratios of columns 7:4 (column 8) have been plotted against pressure in Fig. 129. It should be noted that the ratio of specific volume to velocity gives the cross-sectional area of the nozzle at each pressure for a flow of 1 lb per sec and is measured in square feet. This shows that the nozzle should at first converge and then expand and that the pressure at which the area is a minimum is at about 180 lb, or 51½ per cent of the initial pressure for this particular initial pressure and superheat. For saturated steam the "throat" pressure is 57.7 per cent and becomes somewhat smaller for superheated steam, approaching the value 50 per cent for high temperatures. The throat occurs at that point at which the velocity of the steam ceases to increase more rapidly than its volume.

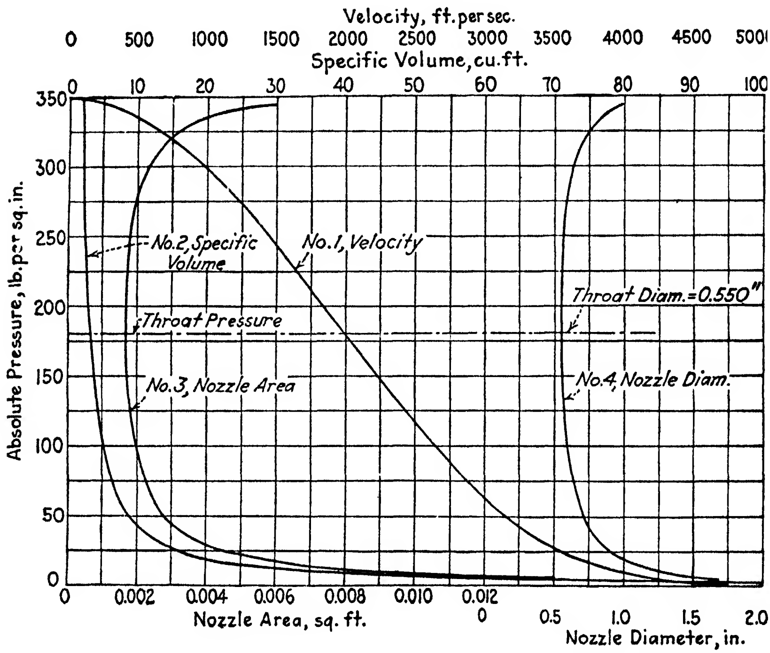


FIG. 129.—Steam velocity, specific volume, and nozzle area and diameter plotted against pressures (Table XII).

148. Steam Nozzles.—Figure 130 shows a typical turbine nozzle which converges to a "throat" and then diverges to the "mouth" or exit. It will be observed that the converging portion is made very short and the diverging portion is simply a long conical taper. The pressure and velocity curves were drawn for the conditions of the curves of Fig. 129. The pressure drop is very rapid in the first inch of length of this nozzle, and the velocity increase is correspondingly rapid at the same time.

In practical turbines such a large enthalpy change as that of Fig. 130 is not ordinarily used in a single nozzle, for reasons that will be explained later, but several "stages" are placed in series using as many nozzles. In this case the enthalpy change and pressure drop in each nozzle are smaller, and, if the discharge pressure is greater than the "throat" pressure, the divergent portion is omitted entirely. This will be understood from the curves of Fig. 130, which show that it would be useless to add the

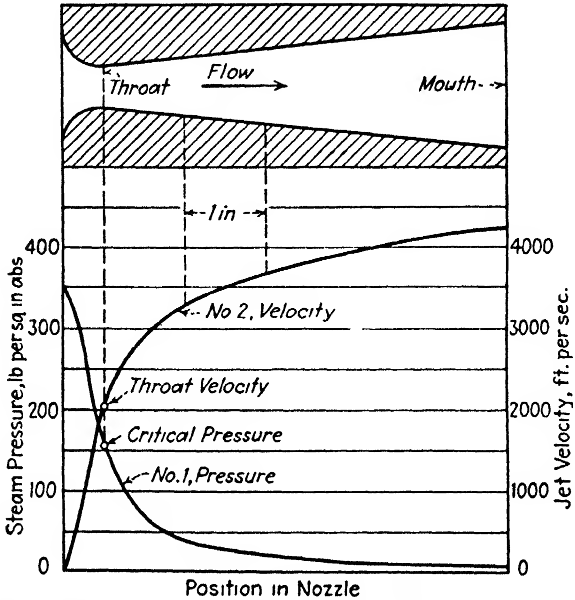


FIG. 130.—Pressures and velocities in a steam nozzle (neglecting friction)

expanding portion unless the pressure drop, in this case, were to a pressure less than 200 lb. The diverging nozzle for steam was

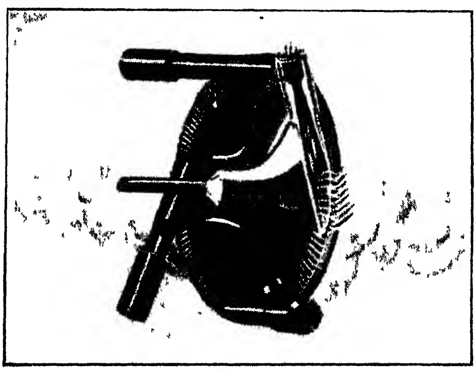


FIG. 131.—Principle of the De Laval single-stage turbine.

patented in the United States, in 1894, by Gustaf de Laval, a Swedish engineer.

149. The Single-stage Impulse Turbine.—The simplest type of turbine and the earliest practical one is that developed by De Laval in the early eighties. This turbine, shown in Fig. 131, uses a single disk carrying a single row of blades or "buckets" similar in shape to that shown in Fig. 128. This is called a "single-stage" turbine. The machine may have two or more nozzles arranged at different points around the disk for additional power.

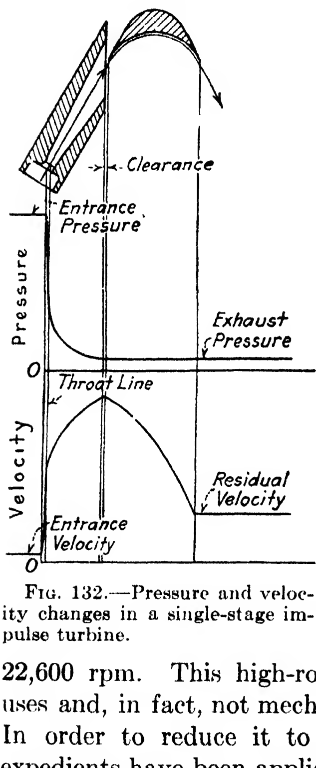


FIG. 132.—Pressure and velocity changes in a single-stage impulse turbine.

The pressure and velocity relations of the steam as it passes through the turbine are shown in Fig. 132. In this simple turbine the entire pressure drop is in the nozzle; the absolute velocity of the steam rises to a maximum in the nozzle and then drops to its exit velocity as it passes through the blading in the wheel, following the curve somewhat like that shown in the figure.

150. The Velocity-stage Impulse Turbine.—It was shown, in Art. 146, that the velocity of the bucket should be approximately half the jet velocity of the steam. This means, for example, that a disk with a bucket radius of 1 ft and a bucket speed of 2,370 ft per sec would have to revolve at $2,370 \times 60 \div 2\pi =$ 22,600 rpm. This high-rotative speed is undesirable for most uses and, in fact, not mechanically possible for most power uses. In order to reduce it to a more practicable figure various expedients have been applied.

One method of reducing the rotative speed (rpm) is to increase the diameter of the bucket disk, but, since the centrifugal stress increases with the diameter, a practical limit for the diameter is soon reached. Another method is to use reducing gears, as shown in Fig. 136. In some small machines the disk may rotate 30,000 to 40,000 rpm while the power shaft rotates at 3,000 rpm or less.

A method applied by De Laval in 1889, for using the high velocities resulting from the very high steam pressures he was using at the time (1,400 to 2,800 lb per sq in.), is suggested by the velocity diagram of Fig. 128. Since the steam leaves the

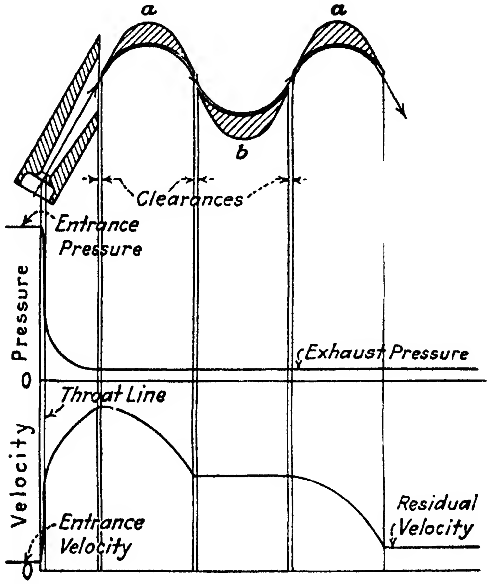


FIG. 133.—Pressure and velocity changes in a velocity-stage impulse turbine.

rotating buckets with a rather high velocity, the energy it possesses could be made available if it were turned around and redirected against the buckets. This is done, as is shown in Fig. 133, by means of a row of stationary blades *b* between two rows of moving blades *aa* on the wheel. In this case the entire pressure drop occurs in the nozzle as before. The residual velocity in the steam after passing through the first moving blades suffers no change as it passes over the stationary blades (neglecting friction), but the direction is changed. A second velocity drop occurs in the second row of rotating blades. This method of reducing the velocity by stages is called ***velocity staging***, and, although originated by

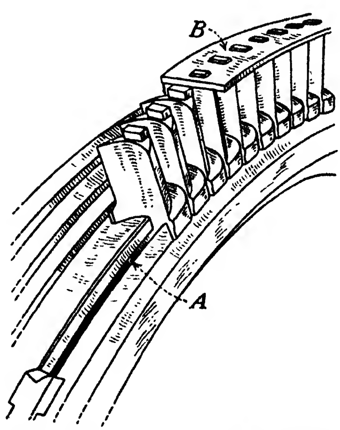


FIG. 134.—Curtis impulse blading details (*Power*).

De Laval in 1889, it was patented in the United States by C. J. Curtis in 1896.

The method of securing the blades to the disks in the Curtis turbine is shown in Fig. 134, in which *A* is a T-slot around the periphery and *B* is a shroud ring to which each blade is riveted.

A single-runner, velocity-stage turbine (built by the De Laval Steam Turbine Company) is shown in Fig. 135. In this machine, the steam enters the balanced governor valve at the top and then

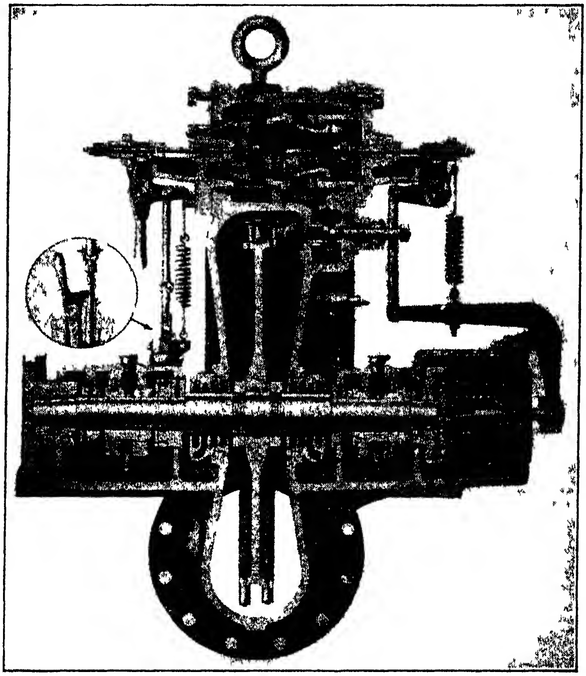


FIG 135.—Axial section of a De Laval velocity-stage turbine.

enters the nozzle through a small hand-controlled valve. It can be seen that the stationary blades extend only a small distance around the disk near the nozzle. The centrifugal governor is shown at the right end of the shaft and controls the throttle valve through two ball-bearing, bell-crank levers. The bell crank at the left of the casing is connected to an emergency stop valve and is operated by a small centrifugal weight on the disk shaft. To prevent the exhaust steam from escaping along the shaft to the atmosphere, carbon packing rings are placed in the casing around the shaft. In the figure, three of these rings are shown on each side with a *leak-off* between the last two rings.

A second form of velocity staging is the "reentry" type, built by the Westinghouse Electric and Manufacturing Company,

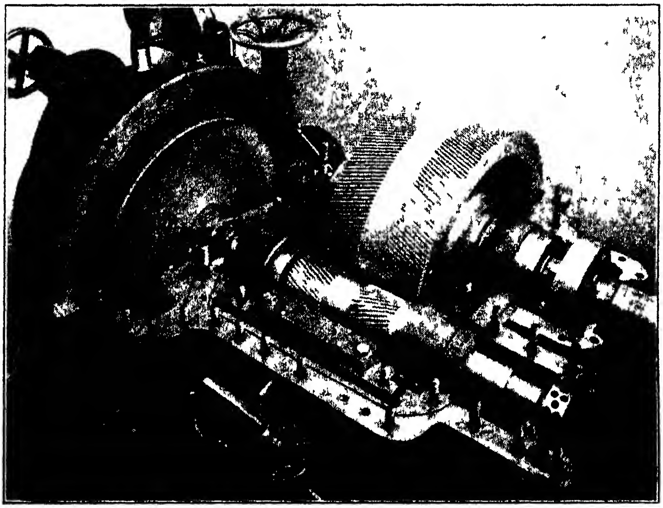


Fig 136 —Westinghouse impulse turbine with reduction gearing

shown in Fig. 136. In this machine the nozzle and disk are the same as for a single-stage turbine, but the steam, after leaving the moving blades, is caught by curved guide passages and redirected through the blading in the reverse direction. A developed section through the nozzle block and reversing chamber is shown in Fig. 137. In the machine shown in the figure, there is but one reentry, and the exhaust is on the same side of the wheel as the nozzle. There may be two or more reentry passages.

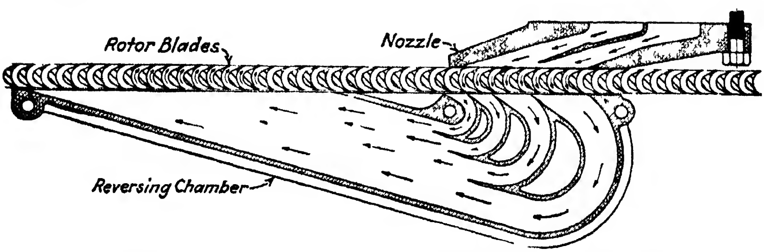


Fig 137.—Nozzle block and reversing chamber showing path of steam in Westinghouse velocity-stage, impulse turbine.

A third type of turbine is the Terry steam turbine, shown in Fig. 138, which uses a single pressure drop in a single set of nozzles but has more than one passage through the blading. In this type the steam is conducted in a helical path tangent to the periphery of the wheel and the action is very similar to a steam worm acting on a steel worm wheel. Figure 139 illustrates the action of the steam in the Terry wheel. Similar turbines are built by the B. F. Sturtevant Company and the Lee Steam Turbine Company.

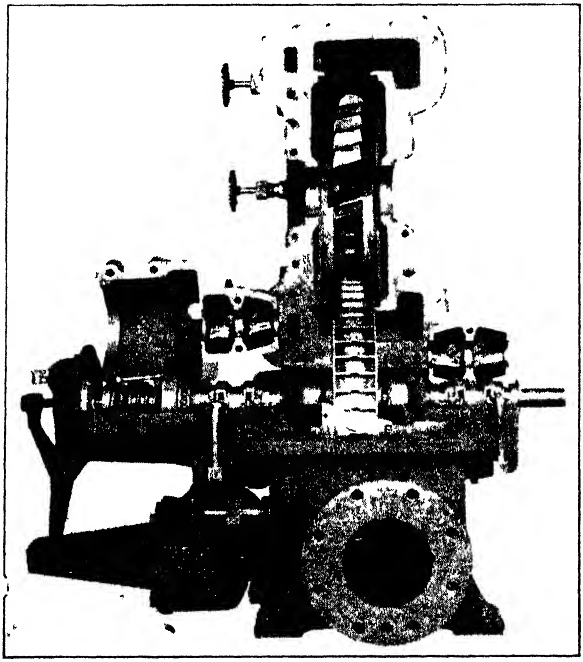


FIG. 138.—Terry steam turbine with upper half of casing lifted.

The simple turbines thus far described are made in the smaller sizes only and are used for mechanical drives, pumps, fans, and small generators.

151. The Pressure-stage Impulse Turbine.—A second method for reducing the jet velocity is to break the *pressure drop* into two or more stages. A set of nozzles expands the steam down to some intermediate pressure and directs the jet on a single impulse wheel. After leaving this wheel the steam is expanded in a second set of nozzles and directed on a second impulse wheel, and so on through the turbine, each set of nozzles producing a pressure drop called a *pressure stage.*

Turbines having this arrangement are called *multistage* and may have from 3 to 20 stages. This type is commonly called the *Rateau type* and sometimes *multicellular.* Such a turbine, built by the Elliott Company, is shown in the sectional view of Fig. 140. It will be observed that the turbine consists of several single-

Fig. 139.—Action of steam in Terry nozzle, wheel and reversing chambers.

stage impulse wheels in series, each having its own set of nozzles in the *diaphragm* preceding it. The diaphragms have packing rings at the shaft to prevent leakage of steam from one stage to the next or lower pressure. Packing rugs will be observed at either end of the casing.

The pressure and velocity relations of the Rateau type of blading are shown in Fig. 141.

152. The Pressure-and-velocity-stage Impulse Turbine.—If the pressure drop be made in two or more nozzle or pressure

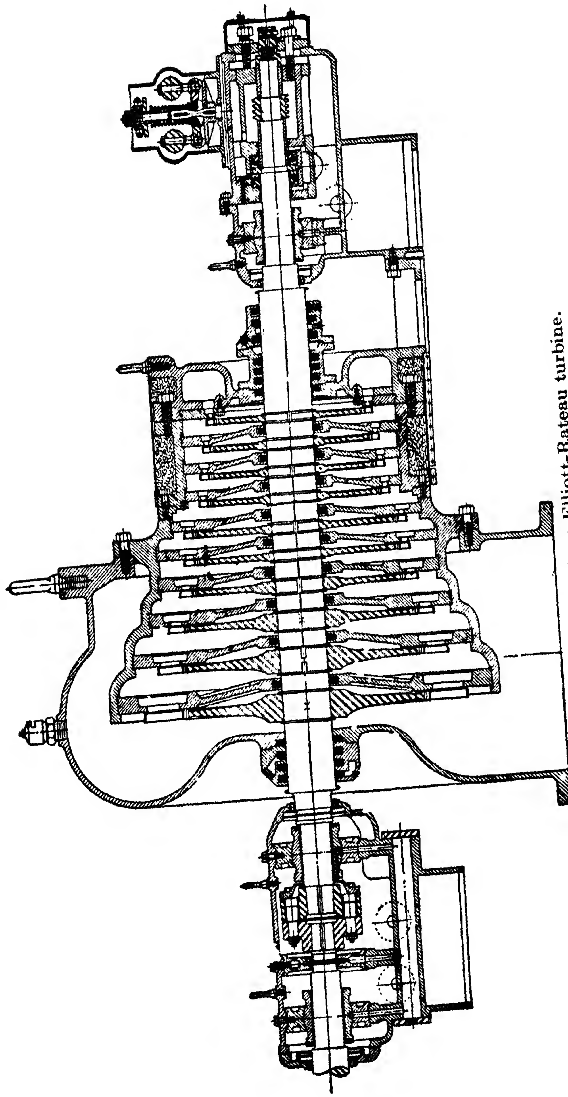


FIG. 140.—Ten-stage Elliott-Rateau turbine.

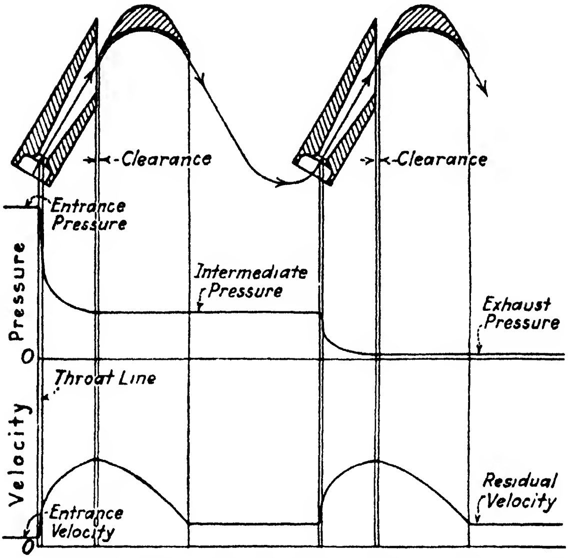


FIG. 141 Pressure and velocity changes in a pressure-stage impulse turbine

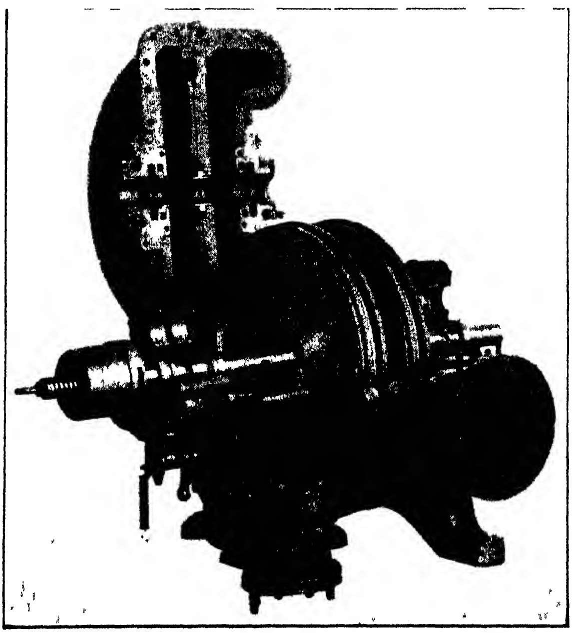


FIG. 142.—Two-stage Curtis turbine with top half of casing lifted.

stages and each pressure stage be broken up into velocity stages, we have the two methods just outlined for reducing the jet velocity combined in one machine. Such an arrangement is used in the Curtis turbine, built by the General Electric Company and

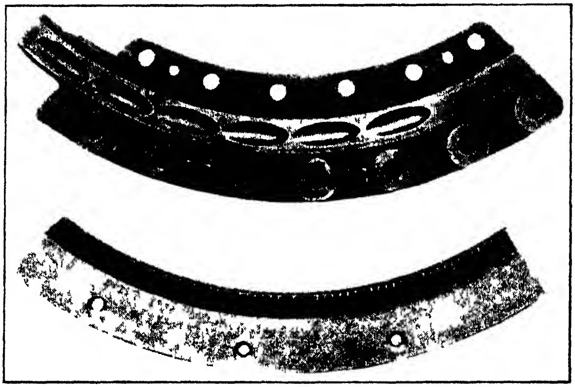


Fig. 143 —Outlet side of nozzle plate (above) and corresponding intermediate stationary blading

shown in Fig 142 This is a two-stage turbine, but several more such stages might be used

The part in the foreground is the nozzle block which has been removed from its normal position on the casing just back of it

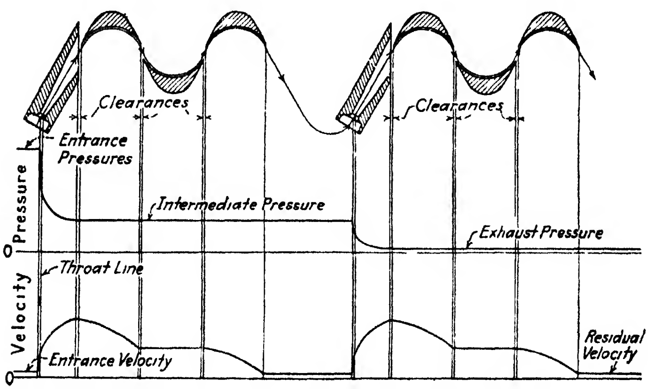


Fig 144 —Pressure and velocity changes in a two-stage Curtis impulse turbine.

The outlet side of the nozzle block and the intermediate or stationary buckets are shown in Fig. 143. The governor spring will be seen protruding from the housing at the left end of the shaft in Fig 142

The pressure and velocity relations of a two-stage Curtis turbine are illustrated in Fig. 144.

153. "Composite"-type Impulse Turbine.—The two types of impulse turbine may be combined in a single machine, as in Fig. 145. In this industrial-type turbine the first stage (the first two disks on right end) is of the Curtis type (velocity stage), and the remaining six are of the Rateau type (pressure stage). At each pressure stage the steam is retained by diaphragms which have labyrinth packings at the shaft. Carbon packing rings are used

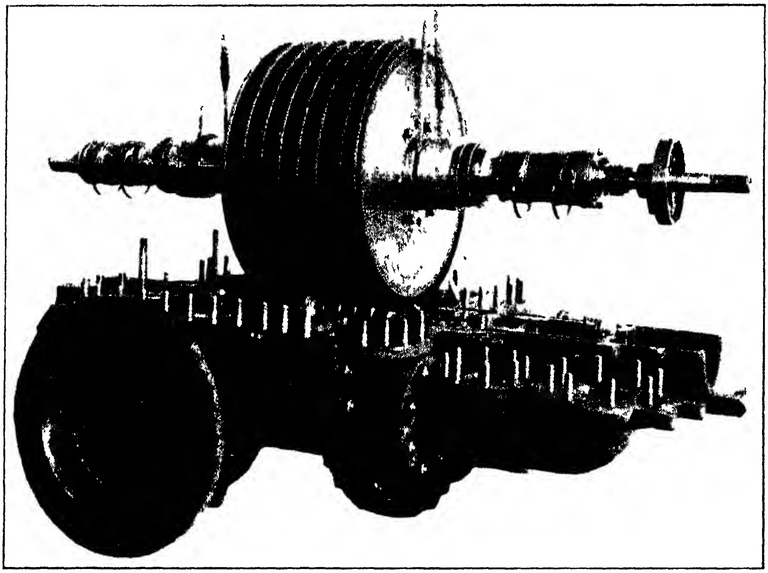


Fig. 145.—Terry turbine of the Curtis-Rateau combination impulse type.

at the ends of the casing. This turbine has a spring-loaded centrifugal governor mounted directly on the shaft on the right-hand end. The governor motion is transmitted through a linkage to a double-seated balanced governor valve, thereby regulating the steam and controlling the speed.

Figure 146 shows a sectional view of a large Curtis-Rateau turbine for electric power service.

The advantage of the Curtis-Rateau combination is that a single velocity stage will efficiently cover an expansion range equal to several pressure stages. The velocity stage is most efficient when operating in the high-pressure end of a turbine, and the pressure staging is most efficient at the low-pressure end.

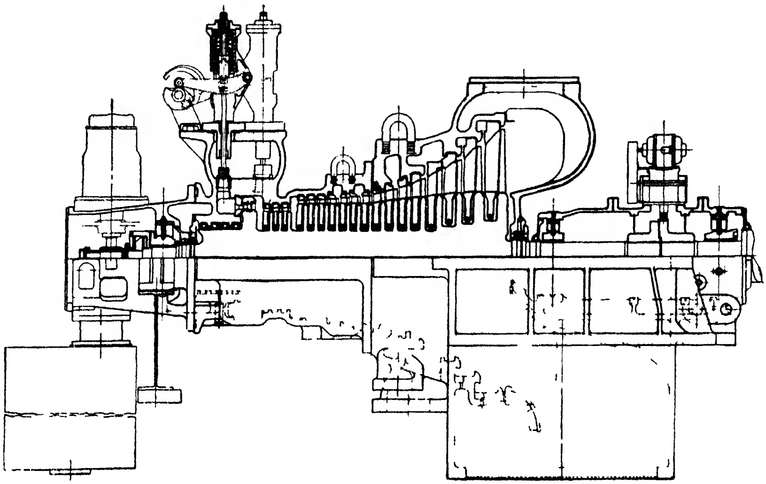


FIG. 146.—General Electric 15,000-kw 3,600-rpm turbine.

154. Nozzle and Blade Friction.—The velocity curves of Figs. 129 and 130 are theoretical and do not take account of friction in the nozzle. The effect of friction is to tend to heat the steam and, in so doing, to return some of the velocity energy back to heat energy. This means that, since a smaller quantity of heat is actually converted into kinetic energy of the jet, the spouting velocity will be less than the theoretical and the steam will be drier than theoretically determined. Nozzle friction may cause a reduction of 10 per cent in the theoretical velocity.

Figure 147 shows the actual enthalpy change through a ten-stage Curtis[1] turbine. If the process had been adiabatic, the path would have been the dotted line *AL* from the initial state to the condenser pressure of 0.5 lb. On account of friction some of the energy is returned to the steam; at the end of the first stage the state is *B* instead of *M*, and at the end of the tenth stage the state is *K* instead of *L*.

155. The "Reaction" Principle.—The force which produces the rotation of a Barker's mill type of water sprinkler is called "reaction." The nozzles on the arms receive a backward thrust due to the acceleration of the water in the jets.

[1] *Gen. Elec. Rev.*, March, 1918.

If the nozzles of the turbines described in the foregoing articles were arranged so as to rotate about the shaft axis, the reaction of the issuing jets would produce a steam Barker's mill. Such a machine would be very inefficient because of the high residual velocity of the steam. If, however, the pressure drop could be broken up into a large number of stages, say 50 or 60, the jet velocity at each of the nozzles would be relatively small and

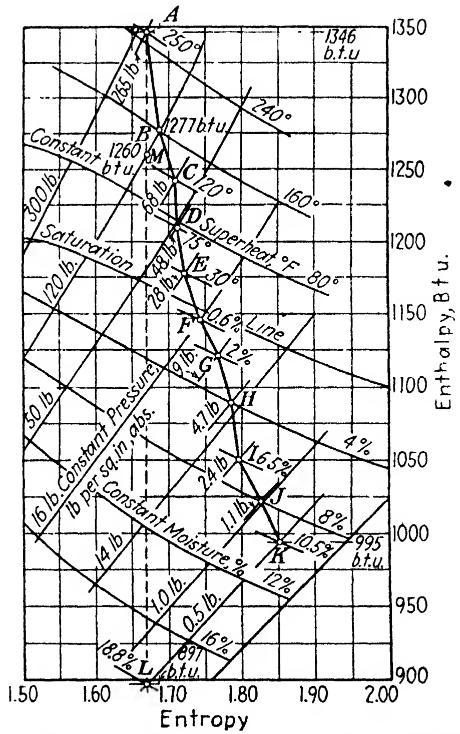


FIG. 147.—Enthalpy change through an actual Curtis turbine.

the residual velocity also small. The residual velocity, moreover, could be used in the next set of rotating nozzle buckets.

In making a practical turbine on this principle, a combination of impulse and reaction is used. Figure 148 shows the blading of a so-called "reaction" turbine, although "impulse reaction" would, perhaps, be a more accurate name, together with the pressure and velocity relations. A complete spindle is shown in Fig. 149. All the blades, both stationary and moving, have the same general shape, although the steam area increases toward the exhaust end of the turbine in order to provide for the greater steam volume at the lower pressures. This is accomplished by

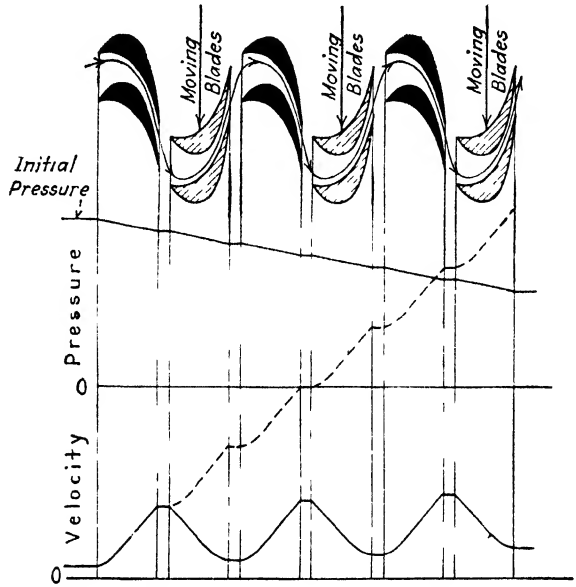


FIG. 148.—Pressure and velocity relations in reaction blading.

increasing either the lengths of the blades or the spindle diameter or both, as shown in Fig. 149. It will be noticed in Fig. 148 that

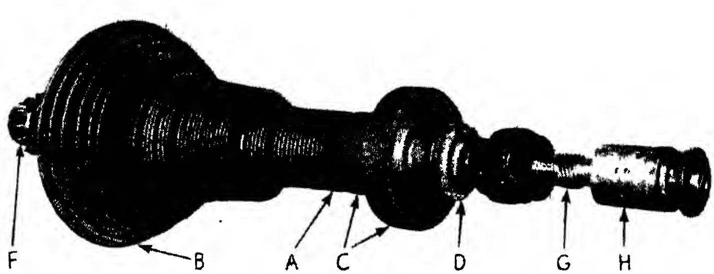


FIG. 149.—Spindle of an Allis-Chalmers Parsons (reaction) turbine.

the steam area in any one row of blades, whether stationary or moving, decreases from the entrance to the exit side. This is because each row of blades acts as a nozzle, producing an increase of velocity with a moderate drop in pressure. Since the drop in pressure in any one row is comparatively small, the steam passages of any row are practically converging nozzles, see Art. 148, and the steam area of that row decreases from entrance to exit, as shown in Fig. 152. Each row, however, has a larger area than the one preceding it, because of the increased volume of the steam.

If one refers to the pressure and velocity relations shown in Fig. 148, it will be observed that the pressure drops progressively in all the rows of blades and the steam is accelerated in each row. The rotating blades recede at such a rate, however, that the entrance velocity plus the velocity acquired due to the nozzle action of the rotating blades is reduced to nearly zero as the steam leaves the rotating blades. The dotted line beginning at the first row of moving blades shows how the velocity would increase if the rotor were held stationary. In that case the entire turbine would act simply as a large expanding nozzle.

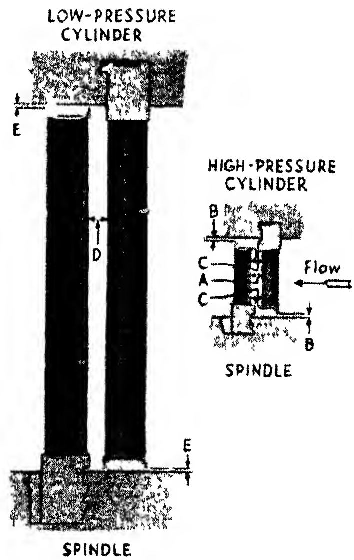


Fig. 150.—Allis-Chalmers reaction blading with roots cast in foundation ring.

156. Parsons Turbines. The "reaction" turbine was developed by Sir Charles Parsons about the same time that De Laval was producing the impulse machine, and for this reason the blading arrangement just outlined is generally called *Parsons' blading.* A typical Parsons spindle, built by the Allis-Chalmers Company, is that shown in Fig. 149. The high-pressure steam enters the small blade on the small drum at A and moves toward the large blading at B. The blades are arranged in steps to avoid too large a number of sizes. The chief characteristic of the Allis-Chalmers blading is the channel "shroud" ring to which the tips of the blades are fastened. Figure 150 shows this blading with cast foundation rings. Figure 151 shows the velocity diagram in impulse-reaction blading. In this type the bucket velocity may be made relatively low because of the large number of stages and relatively low steam velocities. The axial component of the velocity produces an axial thrust of large magnitude which must be counteracted, and, to accomplish this, the spindle of Fig. 149 has two *dummy pistons C* which are subjected to an axial pressure in the opposite direction. These dummies have grooves, which, with corresponding fins in the casing, form a "labyrinth" to retard the leakage of the steam, see Art. 158.

Figure 152 shows an axial section of the Westinghouse reaction blading of the smaller (high-pressure) sizes, having shroud rings

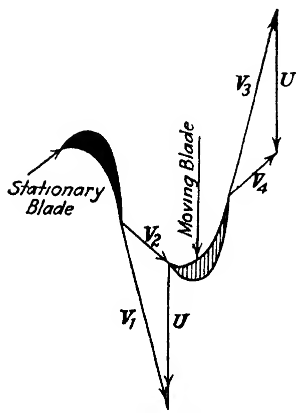


FIG. 151.—Velocity diagram for impulse-reaction blading.

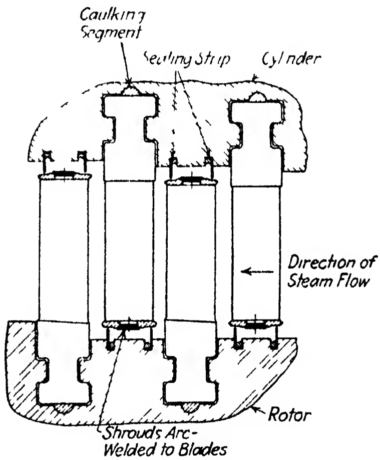


FIG. 152.—Westinghouse reaction blading.

arc-welded to the blades. Figure 153 shows a typical Westinghouse rotating-reaction blade for the low-pressure elements which are subject to erosion from condensed steam. This blade has a protecting strip of Stellite welded to the *back* of the leading edge. Since the lineal speed at the tip is considerably greater than at the root, these long blades must be given a twist to maintain the proper relative entrance angle. The blades are welded together at the streamlined lacing "wire" projections. Figure 154 is a segment of low-pressure stationary blades for a 165,000-kw turbine.

A sectional view of a large Allis-Chalmers Parsons turbine is shown in Fig. 155. Steam enters the turbine through pipe *A* just left of the two balance pistons *B*. After passing the first half of the high-pressure blading, the steam is led back to a reheater where it is reheated and returned to the turbine. The intermediate section *C* expands the steam down to about atmospheric pressure and then exhausts to the inlet *E* of the low-pressure section of the turbine *G* In *G* the steam flows equally toward right and left in a *double-flow* arrangement, thus balancing the end thrust, and then exhausts downward through *FFF* into the condenser at about 1 in Hg abs Steam for heating the feed water is bled off at four points, in addition to the reheat point, through connections *H*.

Figure 156 shows the double-flow, low-pressure spindle of a compound turbine being lowered into the cylinder

FIG 153 — Large Westinghouse reaction blading

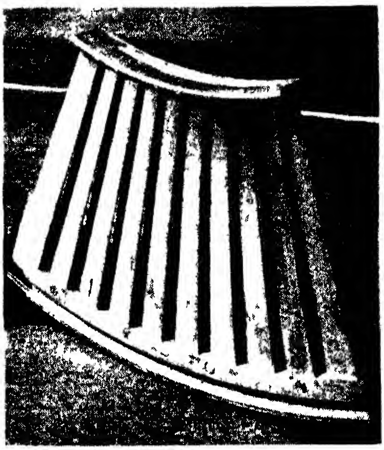


FIG 154 — Westinghouse low-pressure stationary blading segment for 165 000-kw turbine.

157. Combination Impulse and Reaction Turbines.—One disadvantage of the Parsons type of machine is that the spindle becomes very long and tends to distort. An initial Curtis stage is frequently used, to shorten the turbine, and this might be classed as a combination Curtis-Parsons turbine. Figure 157 shows such a turbine built by Westinghouse having one Curtis stage followed by 30 Parsons stages.

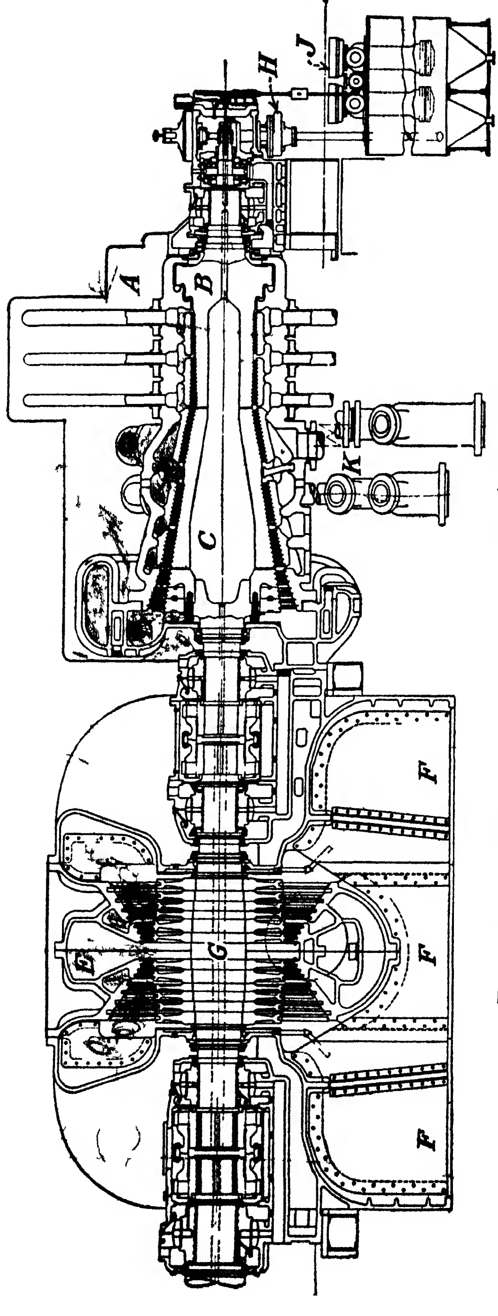


FIG 155.—Axial section of an 80 000-kw Allis-Chalmers reaction turbine

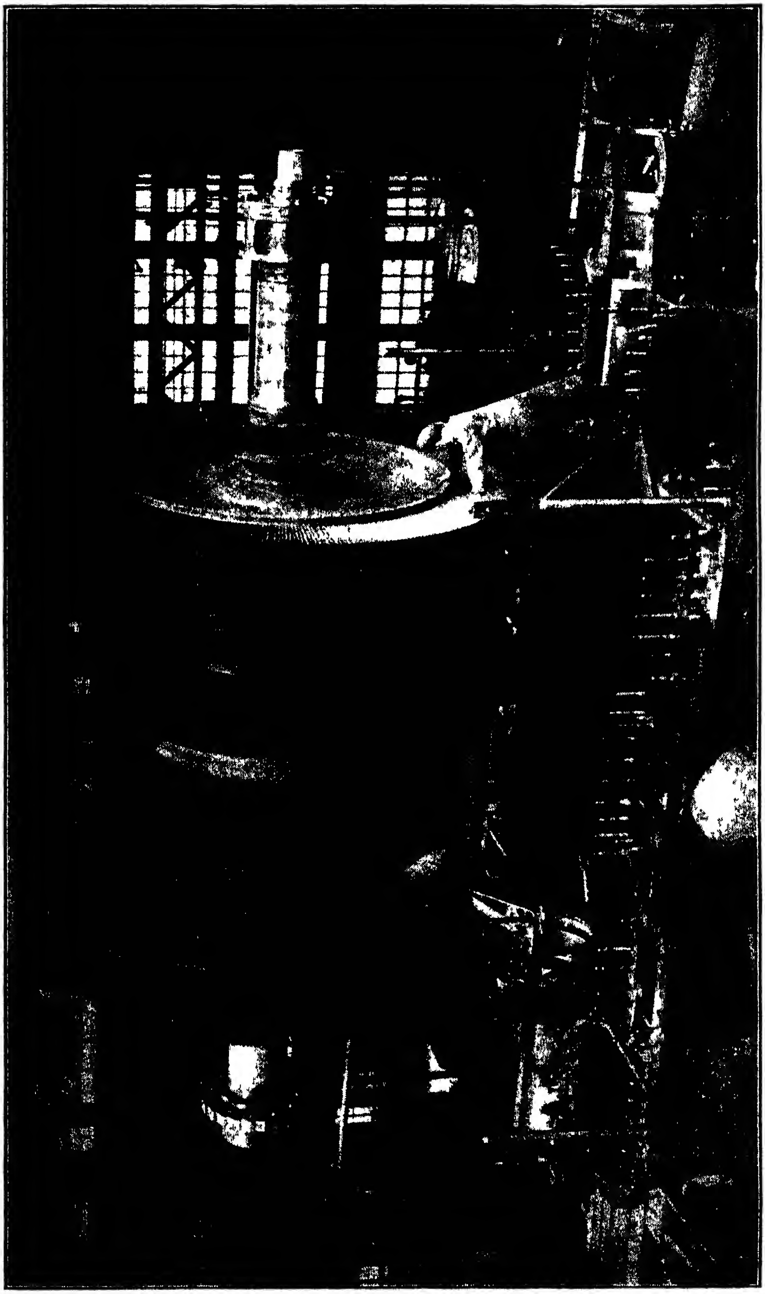


Fig 156 —Spindle and lower half of cylinder of Westinghouse double-flow turbine

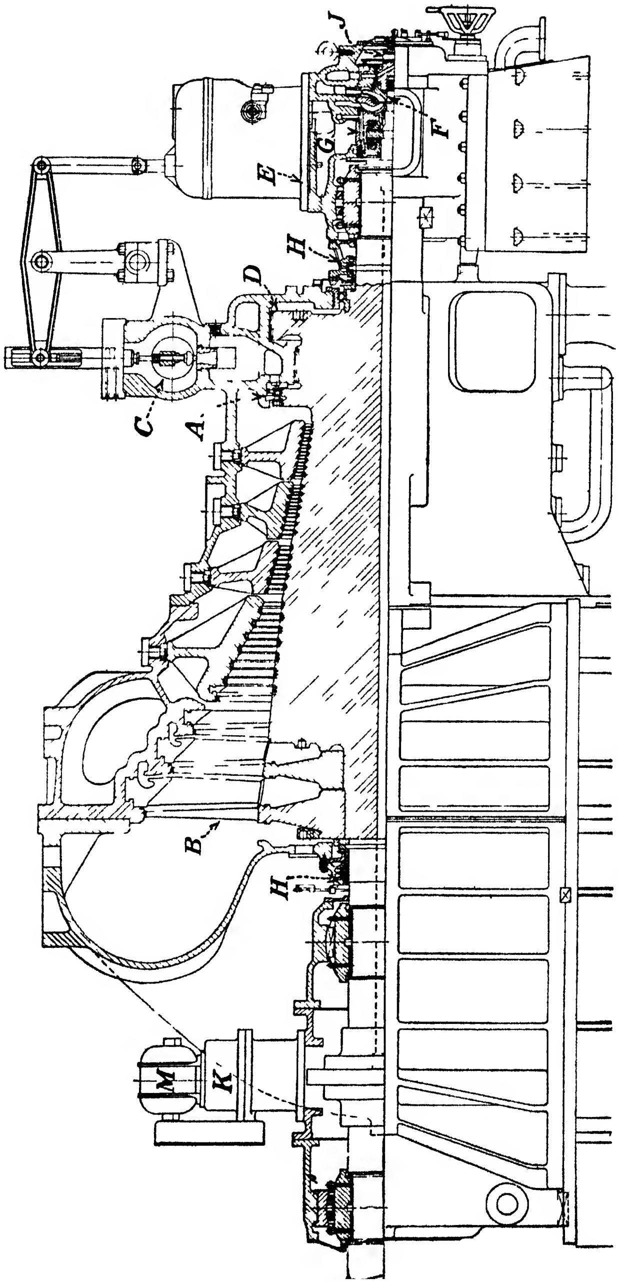


Fig. 157 — Westinghouse combination Curtis-Parsons turbine.

Another combination of the impulse and reaction principles is the Rateau-Parsons type as built by Brown-Boveri in Europe and sometimes seen in American plants on feed heating turbines. Figure 158 shows a section of the high-pressure end of a Rateau-Parsons noncondensing extraction turbine.

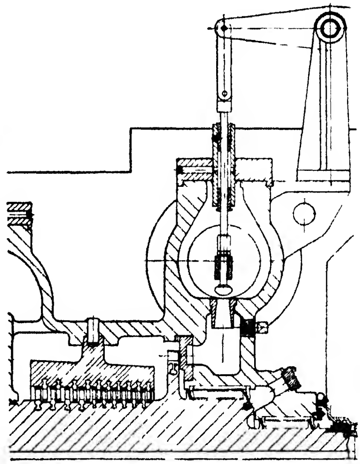


Fig. 158.—High-pressure end of a 50,000-kw Westinghouse extraction turbine of Rateau-Parsons type.

158. Steam Turbine Details. The successful performance of a modern steam turbine depends perhaps as much on the reliable functioning of a number of mechanical details as upon the fundamental design of the machine. Several of these details will be considered here, no attempt being made to mention them in order of importance.

Figure 159 shows a typical water-seal gland to prevent steam from escaping past the shaft at the high-pressure end of a spindle and to prevent air from

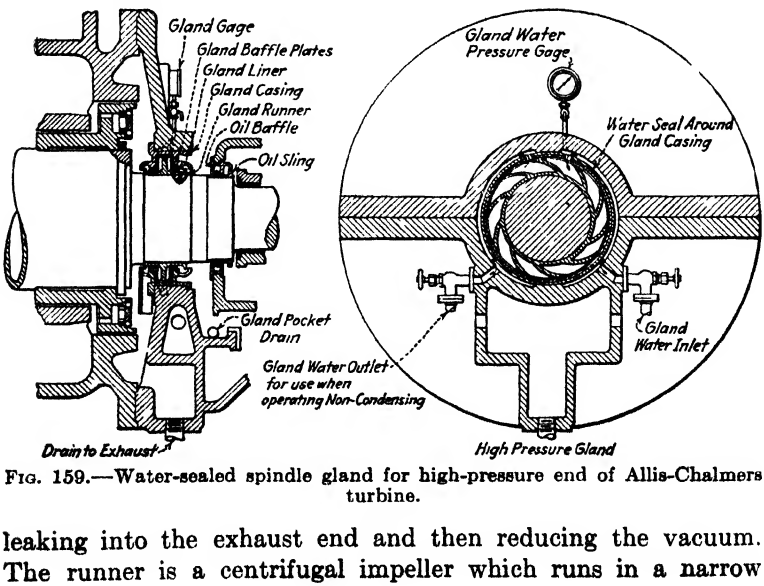


Fig. 159.—Water-sealed spindle gland for high-pressure end of Allis-Chalmers turbine.

leaking into the exhaust end and then reducing the vacuum. The runner is a centrifugal impeller which runs in a narrow housing and throws a solid wall of water across the runner clearance condensate, which is free from dissolved air, usually used to supply these glands. The gland runner is visible just above the man's head, Fig. 156.

In order to minimize the leakage of steam from one stage to the next in a Parsons turbine, a "labyrinth" seal is provided as shown in Fig. 152. A similar labyrinth seal is provided for the axial clearances at balance pistons as shown in Fig. 160. The seal strips are thin so that a rub against the rotating parts will wear them down to proper clearance without serious heating. The theory of the labyrinth is that, as the steam flows through the narrow clearance, it suffers an adiabatic drop which gives it a high kinetic energy, this energy being consumed by swirls set up

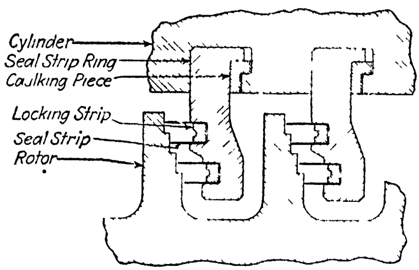


FIG. 160.—Westinghouse axial-clearance labyrinth seal.

by obstructions in the path. What steam does leak past is carried away either to a lower pressure stage or to the exhaust. The labyrinth finds many applications in apparatus in which a frictionless seal is desired.

In Rateau turbines carbon-packing seals are frequently used between stages as shown in Fig. 140.

Since the axial clearances in all turbines are small, being of the order of a few hundredths of an inch, it is necessary to have an adjustable thrust bearing to hold the spindle in place—adjustable to care for changes in length of spindle and cylinder. Such a thrust bearing of the Kingsbury type is shown in the photograph of Fig. 156 at the extreme right end of the shaft. This bearing consists of a disk on the shaft on either side of which are borne segmental blocks held in the spider on spherical seats so as to adjust perfectly to the disk, the whole being flooded by forced lubrication. The spider is adjustable axially in its seat in the cylinder. A simple adjustable thrust bearing is easily seen in Fig. 140 at the right end of the shaft.

The main bearings of large turbines are made self-aligning by having spherical seats as may be seen in Figs. 144 and 149. These seats are provided with shims to allow for adjustment of the radial clearances. Lubrication is by forced oil circulation furnished by pump driven from the main shaft. The spindle of Fig. 157 has a centrifugal oil impeller *F* for lubricating and con-

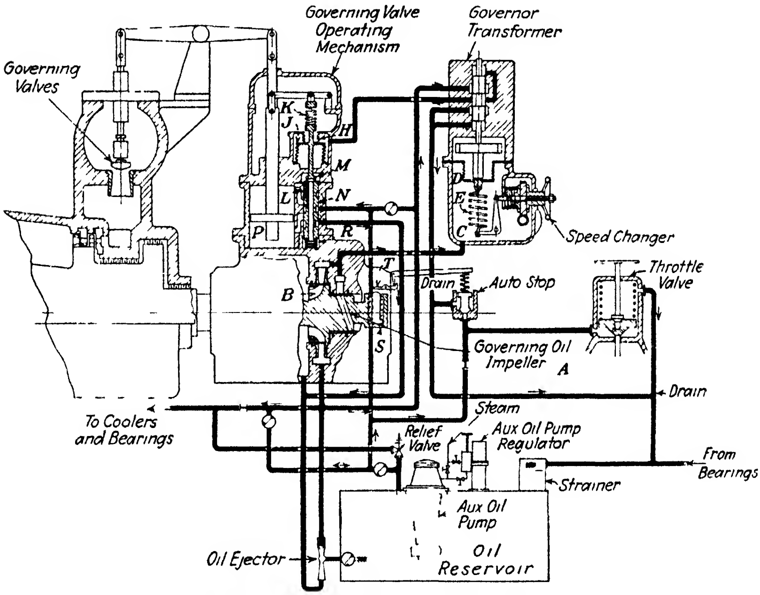


FIG. 161.—Westinghouse oil governor in diagrammatic representation

trol requirements. The turbine of Fig. 155 has a gear pump *H* driven by a vertical shaft off the main shaft.

In order to float the spindle on a film of oil before it is started and to insure against failure of the main pump an auxiliary pump usually driven by a small turbine is provided. Such an auxiliary pump is seen at *J*, Fig. 155 and at the bottom of Fig. 161.

Provision is also made for cooling the circulating oil supply which is accomplished by water coils supplied by the cold condensate.

Governing of small turbines is achieved satisfactorily by a flyball governing element which acts directly on a balanced throttling valve, as may be seen in Figs. 135, 138, 140, and 142. For large turbines the power required is too great and the accuracy demanded is too exacting for such a simple device. The Woodward hydraulic governor of Fig. 31 with flyballs as the speed-sensitive element is applicable to steam turbine governing. Figure 161 shows the principle of the Westinghouse oil governor, the speed-sensitive element of which is an oil impeller *A* consisting of a cylindrical hole drilled at an angle through the shaft. Oil is supplied by the main impeller *B*. Excess oil is thrown out of the open end of *A* into a drain. Suppose an increased load slows down the spindle slightly. The pressure delivered by the governing impeller, which varies as the square of the speed, is slightly reduced. This reduces the pressure in chamber *C* under the diaphragm *D* which allows spring *E* to pull the pilot valve down, permitting oil from the main impeller *B* to flow past valve into chamber *H*. Then the increased pressure on diaphragm *J*, which has been balanced by the spring *K* and the pressure of the oil delivered through the small hole *L* under *M*, causes *M* to lower and with it the relay valve *N*. This admits oil from the center space of *N* to the space above servomotor piston *P*, which moves downward and opens, somewhat, the governing valves.

Now this upward movement of *P* reduces the pressure of spring *K* which allows the oil flow through *L* and out downward through the hole in *N* to the drain, to release the pressure and allow spring *R* to return *N* to its mid-position and thus halt the movement of servomotor piston *P*. The speed now returns to normal and the transformer pilot valve to its mid-position.

In addition to holding the speed of the generator to within a fraction of a cycle the governor must act to shut off the steam entirely within a very short time—measured in seconds—in case of a drop of the generator load, to prevent overspeeding of the machine. To insure against failure of the governor, an overspeed trip *S* is provided. The increased centrifugal force at overspeed throws out the cylindrical weight which strikes the lever *T* releasing the auto stop. This reduces the pressure under the throttle-valve plunger which allows the spring tension to close the valve.

The oil ejector shown near the bottom of Fig. 161 is for the purpose of delivering oil to the main impeller suction.

The Westinghouse steam chest which is somewhat similar to that on the General Electric turbine of Fig. 146 is shown in more detail in Fig. 162. As the governor raises the horizontal bar, the valves open successively, the steam passages 1, 2, 3, etc., thus producing throttling only in the last valves to open.

Turbines of over 15,000-kw capacity are commonly provided with turning gears to rotate the spindle slowly—1 to 5 rpm—during idle periods, to prevent unequal cooling of the spindle

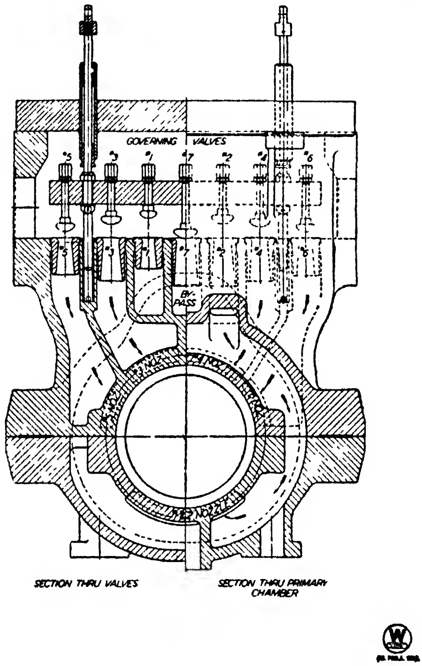


Fig. 162.—Westinghouse turbine steam chest.

which causes unbalance and roughness on starting up. The turbines of Figs. 146 and 157 have such turning gears shown at *M*.

159. Classification of Steam Turbines.—The table on page 254 is a recapitulation of the foregoing paragraphs. The names of the manufacturers are those whose turbines are illustrated here and by no means cover all the American builders.

Turbines are classified as "condensing" if the exhaust pressure is lower than atmospheric and "noncondensing" if the exhaust pressure is greater than atmospheric, regardless of what is done with the exhaust steam. It is frequently desirable to extract

TABLE XIII.—STEAM TURBINES

Technical class	Commercial name	Characteristics		Commercial		Built by
		Nozzles	Blading	Speeds, rpm	Sizes, kw	
Impulse turbines:						
1. Single stage	DeLaval	1 set	1 set rotating	7,500 to 20,000	1 to 300	De Laval Company
2. Velocity stage a	Curtis	1 set	2 sets rotating and 1 set static	3,600	1 to 500	General Electric
2. Velocity stage b	Reentry	1 set	1 set rotating and 1 or more sets of static	3,000 to 7,200	1 to 500	Westinghouse
2. Velocity stage c	Terry	1 set	1 rotating tangential set	3,000 to 4,500	1 to 1,500	Terry Turbine Company
3. Pressure staging	Rateau	2 or more sets	Same number sets rotating	1,200 to 3,600	1 to 1,000	Elliott Company General Electric
4. Pressure and velocity staging	Curtis	2 or more sets	2 sets rotating and 1 set static per set of nozzles	1,200 to 3,600	500 to 6,000	General Electric
5. Combined type	Curtis-Rateau	1 Curtis	Several Rateau	1,200 to 1,800	500 up	Elliott Company General Electric
Reaction turbines:						
1. Pressure staging	Parsons	None	Many rotating and same number static	1,200 to 1,800	1,000 up	Allis-Chalmers
Combined impulse and reaction turbines:						
1. Pressure and velocity staging	Curtis-Parsons	1 set	......	1,200 to 3,600	1,000 up	Westinghouse Allis-Chalmers
2. Pressure staging	Rateau-Parsons	Several	......	1,200 to 3,600	1,000 up	Brown Boveri Westinghouse

some of the steam from a turbine at a pressure somewhere between the inlet and exhaust pressures. Such turbines are called "bleeder" or "extraction" turbines.

160. Large Turbines.—Each year finds the capacities of steam turbine units steadily increasing. Single-cylinder turbines are being built in sizes up to 75,000 kw, and compound turbines over 200,000 kw; the end is not in sight.

The concentration of large capacities in a single cylinder has certain disadvantages, and these have led to many arrangements[1]

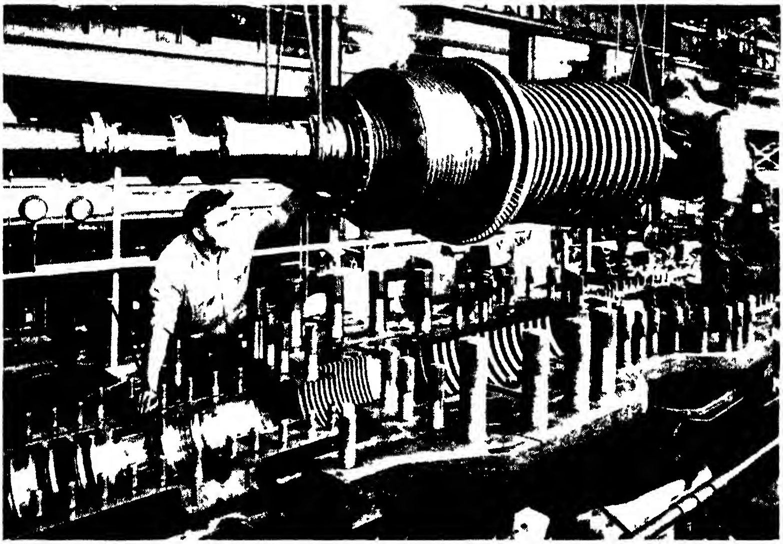


FIG. 163.—Westinghouse 50,000-kw, 3,600-rpm, superposed steam spindle

to provide high efficiency and, at the same time, provide great reliability. One such arrangement is to make the spindle in two parts and connect them in tandem to the same generator, the high-pressure exhausting into the low-pressure element. This is called a *tandem-compound* turbine, see Fig. 155. Another method is to place the high-pressure and low-pressure elements side by side in a *cross-compound* arrangement, each element driving its own generator. Other turbines have three cylinders, sometimes arranged in tandem and sometimes cross compound with one high-pressure cylinder and two low-pressure cylinders, each having its own generator.

[1] *Power*, Nov. 6, 1928, p. 754.

In recent years a new classification of turbine has found wide application which is known as the "superposed" or "topping" turbine. As steam at pressures of 1,200 to 2,400 and temperatures up to 1000°F. have become practical through the development of suitable alloys for boilers and turbines, the existing plants designed for, say, 400 to 600 lb pressure have been modernized by the addition of high-pressure boilers supplying steam to a high-pressure noncondensing turbine; this exhausts into the old steam turbines. This is, in effect, cross compounding. Savings in fuel by topping have been as high as 30 per cent and many old plants have attained economies equaling those of modern high-

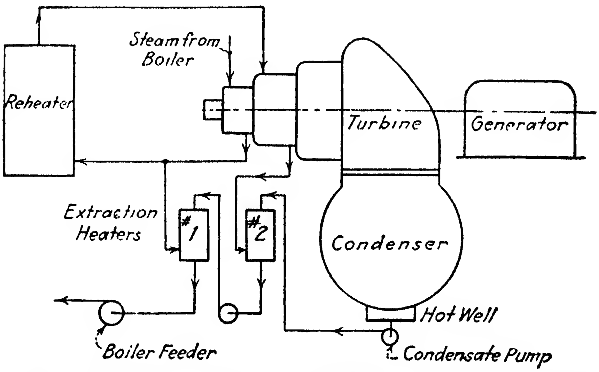


FIG. 164.—Steam-flow diagram of reheating, regenerative turbine.

pressure plants. Figure 163 shows the spindle of a modern topping turbine operating at 1,200 lb and 900°F. and exhausting at 200 lb back pressure.

Some compound machines pass the steam exhausted from the high-pressure cylinders into *reheaters*, which raise the temperature but not the pressure of the steam before passing it into the low-pressure cylinder or section, thus increasing the efficiency of the entire combination. The reheating may be done either with live steam or by fired reheaters. The turbine of Fig. 155 is a notable example.

Another method for improving the efficiency of the turbine plant is to bleed steam from the turbine at various points after partial expansion has been accomplished and to use this low-pressure steam for heating the condensate as it passes back to the boiler. This is called "regenerative" heating of the feed water, and the cycle is called a "regenerative" cycle. The openings under the cylinder of the turbine of Fig. 155 are provided for extraction of steam for feed-water heating.

Still other plants provide for a combination of reheating and extraction heating, and in some cases the actual efficiency has been greater than that of the Rankine cycle. Figure 164 shows a diagrammatic arrangement of a reheating, regenerative turbine.

161. The Field of the Steam Turbine.—Since the steam turbine is a high-speed, constant-speed machine, it is particularly adapted

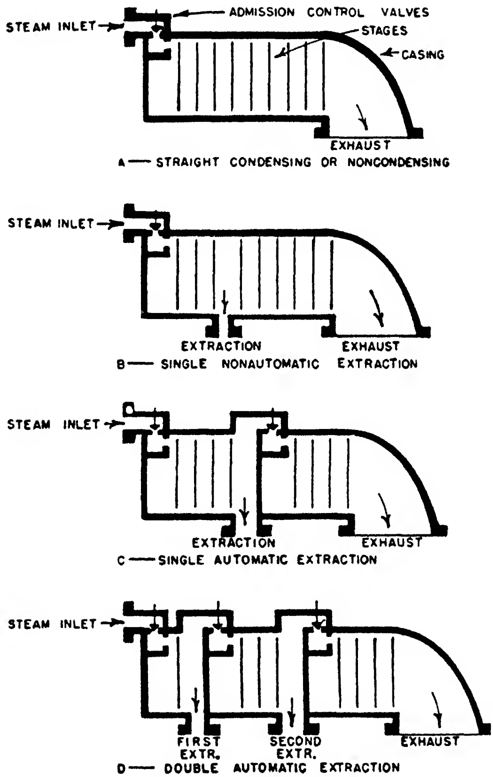


FIG. 165.—Diagrammatic representation of extraction turbines (*Power Plant Engineering*).

for driving electric generators, centrifugal pumps, blowers, and, when provided with reducing gears and a reversing turbine, for ship propulsion. Since turbines are designed to run at a particular speed for good efficiency, it is not economical to operate them at varying speed, although variable-speed installations are made in which economy is not vital. As central-station prime movers, turbines are at their best because of their close regulation, high economy over wide range of loads, small space per kilowatt of capacity, and the enormous capacities for which they can be built. From the boiler standpoint there is a great advantage in that the exhaust steam, when condensed, is free from oil for reuse as boiler feed.

In the industrial steam plant the turbine has proved to be a very flexible and versatile drive. Here frequently a manufacturer needs both electric power and steam for processes or for building heating, and for this situation the steam turbine is well adapted. If power is generated as a by-product the turbine exhausts into the factory steam mains and simply acts as a reducing valve. This is illustrated diagrammatically at *A*, Fig. 165. When steam is extracted from *A* with no regulator, as at *B*, the pressure will vary with the power load as the inlet pressure is controlled by the governor. Turbines are sometimes provided with a second set of inlet nozzles beyond the extraction point which is controlled by the extraction pressure as shown at *C*. If extracted steam is wanted at two different pressures, the arrangement at *D* is employed.

In general, piston engines are better adapted to the high pressures, while turbines are better adapted for high vacuums.

162. Performance of Steam Turbines.— The steam consumption of steam turbines varies greatly with the size of the machine, initial steam pressure and temperature, and the back pressure. In general, the best economies are made by the large turbines, because the smaller machines do not ordinarily use so high pressures, superheat, or vacuum.

Economies and efficiencies of turbines are usually calculated on the basis of the electrical output. If the generator efficiency is known, the performance may be reduced to a shaft basis. It is not possible to segregate the friction losses except by very elaborate tests, and so the term "mechanical efficiency" is not used in connection with turbines.

Thermal efficiency is computed in the same manner as for steam engines and pumps, the turbine being given credit for the heat of the liquid of the exhaust. In plants in which steam is "bled" or "extracted" from certain stages of the turbine for feed-water heating, the turbine is also given credit for this heat.

Figure 166 shows graphically the ordinary range of steam rates to be expected from steam turbines of various sizes, most turbines falling between the two lines.

The performance of a steam turbine varies with the load in nearly the same manner as that of a reciprocating engine. The hourly steam consumption plotted against load gives a straight line, or "Willans's line," for turbines with throttling governors.

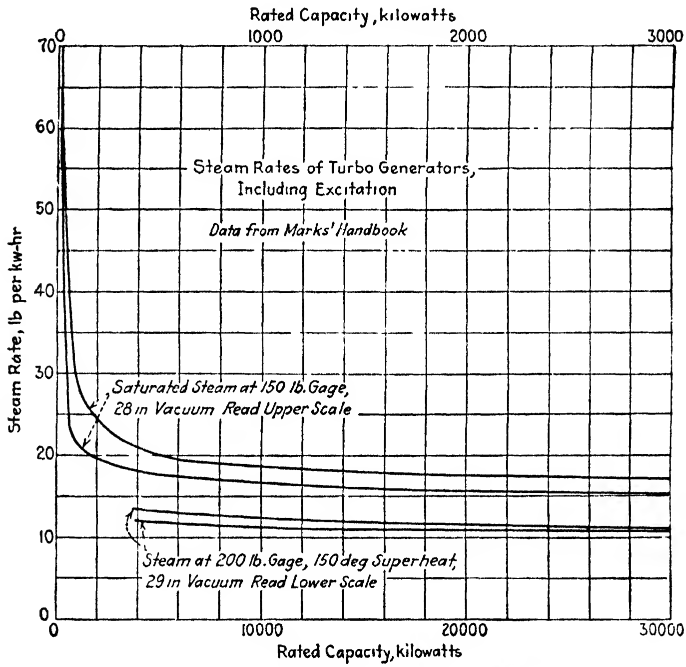


FIG. 166.—Steam rates of turbines. (*Data from Marks' Handbook.*)

For turbines having two or more inlet valves operated by the governor, the steam consumption line is a broken line, changing direction at the load at which each additional inlet valve is brought into service.

Problems

1. What will be the theoretical spouting velocity of a water jet passing through an orifice from a pressure of 100 lb per sq in. gage into the atmosphere?

2. What will be the theoretical jet velocity of dry steam expanding adiabatically through an orifice from 100 lb gage into the atmosphere at 14.7 lb?

3. A steam nozzle expands steam from 175 lb per sq in. abs and 98 per cent quality to 80 lb abs. If the expansion is adiabatic, what will be the theoretical jet velocity?

4. The first stage nozzles of a turbine expand steam from 275 lb per sq in. abs and 250° superheat to 110 lb abs. If 10 per cent of the available energy is returned to the steam by friction, calculate (*a*) the jet velocity; (*b*) the condition of the steam leaving the mouth of the nozzle.

5. Steam at 125 lb per sq in. abs and a quality of 98 per cent is expanded through a nozzle to a pressure of 55 lb abs. If 8 per cent of the available energy is returned to the steam by friction, calculate (*a*) the jet velocity; (*b*) the condition of the steam leaving the mouth of the nozzle.

6. A steam turbine having a throttling governor receives steam at 198 lb per sq in. abs and at a temperature of 470°F. The governor reduces the pressure, at a certain load, to 59 lb abs at the nozzle. The pressure at the mouth of the nozzle is 7.3 lb abs. Calculate (*a*) the theoretical velocity of the steam leaving the nozzle; (*b*) the state of the steam before and after passing through the nozzle. (*c*) If the actual velocity is 96 per cent of the theoretical, what proportion of the available energy is reconverted into heat by friction?

7. Steam at 195.3 lb per sq in. abs and at 475°F. is throttled by the governor valve to 137.3 lb abs. and then passes through a nozzle to a pressure of 17.1 lb abs. Calculate (*a*) the condition of the steam before and after passing through the nozzle; (*b*) the theoretical velocity of the jet; (*c*) the velocity if 7 per cent of the available energy is reconverted into heat by friction.

8. A single-stage impulse wheel moves at a blade speed of 600 ft per sec. The jet velocity is 1,400 ft per sec, and the nozzle makes an angle of 20 deg with the plane of the wheel. Neglecting blade friction, determine graphically and check analytically (*a*) the blade angles (make blade symmetrical with plane of rotation); (*b*) the absolute exit velocity; (*c*) the tangential components of the absolute entering and exit velocities; (*d*) the horsepower developed per pound of steam per second; (*e*) efficiency.

9. The turbine of Prob. 7 is a single-stage, velocity-stage machine with two rows of moving blades and one stationary. The nozzle angle is 20°. The second row of moving blades has such an angle that the tangential component of the exit velocity is zero, and both rows are symmetrical. Blade velocity equals one-fourth that of the jet. Neglecting blade friction, determine graphically and check analytically (*a*) the blade angles; (*b*) the absolute leaving velocity; (*c*) the tangential components of entering and leaving velocities; (*d*) the horsepower developed per pound of steam per second; (*e*) the efficiency.

10. A five-stage Rateau turbine is to be designed to run at 1,800 rpm. The initial steam pressure is 140 lb per sq in. abs and the exhaust pressure 15 lb abs. Assume the enthalpy change to occur in five equal steps and a reduction of 7 per cent in the available energy to be made for friction.

Determine (*a*) the pressure at the end of each stage; (*b*) the diameter of the blade circles, assuming the blade speed to be 40 per cent of the jet velocity. Assume all five disks to have the same mean diameter.

11. A three-stage Curtis turbine is designed to operate on dry steam at a pressure of 125 lb gage and at an absolute exhaust pressure of 1 atm at sea level. Assume that the adiabatic heat drops of the three stages are equal and that 20 per cent of the available energy of each stage is returned to the steam by friction, half of this being in the nozzle. Determine (*a*) the steam state at the end of each stage; (*b*) the nozzle velocities.

12. An experimental single-stage Curtis turbine was subjected to a series of tests, being loaded by means of a hydraulic brake having a brake constant of 0.0007933. The following data were taken:

Run	1	2	3	4	5
Throttle pressure, lb. Abs.	197	195	199	193	195
Throttle temperature, deg. F.	472	470	469	473	475
Back pressure, lb. Abs.	7.3	9.6	11.5	14.5	17.1
Net brake load, lb	368	495	595	774	914
R.p.m	3,549	3,527	3,565	3,590	3,557
Steam condensed, lbs. per hr	26,825	34,375	41,216	51,409	60,535

Calculate for each load (*a*) brake horsepower; (*b*) steam rate per shaft horsepower-hour; (*c*) thermal efficiency. (*d*) Plot curves with total steam (Willans's line), steam rate, and thermal efficiency as ordinates against shaft horsepower as abscissas.

13. A 2,500-kw impulse turbine is guaranteed to give the following performance with steam at 300 lb gage and 125° superheat with a condenser pressure of 2 in. Hg abs. Power factor 80 per cent, except at 1.25 load, which is at 100 per cent.[1]

Load, fraction	1/4	1/2	3/4	Full	1 1/4
Load, kilowatts	625	1,250	1,875	2,500	3,125
Pounds steam per kw.-hr	16.90	14.15	13.15	12.75	12.75

Barometer 29.8 in.

Calculate for each load (*a*) the guaranteed thermal efficiency; (*b*) the Rankine-cycle efficiency ratio. (*c*) Plot the total steam per hour, steam rate per kilowatt-hour, and thermal efficiency against load.

[1] *Power*, 63: 550.

14. A 2,000-kw steam turboalternator[1] running at 3,600 rpm was guaranteed to give the following performance with steam at 200 lb gage and 100° superheat and 2 in. Hg abs. back pressure in the exhaust chamber:

Load, kw. at 80% P. F.	2,000	1,500	1,000
Pounds steam per kw.-hr.	15.1	15.6	17.1

Barometer 14.7. No steam bled.

Calculate for the three loads (*a*) the guaranteed thermal efficiency; (*b*) the Rankine-cycle efficiency ratio. (*c*) Plot total steam per hour, steam per kilowatt-hour, and thermal efficiency against load.

15. The 50,000-kw turboalternators at the Richmond station, designed for extraction heating at four stages, are guaranteed to give a steam rate of 9.55 lb per kw-hr, without bleeding, at 42,000 kw output.[2] Steam pressure at throttle, 375 lb gage; steam temperature, 675°F.; vacuum, 29 in. referred to 30-in. barometer. Calculate (*a*) the thermal efficiency; (*b*) the Rankine-cycle efficiency under these conditions.

16. A 35,000-kw turboalternator[3] is guaranteed to give the following performance with steam at 350 lb gage, 635°F. total temperature, and 29-in. vacuum referred to a 30-in. barometer:

Load, kw. at 80% P. F.	43,750	35,000	31,000	25,000	15,000
Steam rate, lb. per kw.-hr. (without bleeding)	10.50	10.20	9.85	10.05	10.70

Calculate for each load (*a*) the guaranteed thermal efficiency; (*b*) the Rankine-cycle efficiency ratio. (*c*) Plot total steam per hour, steam per kilowatt-hour, and thermal efficiency against load.

17. A 20,000-kw turboalternator was guaranteed to give the following performance with steam at 275 lb gage and 200° superheat at the throttle and a 28.5-in. vacuum referred to a 30-in. barometer:

Load, kw.	10,000	15,000	20,000	25,000
Power factor, %	80	80	80	100
Steam rate per kw.-hr.	11.80	11.20	10.81	10.91
Generator efficiency, %	92.7	94.8	95.7	96.5

[1] *Power*, 63: 246.

[2] *Power*, 63: 742.

[3] *Power*, 63: 669.

During the acceptance tests, the following data were taken:

Load, kw	5,467	10,495	15,565	20,620	25,393
Power factor, %	0.850	0.808	0.810	0.808	0.980
Throttle pressure, lb. Abs	288.7	281.4	286.2	281.5	272.1
Throttle temperature, ° F	575	578	600	606	593
Vacuum, in. Hg	28.18	28.31	28.23	28.9	28.00
Barometer, in. Hg	29.74	29.76	29.75	29.76	29.76
Total steam, lb. per hr	78,875	122,220	169,700	219,330	272,100

Calculate for each load (*a*) the guaranteed thermal efficiency; (*b*) Btu chargeable to turbine per pound of steam; (*c*) steam rate per kilowatt-hour; (*d*) actual thermal efficiency; (*e*) shaft horsepower; (*f*) steam rate per shaft horsepower-hour. (*g*) Plot curves with total steam, steam per kilowatt-hour, and guaranteed and actual thermal efficiencies against load.

18. A 60,000-kw, three-cylinder turboalternator was guaranteed to give the following performance with steam at 265 lb gage and 175° superheat with 29-in. vacuum referred to 30-in. barometer:

Load, kw	40,000	50,000	60,000
Steam rate, lb. per kw.-hr	10.79	10.58	10.79

The three generators were each guaranteed to give the following efficiencies:

Load, kw	10,000	15,000	20,000
Overall efficiency, %	91.9	93.9	94.5

Calculate (*a*) the Rankine-cycle efficiency of the guarantee conditions and, for each load, (*b*) the thermal efficiency referred to electrical output and to shaft output.

19. An impulse steam turbogenerator rated at 2,000 kw[1] was guaranteed to give the following performance with steam at 200 lb gage, and 100° superheat, 7.5 lb gage pressure at the extraction opening, and 2 in. Hg abs. back pressure in the exhaust chamber:

Load, kw at 80% P. F.	2,000	1,500	1,000
Pounds steam extracted per hr	25,000	25,000	25,000
Total steam supplied per hour	44,500	38,000	31,800

Barometer 14.7 lb.

[1] *Power*, 63: 245.

Calculate for the three loads (*a*) the guaranteed thermal efficiency; (*b*) the efficiency of the ideal extraction cycle for the conditions given.

20. A Terry steam turbine driving a 200-kw exciter[1] is guaranteed to deliver a kilowatt-hour at full load with steam at 275 lb gage and 125° superheat and exhausting at 1 lb gage, with barometer at 30 in. on 56.5 lb of steam. Calculate (*a*) the thermal efficiency of the unit referred to electrical output; (*b*) the Rankine-cycle efficiency; (*c*) the Rankine-cycle efficiency ratio.

21. A 12,500-kw, nine-stage, Curtis turboalternator[2] is guaranteed to give a steam rate of 10.4 lb per kw-hr at a load of 9,375 kw with steam at the throttle of 300 lb gage, 635° total temperature, and with 29-in. vacuum referred to a 30-in barometer. Power factor 80 per cent. Calculate (*a*) the thermal efficiency; (*b*) the Rankine-cycle efficiency ratio.

[1] *Power*, 65: 158

[2] *Power*, 66: 196.

CHAPTER XVI

CONDENSING EQUIPMENT

163. Condenser Types.—There are two general methods of condensing the steam exhausted from engines or turbines. One is to mix cold water with the steam, and this is called *mixed* or *jet* condensation. The other method is to pass the steam over cold metallic surfaces which are kept cold by water circulation. This method is called *surface* condensation.

Jet condensers may be of the *rain*, *low-level*, *barometric*, or *ejector* type. The rain type is not used extensively in this country, except in open feed-water heaters, see Art. 217, although a recent large installation of this type has been made.[1] Surface condensers may be of the *standard* type or of the *waterworks* type.

164. Low-level Jet Condensers.—Figure 167 shows a sectional view of a Westinghouse jet condenser for use with a steam turbine. The steam enters from above, where it meets a horizontal sheet of water sprayed from the ring of jets around the top. The water may be delivered to the water box by a centrifugal pump or may be drawn in by the vacuum in the condenser.

As the steam is condensed, the condensate and injection water drop to the bottom where the mixture is removed by the centrifugal removal pump. In case of failure of the removal pump, the water level will rise in the condenser, but, to prevent the water backing up as far as the turbine, a float valve is provided which is opened up by the water, thus allowing air to rush in and break the vacuum.

Since the injection water always contains air in solution and the steam contains air which has entered through glands and other leaks, as well as air and other gases which were in the feed water, it is important to have some means of removing this air which collects in the condenser and impairs the vacuum. The annular space around the central passage is arranged to collect

[1] See *Power Plant Eng.*, July 1, 1926.

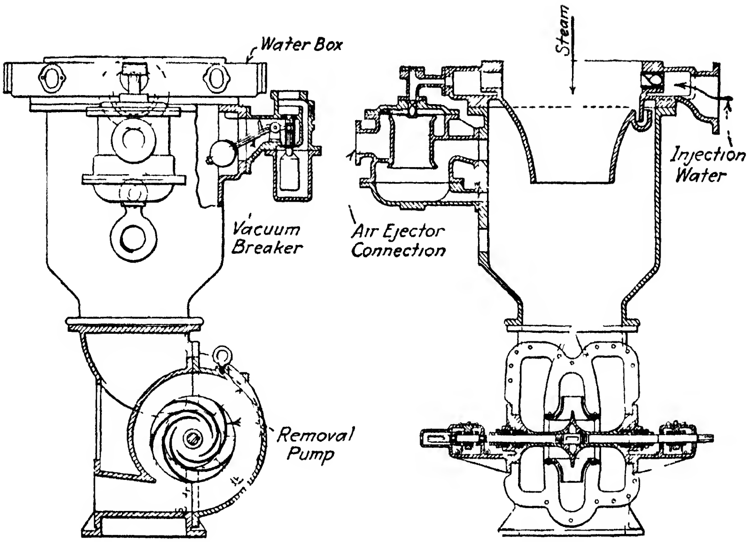


FIG 167 — Westinghouse low-level jet condenser

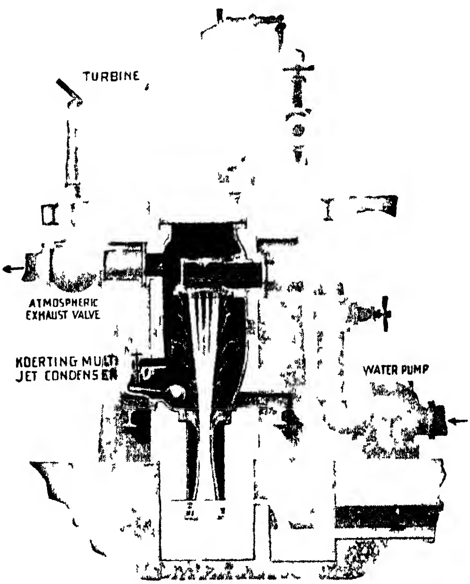


FIG 168 —Schutte-Koerting multijet condenser.

this air, and an air pump is provided which withdraws and compresses the air and delivers it to the atmosphere.

165. The Ejector Condenser.—Figure 168 shows a jet type of condenser operating on a slightly different principle. The

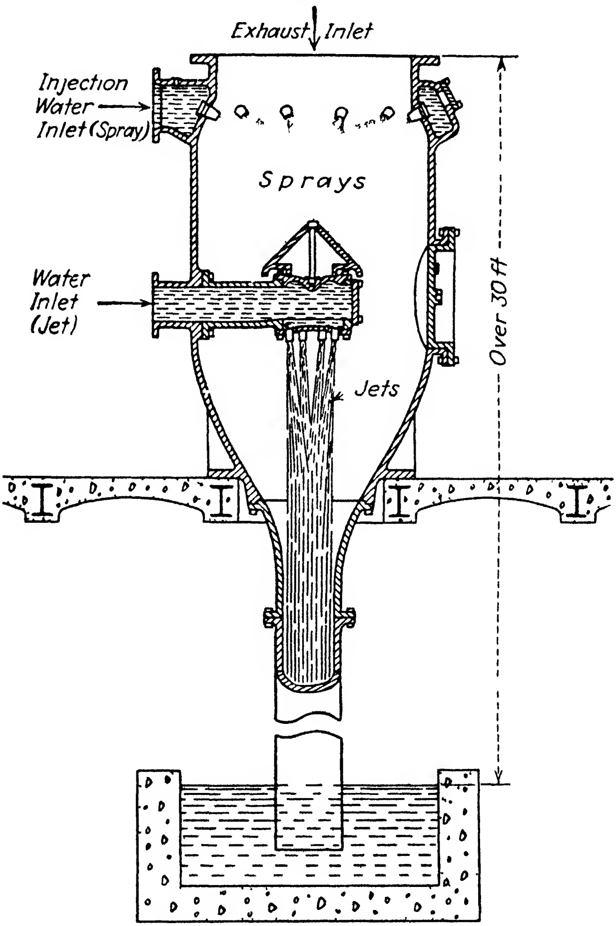


FIG. 169.—Schutte-Koerting barometric condenser.

injection water enters at the top and is directed downward in several powerful jets with sufficient kinetic energy to force itself out through the venturi throat to the atmosphere, carrying with it the condensate and the entrained air. Hence no removal pump is required, and, if a long tail tube is used, the vacuum breaker is not needed.

In this condenser the high-velocity jets draw out the entrained air as well as the steam or condensate, and an air pump is not necessary.

166. The Barometric Condenser.—If a jet condenser is elevated so that the tail pipe is longer than the water column that can be supported by the atmospheric pressure (about 34 ft.), the water will run out by gravity in spite of the vacuum inside. Such an arrangement, called a "barometric condenser," is shown in Fig. 169. This condenser has the advantage of not requiring a removal pump or a vacuum breaker but has the disadvantages (1) that a long exhaust pipe is usually required between the engine or turbine and the condenser head and (2) that the injection water must be pumped against a rather high head. While first cost may be low, a high operating cost may more than offset that advantage.

167. Water Required for Mixed Condensation.—The enthalpy possessed by each pound of exhaust steam as it leaves the engine or turbine may be represented by

$$Q_e = h_f + xh_{fg}$$

But, since the quality x is not often known and is difficult to determine by direct measurement, the following approximate method for finding Q_e is frequently used:

If Q_1 = the enthalpy of the steam at the prime-mover throttle.
s = the steam rate of the prime mover per horsepower-hour.
Q_R = the quantity of heat from each pound of steam supplied that goes into radiation and friction losses. This may be assumed to be about 1 per cent of Q_1.

Then

$$Q_e = Q_1 - \frac{2,543}{s} - Q_R \qquad (110)$$

The heat absorbed by the injection water equals the heat given up by the steam during the same interval of time; therefore

$$W_w(t_o - t_i) = W_e[Q_e - (t_e - 32)] \qquad (111)$$

in which W_w = weight of injection water per hour.
W_e = weight of exhaust steam per hour.
t_e = exhaust temperature.
t_o = outlet-water temperature.
t_i = inlet-water temperature.

It is assumed that $t_e = t_o$ for mixed condensation, which is nearly true except for very high velocities of injection water.

168. Surface Condensers.—Figure 170 shows a *standard-type* surface condenser in which the steam enters the shell at the top and is directed to the outer surfaces of many small brass tubes, where it is condensed. The condensate collects in the bottom of the shell in the "hot well" from which it is removed by some sort of *condensate pump*. The *circulating* or *cooling* water enters the water box at the right, flows through the lower half or *pass* of the tubes to the water box on the left end, and then flows

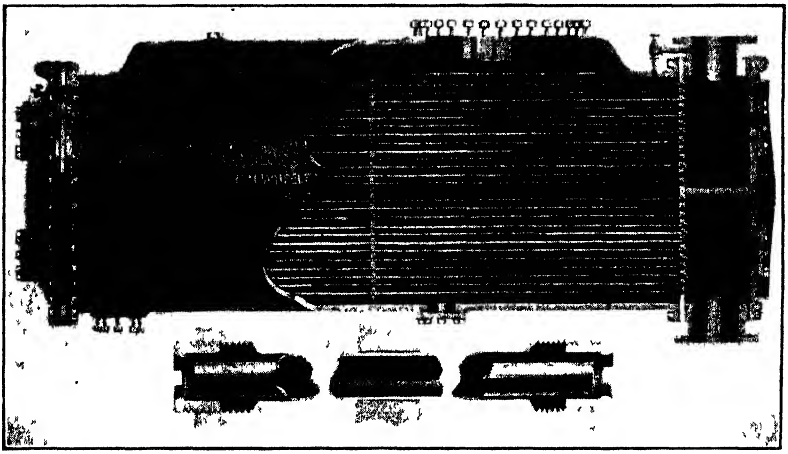


FIG. 170.—C H. Wheeler standard-type surface condenser.

through the upper pass to the right upper water box and out at the top. This is called a *counterflow condenser*, because the progress of the steam is downward while that of the water is upward. This arrangement, which is the usual one, brings the coldest tubes into contact with the coolest condensate and the warmest tubes into contact with the warmer steam, giving the best heat transfer from steam to water. The difference in temperature between steam and condensate, however, is small.

Large, high-vacuum condensers present difficult problems in design, and the several manufacturers have produced different solutions. Figures 171 and 172 show two condensers of 101,000 sq ft each, both designed for the same conditions in the same plant, the shell of Fig. 171 having a triangular form and tube spacing narrowing toward the bottom as the steam volume diminishes. The condenser of Fig. 172 has the tubes arranged in two nests, each of which has lanes which direct the steam and condensate and the entrained air toward the air cooler on each side under the sloping baffle.

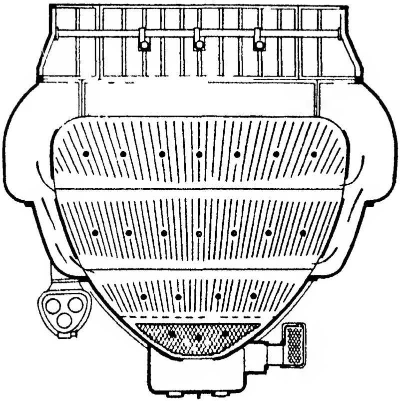


FIG. 171.—Ingersoll-Rand tube arrangement

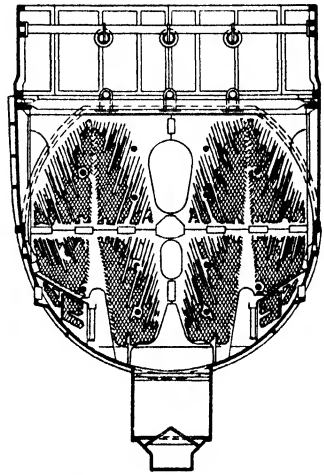


FIG. 172.—Worthington tube arrangement.

Figure 173 shows the *radial-flow* arrangement of the tube nest with air cooler and offtake at the center. This condenser, 113,000 sq ft, was the world's largest when installed in 1935.

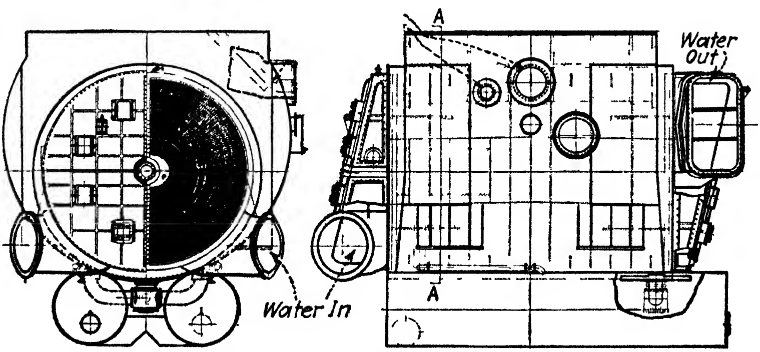


FIG. 173.—Westinghouse 113,000-sq ft single-pass condenser.

Until recently condenser shells have been built of cast iron, but with the development of electric welding sheet-steel construction has become favored by most manufacturers.

The tubes are connected into the tube sheets either by a threaded packing or by rolling the ends into the drilled holes. When the rolled method is used, some means must be provided to care for the difference in expansion between the tubes and the shell.

The tubes in surface condensers are made of special alloys which will withstand corrosion, the most important being Muntz metal, Admiralty metal, arsenical copper, and Monel metal. The size of tubes varies from $\frac{5}{8}$ in o.d. in small condensers to 1 in. o.d. in large condensers, the large size being favored because of ease in cleaning and the smaller number required.

169. Heat Transfer in Surface Condensers.—The amount of heat that can be transferred from the condensing steam to the

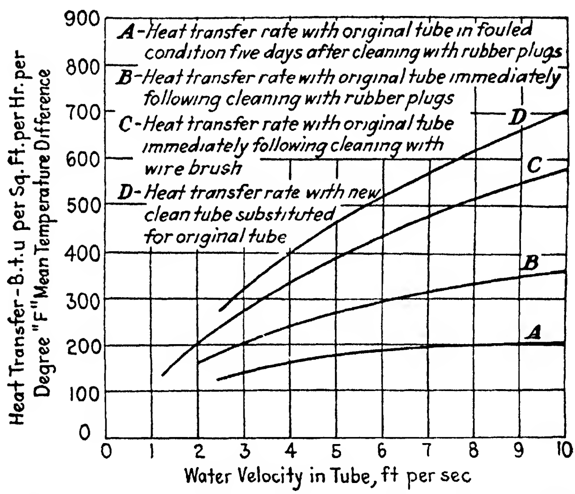


FIG. 174.—Heat transfer through condenser tubes. (*Tests by Westinghouse Electric & Manufacturing Co.*)

water per square foot of tube surface per hour per degree difference in temperature of steam and water depends chiefly upon two things: (1) the cleanliness of the tube surfaces and (2) the velocity of the water through the tube. Figure 174 illustrates the great influence of these two factors.

Since the temperature of the water rises as it passes through the condenser, the "mean"-temperature difference must be used in heat-transfer problems. The value of the mean-temperature difference is obtained as follows: Referring to Fig. 175, assume that the condenser consists of a single pass, and, for simplicity, consider only one tube.

Let t_e = temperature of exhaust steam.
t_i = temperature of inlet water.
t_o = temperature of outlet water.
t = temperature of water at any point.
U = heat-transfer rate in Btu per square foot per hour per degree temperature difference.
w = pounds of water passing through each tube per hour.

Then, if we consider a small elemental area of the tube da at a distance represented by the area a from the end of the tube, the

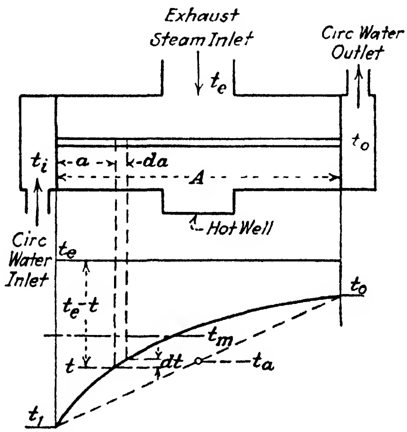


FIG. 175.— Temperature difference in a surface condenser.

temperature difference at this point will be $(t_e - t)$. The heat transferred per hour through the wall of the tube through the area da will be

$$Uda(t_e - t) \tag{112}$$

and the heat absorbed by the water passing over this area will be

$$wdt \tag{113}$$

These must be equal, and therefore

$$U(t_e - t)da = wdt \tag{114}$$

and

$$Uda = \frac{wdt}{(t_e - t)}$$

Integrating between 0 and A, and t_i and t_o, we have

$$UA = w \log_e \left(\frac{t_e - t_i}{t_e - t_o}\right) \tag{115}$$

The heat transferred per hour through the entire tube wall is the same as that absorbed by the water in passing the length of the tube. That is,

$$UA(t_e - t_m) = w(t_o - t_i)$$

and

$$UA = \frac{w(t_o - t_i)}{(t_e - t_m)} \tag{116}$$

Setting Eq. (115) equal to Eq. (116), we have

$$w \log_e \left(\frac{t_e - t_i}{t_e - t_o}\right) = \frac{w(t_o - t_i)}{(t_e - t_m)} \tag{117}$$

from which we have as the mean-temperature difference

$$(t_e - t_m) = \frac{t_o - t_i}{\log_e \left(\frac{t_e - t_i}{t_e - t_o}\right)} \tag{118}$$

This is called the "logarithmic mean"-temperature difference and is smaller than the "arithmetic mean" (which would be $t_e - \frac{t_i + t_o}{2}$) by a large amount for large rises in cooling-water temperature. For surface condensation the weight of cooling water required per pound of steam condensed is slightly greater than for mixed condensation, for the reason that the outlet temperature t_o is lower than the exhaust temperature t_e by 5 to 10°, as is shown in Fig. 175, while in mixed condensation the condensate and the condensing water leave together and at a common temperature.

170. Condenser Auxiliaries.—*Circulating pumps* for surface condensers are usually of the centrifugal type which is well adapted to handling the large quantities of water required at the relatively low heads that are usual even for two-pass condensers. For single-pass condensers the propeller type of pump which is a high-speed, low-head pump, such as that of Fig. 176, is finding favor.

Injection and *removal pumps* for jet condensers usually operate under greater head than circulating pumps for surface condensers.

Condensate pumps for turbines are centrifugals, and, since the total head against which they operate may be over 100 ft, multistage pumps are frequently used. For steam engines reciprocating-condensate or wet vacuum pumps, which remove both air and condensate from the condenser, are generally used.

Air-removal Pumps.—If air were allowed to collect in a condenser, it would greatly reduce the vacuum that could be maintained because of the partial pressure of the air. In the ejector

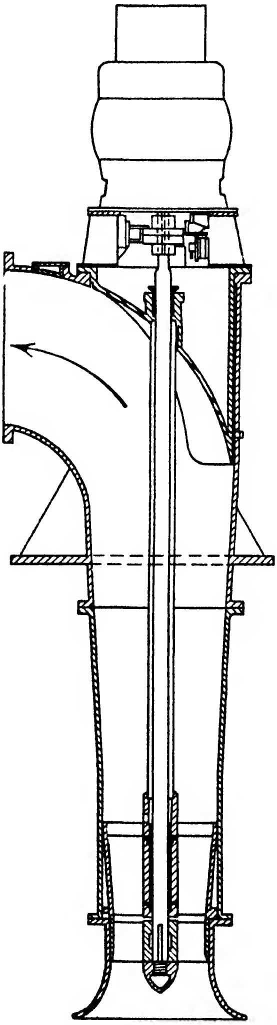


FIG. 176.—Westinghouse propeller-type circulating pump.

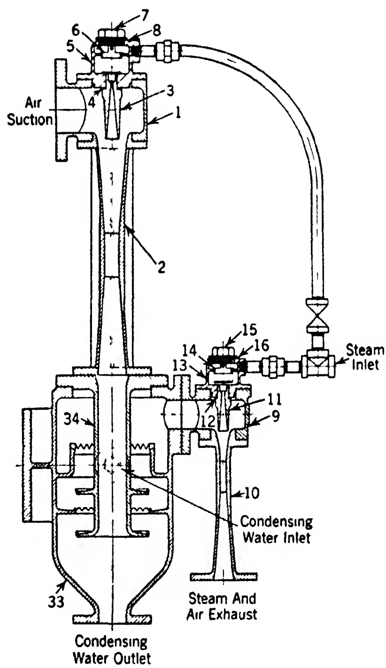


FIG. 177.—Foster Wheeler two-stage steam-jet ejector.

and barometric types of Figs. 168 and 169 the high-velocity water jets produce a suction that carries the air along with the water. Other condensers require a special air pump, which for small installations may be a reciprocating pump. Modern turbine condensers are provided with steam-jet air ejectors, one form of which is shown in Fig. 177. This is a two-stage ejector with an intercondenser of the jet type. From the steam inlet on the right the steam passes through the chamber 5 to the first stage nozzle 3 which directs a powerful steam jet into the diffuser 2. This

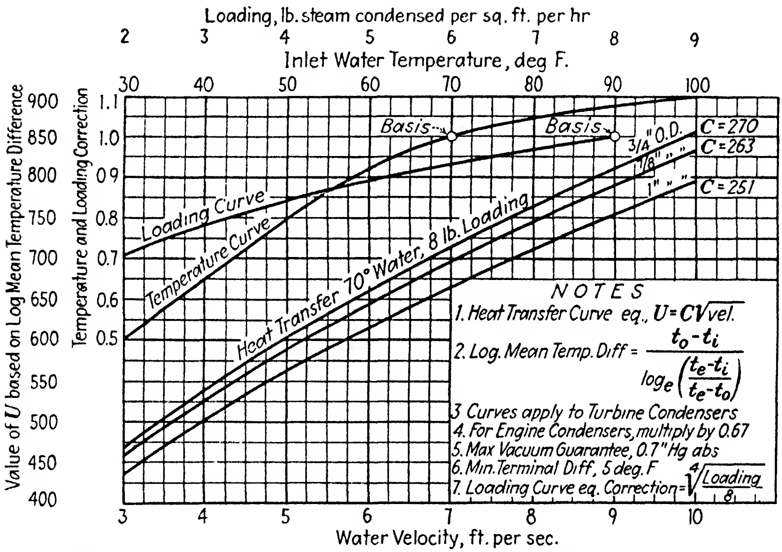


FIG. 178.—Heat-transfer curves for surface-condenser design calculations.

produces a suction in the air-suction chamber 1 which carries the air into the jet condenser 33. Here the first stage is condensed, and the air is removed by means of the second stage ejector at the right, the volume of the air being reduced by the cooling in the intercondenser. The condensing-water outlet may be connected to a long tail pipe to form a small barometric condenser.

In 1934 a number of American condenser manufacturers cooperated in the effort to establish a reasonable basis for design in view of the many variables involved. The curves of Fig. 178 represent such a basis. Studies of condenser economics place the proper water velocity through the tubes between 6 and 7 ft per sec. The use of this chart is illustrated by the following example.

Example A.—What heat-transfer rate U could be expected with 1-in. tubes to condense 7 lb steam per square foot per hour with 60°F. inlet cooling-water temperature at 6.5 ft per sec through the tubes.

SOLUTION: From diagram, Fig. 178:

$$U \text{ at } 70° \text{ and 8-lb loading} = 640$$
$$\text{Temperature correction at } 60 = 0.92$$
$$\text{Loading correction at 7 lb} = 0.965$$
$$\text{Then corrected value of } U \text{ to be expected} = \frac{640}{0.92 \times 0.965} = 721$$

It is interesting that, owing to improved arrangement of tube nests, air removal, and better tube cleanliness, design-transfer

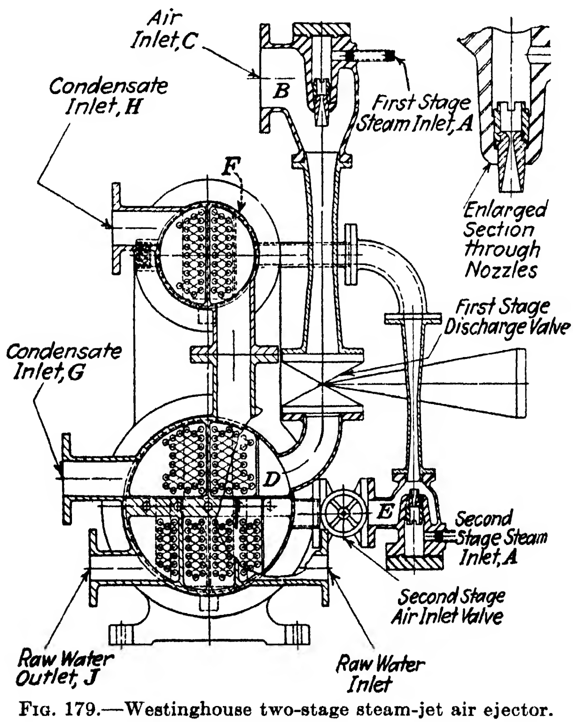


FIG. 179.—Westinghouse two-stage steam-jet air ejector.

rates have risen steadily from 400 in 1922 to about 660 in 1934 at a velocity of 6 ft per sec. Values as high as 1,200 are attainable for a single tube under laboratory conditions which are not subject to the limitations of condenser practice.

Figure 179 illustrates a two-stage ejector with surface inter- and aftercondensers. *A* is the steam-supply pipe, *B* the first stage ejector, *C* the air connection to the condenser, *D* the intercondenser, *E* the second stage ejector, *F* the aftercondenser, and *G* and *H* the cooling-water inlet and outlet. *J* is a

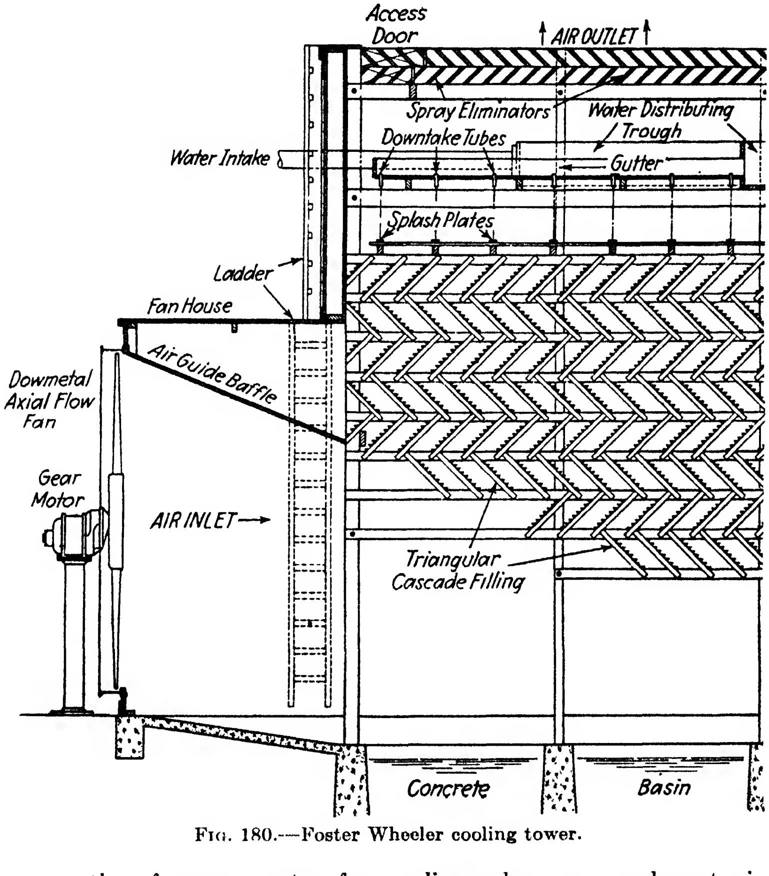


FIG. 180.—Foster Wheeler cooling tower.

connection for raw water for cooling when no condensate is available.

171. Recooling Condensing Water.—In localities where condensing water is scarce or expensive, some means must be provided for recooling the discharge from the condensers for further use. If sufficient area for an open pond is available, the water may be cooled by evaporation from the surface. If the space for a pond is limited, the discharge from the condensers may be carried out over the pond and sprayed into the air from many spray heads, the evaporation from the spray cooling the remainder which falls back into the pond.

Plants that are in still more crowded quarters may use cooling towers. The water is pumped to the top of these structures and then allowed to fall to the bottom through the air, encountering obstructions on the way which break the streams into fine sprays. An upward current of air produced either by natural draft or by a blower evaporates part of the water and cools the remainder. Figure 180 shows such a cooling tower for an industrial power plant. Cooling towers have found wide application for numerous cooling purposes other than for condenser water.

172. Selection of Condenser.—The proper choice of the type of condenser depends upon (1) the cost of the condenser and auxiliary equipment; (2) the quality and abundance of the water supplies for boiler feed and for condenser cooling; (3) the cost of power for operating the condenser pumps.

If the water for boiler feed is bad, it will be advisable to use a surface condenser in order to save the condensate for the boiler. But, if the water supply for cooling is, also, bad, that is, corrosive or scale forming, it will be advisable to use a jet condenser, even at the expense of treating the raw water for boiler feed, because the replacement of tubes in a surface condenser would be frequent and expensive.

It should be pointed out that most condenser manufacturers build both the jet and surface types. Figure 181 shows a 35,000-kw, twin-jet condenser built by the Elliott Company for the Narragansett Electric Lighting Company. This condenser uses salt water for injection. Figure 182 shows a modern welded-shell surface condenser. This condenser is of the single-pass type, the cylindrical sections at the two ends being the water boxes, the water passing from one box to the other but once. Single-pass condensers are used when an abundance of cooling water is available or the pumping head is low. Two-pass condensers are used when water is scarce or the pumping head is high.

The selection of the size of a condenser for a particular service is a problem in economic selection. The improvement in tur-

FIG. 181.—Elliott 35,000-kw twin-jet condenser.

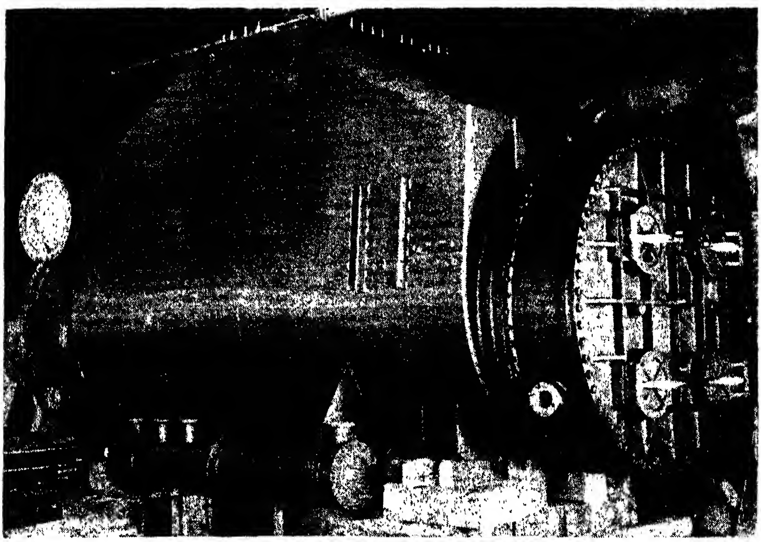


FIG. 182.—Foster Wheeler 13,000-sq. ft. welded-shell surface condenser.

bine performance is very great for each small increment in reduction of the last inch or two of back pressure. Therefore the designer must balance the dollar value of such improvement against the increased investment charges and the increased pumping costs for each proposed increase in condenser surface required to obtain better vacuum.[1]

Problems

1. A 150-hp steam engine uses 31 lb steam per indicated horsepower-hour. The steam is supplied at 125 lb gage and 98 per cent quality, and the exhaust is at 26-in. vacuum with a 29.5-in. barometer. The condenser is of the jet type and receives injection water at 70°F. Assuming no loss due to radiation, calculate (*a*) the ratio of injection water to steam condensed; (*b*) gallons of injection water per hour with the engine at full load.

2. A 100-kw turbogenerator running at full load uses each hour 2,670 lb steam at a pressure of 150 lb gage and 50° superheat, exhausting into a jet condenser at a vacuum of 27 in. with a 29.7-in. barometer. Injection-water temperature 48°F. Neglecting radiation and friction, calculate (*a*) the quality of steam entering the condenser; (*b*) the number of gallons of injection water required per hour.

3. An engine uses 17 lb steam per indicated horsepower-hour. Initial steam condition 140 lb per sq in. abs and 40° superheat. Absolute condenser pressure 4 in. Hg. Injection-water temperature 70°F. Calculate (*a*) the condition of the steam entering the condenser; (*b*) the injection-water-to-steam ratio. Assume loss due to radiation to be 1 per cent of the initial heat supplied.

4. A steam turbine uses 12 lb steam per kilowatt-hour. Assume mechanical and electrical losses to be 5 per cent. Steam condition 165 lb per sq in gage and 200° superheat. Vacuum 28.7 in.; barometer 29.8 in. Jet condenser using injection water at 50°F. Calculate (*a*) the condition of the steam entering the condenser; (*b*) the injection-water-to-steam ratio.

5. A turbogenerator uses 15 lb steam per kilowatt-hour. Generator efficiency is 91 per cent. Assume radiation and turbine-friction loss to be 0.5 per cent of the initial heat supplied. Steam condition 175 lb per sq in. abs and 210° superheat. Vacuum 28.5 in., barometer 30 in. Injection-water temperature is 47°F., and t_o is 4° lower than t_e. Calculate (*a*) the condition of the steam entering the condenser; (*b*) the injection-water-to-steam ratio.

6. A steam turbine using 18 lb steam per shaft horsepower-hour exhausts at 1.5 in. abs into a surface condenser. The initial condition of the steam is 165 lb per sq in. abs and 100° superheat. The circulating water has entering and leaving temperatures of 45 and 77°F., respectively. Neglecting radiation losses, calculate (*a*) the condition of the steam entering the condenser; (*b*) the weight of cooling water per pound of condensate.

[1] *Power*, November, 1934, p. 626.

7. A 500-ihp steam engine uses 16 lb steam per indicated horsepower-hour. Initial steam condition, 125 lb; gage and quality, 99 per cent; vacuum, 26 in.; barometer, 30 in. Cooling-water temperatures: entering, 70°F.; leaving, 106°F. Surface condenser, condensate temperature 15° below exhaust steam temperature. Neglecting radiation loss, calculate (*a*) the enthalpy of the exhaust steam; (*b*) the number of gallons of cooling water required per hour.

8. During a test of a 20,000-kw turboalternator, the following data were taken: kilowatts load, 25,393; throttle pressure, 272 lb abs; throttle temperature, 593°F.; vacuum, 28 in.; barometer, 29.8 in.; total steam condensed per hour, 272,100; entering temperature of cooling water, 58.9°F.; leaving temperature of cooling water, 74.3°F.; condensate temperature, 89.5°F. The generator efficiency at this load is 96.5 per cent. Neglect radiation. Calculate (*a*) the enthalpy of the steam entering the condenser; (*b*) the number of gallons of cooling water used per hour.

9. Calculate the logarithmic and arithmetic mean-temperature differences for the conditions of Prob 8.

10. What should be the number of square feet of tube surface in a condenser to serve a 1,000-kw turbine? Assume the heat-transfer rate to be 500 Btu per sq ft per hr per degree logarithmic mean-temperature difference; vacuum, 28 in.; barometer, 30 in.; steam rate, 16 lb per kw-hr; enthalpy of exhaust, 1000 Btu per lb; cooling-water entering temperature, 75°F., and outlet temperature, 20° below exhaust steam temperature.

11. A 30,000-kw turbogenerator has a guaranteed steam rate of 10.1 lb per kw-hr and a generator efficiency of 95 per cent. Initial steam condition, 250 lb per sq in. abs, and 200° of superheat; vacuum, 29 in.; barometer, 30 in.; initial cooling-water temperature, 70°. Assume outlet temperature 10° below t_c. Using 450 Btu per sq ft per hr per degree logarithmic mean-temperature difference, calculate the number of square feet of condenser surface necessary.

12. A 27,000-sq ft, single-pass condenser has 1-in. O.D. tubes 19 ft long of 18-gage thickness and is guaranteed to condense 173,000 lb steam per hour, with 20,000 gal cooling water per minute at 55°F. and maintain an absolute pressure at the condenser inlet of 1 in. Hg, the steam temperature to be not more than 1°F. above the hot-well temperature. Assuming t_o to be 8° lower than t_c, calculate (*a*) the logarithmic and arithmetic mean-temperature differences; (*b*) the number of tubes; (*c*) the velocity of flow through the tubes in feet per second; (*d*) the rate of heat transfer.

13. Draw the logarithmic curve of temperature rise for Prob. 12.

14. A test of a 50-000-sq ft condenser gave the following data: steam condensed per hour, 300,000 lb; vacuum, 29.37 in.; barometer, 30.38 in.; inlet-water temperature, 50.0°F.; outlet-water temperature, 58.0°F.; cooling-water flow, 71,500 gal per min; pounds cooling water per pound of steam, 119. Calculate (*a*) the logarithmic mean-temperature difference; (*b*) the heat-transfer rate; (*c*) enthalpy per pound of steam condensed.

CHAPTER XVII

STEAM BOILERS

173. The Steam Boiler.—In the original application of steam for power purposes, the boiler and engine were one piece of apparatus, but it early became evident that, in order to give flexibility to the plant, it would be better to have a separate steam generator in which heat could be stored, and this led to the development of what is called the "steam boiler." At first the boiler was simply a tank in which water was evaporated at about atmospheric pressure, and there was therefore little danger of explosion. As the demand for higher pressures and quantities of steam developed, it became necessary to provide numerous devices and details that would make the operation of a boiler both safe and economical. The vital details have become well standardized and, in many instances, are required by state and federal laws.

Boilers are commonly classified as *fire tube*, in which the hot gases pass through tubes that are surrounded by water, and *water tube*, in which the water passes through tubes that are surrounded by hot gases.

174. An Elementary Boiler.—In Fig. 183 is shown one of the simplest commercial boilers having the essential features of all types of boilers. These features are listed here for reference and may be identified in the illustrations of the designs that are described later in the chapter.

a. A closed vessel called the "shell" capable of safely withstanding the internal pressure for which the boiler is designed.

b. A gage for indicating the difference in pressure between the steam within the boiler and the atmosphere without.

c. The vessel is only partially filled with water, thus providing space for steam.

d. A pipe, or nozzle, for removal of the liberated steam.

e. A gage glass for showing the height of the water surface in the boiler.

f. A feed pipe for introducing the water into the boiler to replace that which has been evaporated and removed as steam, in order to keep the water level constant.

g. A drain or blowoff for removing sediment that is carried in with the water in suspension or precipitated by the heat within the boiler.

h. A proper amount of disengagement surface where the steam may leave the water. This must be large enough to prevent the bubbles of steam rising to the surface so rapidly that they carry water particles along with them into the steam outlet, thus producing what is called "priming."

i. Provision for circulation of the water. Since water at rest is a very poor conductor of heat, ample opportunity must be given it to flow over the heating surface.

j. A furnace for combustion of fuel.

k. Provision for passing the hot gases of combustion over the heating surface in a devious path by means of baffles or walls. "Heating surface"

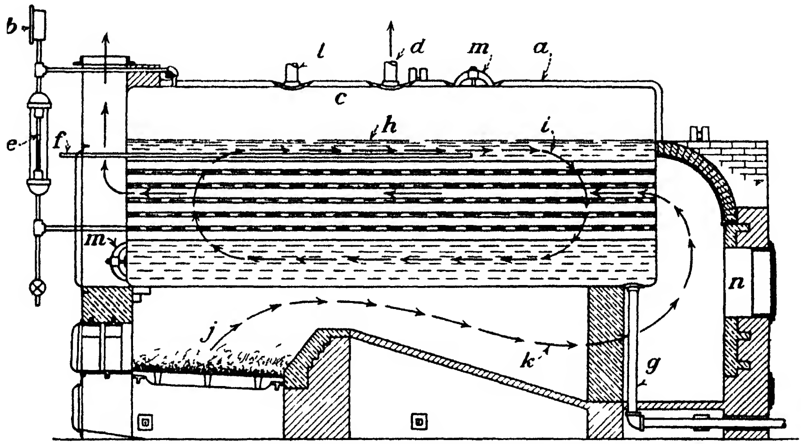


Fig. 183. Horizontal return tubular (fire-tube) boiler.

is boiler surface which is exposed to the radiant flame or the hot gases of combustion on one side and water on the other side, the area being measured on the gas side. Surface with hot gases on one side and *steam* on the other is called "superheating surface." In modern boilers great effort is made to have a portion as large as possible of the heating surface exposed to the radiant energy of the fire, as this is by far the most effective in absorbing heat. Figure 183 has no superheater.

l. A safety valve.

m. Provision for opening the boiler for inspection or cleaning either by manholes or handholes or both.

n. Provision for cleaning the gas side of soot.

o. A method of supporting the boiler that will allow free expansion (none shown in Fig. 183).

This is probably the most common form of commercial boiler, considering numbers in use, and, although one of the earliest, is still very widely used because of its simplicity, cheapness, and reliability. The boiler consists of a cylindrical shell with a large number of *fire tubes* running from end to end, through which the hot furnace gases pass from the rear to the front and then up the chimney. The chimney is always at the front of an H.R.T. boiler, the boiler of Fig. 183 having what is called a "flush front," the smoke box being riveted to the boiler shell. The shell is supported by brackets riveted to it near the front and rear. The front brackets are fixed to a steel framework, while the rear ones are suspended on hangers so as to allow the boiler to expand and contract without cracking the brickwork.

The grates are under the front third of the shell, and just to the rear of the grates is a wall called the *bridge wall,* extending up close to the shell in order to force the hot gases against the shell. The space to the rear of the bridge wall is the "combustion chamber." At the rear of the shell is a space between the wall and the end of the shell to allow the hot gases to pass upward and into the tubes. The brick column under the rear of the shell protects the blowoff pipe from the hot gases.

Fire-tube boilers, while having a definite field of usefulness in small plants, for moderate pressures, are faced with three limitations: (1) Since the thickness of the shell for a given internal pressure varies directly as the diameter, the size and therefore the capacity is limited to relatively small plants; (2) practical pressures are usually not over 250 lb; (3) they cannot be forced very well beyond 150 per cent of rating. The heating surface varies with the diameter and the volume as the square of the diameter, which means that in the large sizes the quantity of water contained in the shell per unit of surface becomes excessive. For safety and quick response the quantity of water should be kept as small as practicable.

Since the thickness of a boiler shell varies directly as the diameter of the shell for a given pressure, the diameter should be kept as small as possible. In the commercial types that are described, therefore, many different schemes have been resorted to in order to increase the heating surface and decrease the weight of boiler and of water contained and, also, to decrease the danger from boiler explosions.

175. A.S.M.E. Power Boiler Code.

The A.S.M.E. Boiler Construction Code contains rules formulated by the Boiler Code Committee, and approved by the Council of the Society, to cover boilers and unfired pressure vessels. Specifications for materials used in such constructions have been adopted and included. The primary object of these rules is safety.

The Boiler Code Committee does not approve, recommend, or endorse proprietary or specific designs, nor does it assume to limit in any way the builders' right to choose any method of design, or form of construction that conforms to the code rules.

The committee meets at regular intervals for the purpose of considering inquiries relative to the Boiler Code.

Each state and municipality that has accepted one or more of the sections of the Boiler Code is invited to appoint a representative to act on the Conference Committee to the Boiler Code Committee.

It should be pointed out that the state or municipality where the Boiler Code has been made effective has definite jurisdiction over any particular installation.[1]

Any manufacturer who complies with the provisions of the code is granted permission to stamp the boiler or part manufactured by him with the official boiler-code stamp.

The construction code should not be confused with the A.S.-M.E. Boiler Test Code, see Art. 228, which prescribes rules for conducting performance trials of steam boilers.

176. Other Fire-tube Boilers.—A number of smaller forms of fire-tube boilers have found favor for portable power installations and for building heating. Three such portable boilers having internal fireboxes are shown here.

Figure 184 shows a *Scotch Marine* type in which the furnace is inclosed in a large corrugated flue which runs through the shell. The hot gases pass over the low bridge wall to the brick-lined chamber in the rear, and then forward through the fire tubes to the smoke box at the front. The horizontal pipe at the steam outlet is a "dry pipe." The large rods above the tubes are through stays for the flat ends of the shell. Originally used at sea this totally self-contained unit has been very useful in the land form.

[1] Quotations from Foreword of 1937 edition of A.S.M.E. Power Boiler Code.

Figure 185 shows a portable locomotive-type boiler. The firebox is the rectangular space at the front third of the boiler and is entirely surrounded by "water legs." The tubes extend from

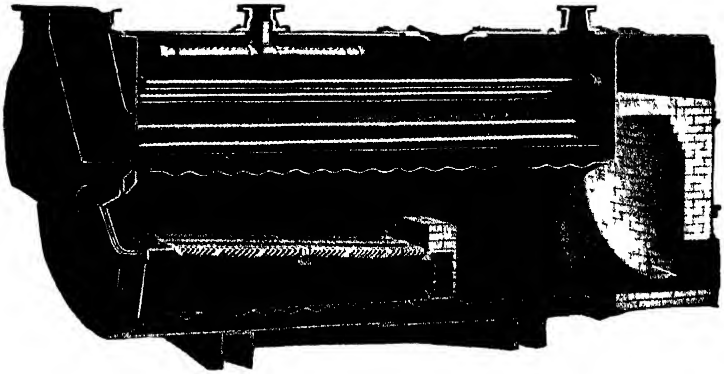


FIG. 184.—Portable Scotch Marine boiler, dry-back type

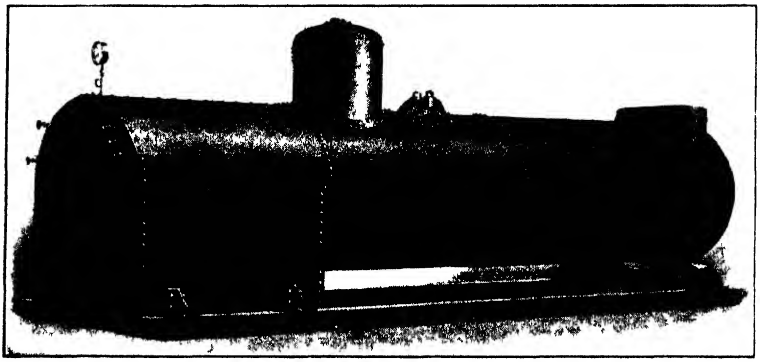


FIG. 185.—Kewanee portable, locomotive-type boiler

the firebox through the cylindrical shell to the smoke box This boiler has a steam dome at the top which serves as a steam drier.

Figure 186 shows one form of the vertical, internally fired boiler. The firebox is at the bottom surrounded by a water leg, the hot gases passing up through the vertical tubes to the smoke box at the top. The boiler shown has submerged tubes, that is, the tubes are surrounded by water to their tops which protects them against the hot gases. The steaming capacity of this boiler is very low due to the small steam space.

177. Water-tube Boilers.—Water-tube boilers have been built in a great many different forms, but those which have been commercially successful have been developed along a few easily classifiable lines. We might classify water-tube boilers on the basis of the shape of the tubes into (*a*) straight-tube and (*b*) curved-tube types. A typical example of each of these is given in the next two articles.

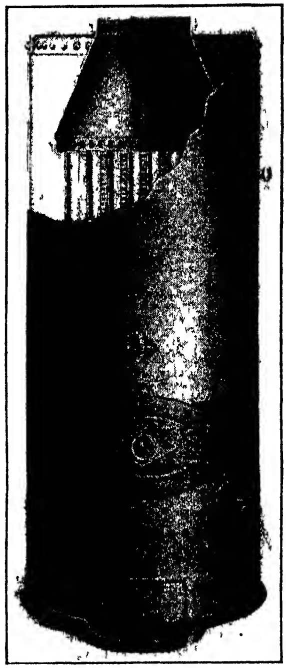


FIG. 186.—Kewanee vertical fire-tube boiler, submerged-tube type.

A Straight-tube, Water-tube Boiler.—Figure 187 shows a straight-tube boiler consisting of two decks of tubes connected to headers at the ends, which in turn are connected by tubes to a drum across the boiler over the rear header. The lower deck receives largely radiant energy from the fire while the upper deck, not in sight of the flame, receives heat by convection. The tubes are "staggered" in a vertical plane to force the gases to follow a sinuous path between them.

The water circulation is up the 20° incline of the tubes to the front header, then vertically up this header to the tubes running back to the front of the drum where the steam is released. The water then flows from the bottom of the drum, through the vertical connecting tubes to the rear header. The drum, being only about half full of water, has a large steam space and disengagement surface.

The path of circulation of the hot gases is up across the lower tube deck, across the front half of the upper deck, in front of the "baffle" rising from the top and rear of the lower deck, downward across the middle section between the baffles, then upward toward the bottom of the drum, and out through the economizer. This is a three-pass, cross-baffled boiler. In the high-temperature region the baffles are of refractory tile, while for the cooler parts metal baffles suffice.

The inside surfaces of the tubes are cleaned by removing handhole covers in the headers opposite the ends of each tube and

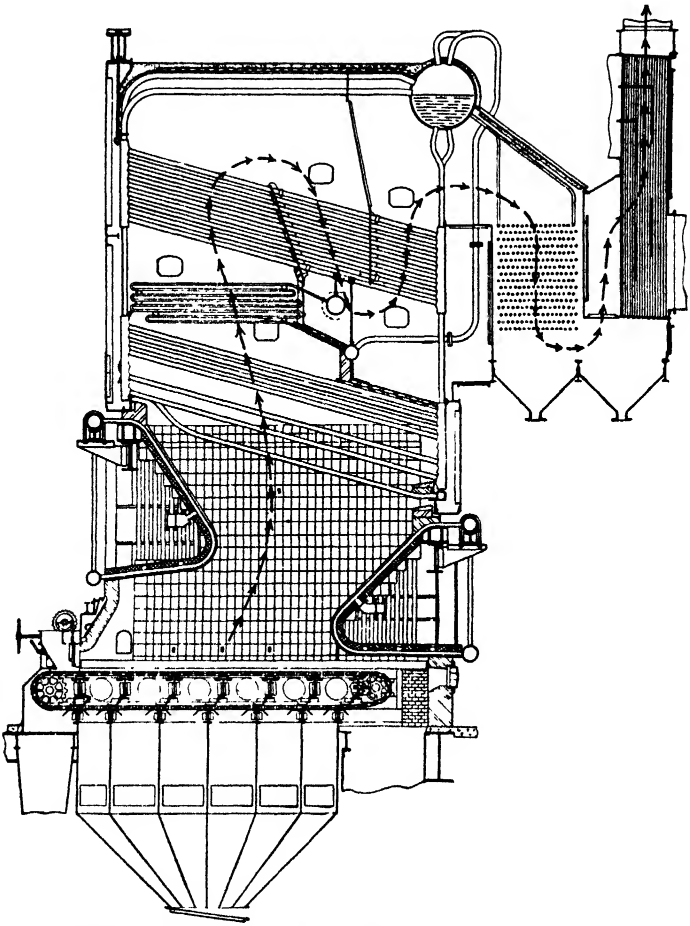


FIG. 187.—Babcock and Wilcox straight-tube boiler with superheater water walls, economizer, and air heater.

running a tube cleaner through them. Access to the inside of the drum is provided by manholes at the ends.

Water is fed to the drum where it is warmed by the hot water in the drum before it enters the tubes. The blowoff is from the square box at the bottom of the lower rear header.

A Curved-tube, Water-tube Boiler.—A second type of water-tube boiler, one using curved tubes, is shown in Fig. 188. This boiler consists of three horizontal cross drums at the top sup-

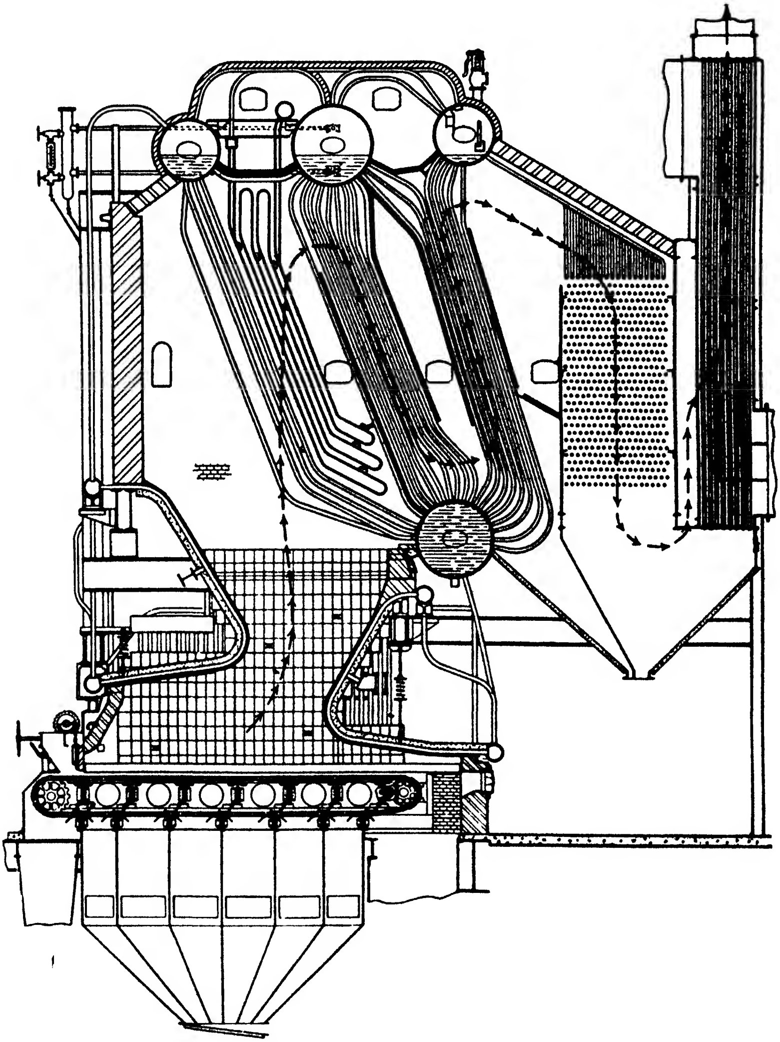


Fig. 188.—Babcock and Wilcox Stirling curved-tube boiler.

ported by steel beams under the ends just outside the setting. From the upper drums three banks of tubes run downward to a fourth drum, called the "mud drum." Short downward-curved tubes connect the two front drums below the water line, which is near the middle of the drums, and other short upward-curved tubes connect the steam spaces of all three drums.

The water circulation is upward in the first bank of tubes to the front drum, across to the middle drum, and then downward in the middle bank to the mud drum. The feed water enters the small curved pipe in the rear drum and empties into a V-shaped trough riveted to the side of the drum. Feed water flows down the last bank of tubes to the mud drum and then joins the circulation up the front bank, across to the second drum, and down

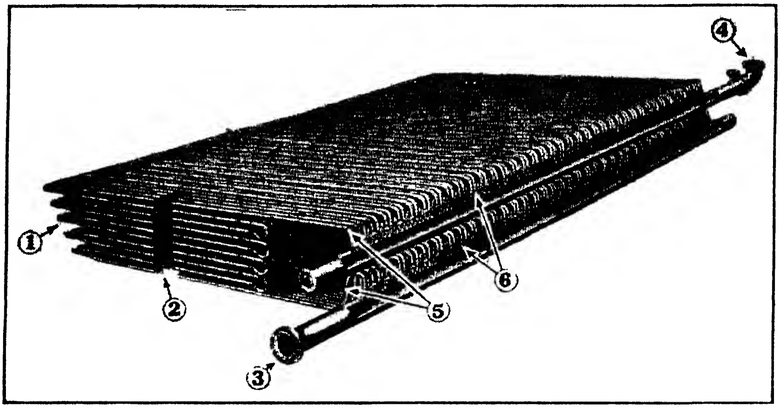


FIG. 189. - Elesco convection superheater. (*The Superheater Co.*)

again. Sediment is deposited in the mud drum and discharged through the blowoff at the bottom. The steam is practically all liberated in the front drum but is taken off from the rear drum, thus giving it time to be freed from any entrained moisture.

The path of circulation of the hot gases is upward in front of the baffle resting on the back of the first tubes of the second bank, then downward in the second pass, and lastly upward in the third pass, the gases in the main flowing lengthwise of the tubes. The wider spacing in the first bank allows better absorption of the radiant heat of the furnace.

The tubes are cleaned inside by a man entering the upper drums through the manholes at the ends and running a flexible cleaning tool down through the tubes. While straight tubes such as those in the B. and W. boiler have the advantages of simplicity and easy inspection, curved tubes provide opportunity for each one to expand independently of the others.

These two water-tube boilers have an additional U-shaped bank of tubes added for the purpose of superheating the steam.

Superheaters consist of a series of tubular elements through which the steam flows after leaving the boiler drum, the superheaters being placed so as to receive heat from the furnace gases. There are two general types of superheaters, namely, *convection* superheaters, which are placed in one of the boiler passes so that

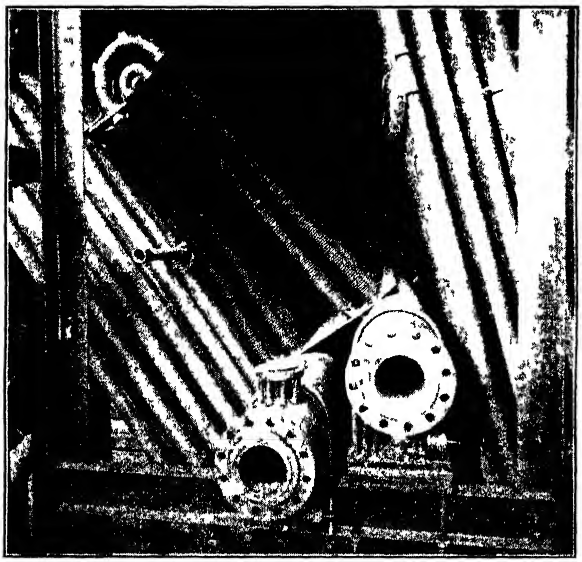


FIG. 190. Foster double-loop superheater.

the hot gases pass over them, and *radiant* superheaters, which are placed so that the radiant heat from the furnace will strike them.

Figure 189 shows an Elesco superheater of the convection type, which consists of units of heavy-gage, seamless-steel tubing of small diameter. These units are connected in parallel to the headers by metal-to-metal ball joints, secured by means of clamps and studs. Steam enters the superheater from the bottom header and leaves through the opposite end of the top header, thus making the pressure drop through all units the same. Several of the boilers illustrated in this chapter show Elesco superheaters installed.

Figure 190 shows a Foster double-loop superheater of the convection type installed in a Connelly boiler. This superheater consists of larger tubes than those in the Elesco heater, and the heat-absorbing surface is increased by means of cast-iron fins placed over the outside. Figure 191 shows the construction of the Foster tubes.

In Fig. 187 the steam leaving the top of the drum is carried to the bottom header of the superheater, through the tubes and out the top header. In Figure 188 the superheater is in the second pass of the boiler. These are both convection superheaters.

178. Improvements to the Simple Boilers.—Since the beginning of the water-tube boiler, continuous efforts have been made

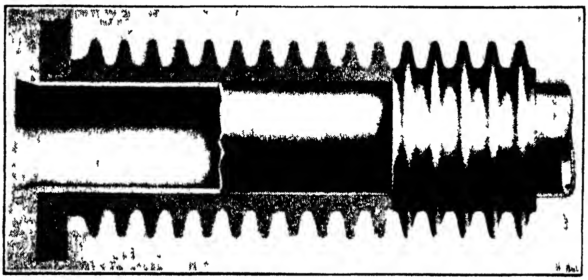


Fig. 191.—Heating surface of Foster superheater

to improve its performance and reduce its cost. These efforts have been directed chiefly toward four points.

Circulation.—As steam is generated in the tubes, it is important that this steam find its way as rapidly as possible out of the tube to the disengagement surface in order to keep the heating surface working effectively. This has accounted largely for the many arrangements of tubes and drums. In some cases it has been found that during heavy firing some tubes are only partly filled with water, and other tubes have a flow opposite to that intended by the designer; this, of course, reduces the effectiveness of the heating surface.

Baffling.—High rates of heat transfer from fire to water involve leading the hot gases through the intertube spaces so that all parts of the heating surface may be used most effectively. Baffles are used to force the gas stream where it is needed and, in general, are so placed as to force the gases across the tubes several times at high velocity rather than lengthwise. The gas velocity and number of passes are limited by the draft loss set up.

Dry Steam.—Steam leaving the boiler drum should be free from moisture whether it is to be superheated or not, see Art. 223. A great deal of thought and experimentation has been devoted to placing of the steam drums and interconnecting them so as to have dry steam leaving the drum.

Cleaning.—A very thin layer of scale on the water side or of soot on the fire side of boiler tubes is capable of causing a substantial loss in efficiency. Therefore tubes and baffles must be so arranged as to be easily cleaned inside and outside.

The hot gases leaving the boilers so far illustrated may have a high temperature, especially at heavy loads, because the temperature of the water and steam in the boiler is in the neighborhood of 400 or 500° and a temperature difference between gases and tubes is necessary for heat transfer. Two types of apparatus are used to reduce the temperature of the gases. These are economizers and air preheaters.

Economizers.—Economizers are closed heaters consisting of cast-iron or steel tubes placed in the gas passages between the boiler and the chimney, and through these tubes the feed passes on the way to the boiler. The economizer might, therefore, be considered a part of the boiler-heating surface, and some boilers are built with *integral economizers,* that is, a part of the boiler heating surface in the last pass is used for preheating the feed. For best economy economizers are arranged *counterflow,* that is, with the hottest gases in contact with the hottest tubes and the coolest gases in contact with the coldest tubes, the water flowing through the tubes *counter* to the passage of the gases on the outside.

Figure 192 shows a steel-tube economizer arranged with horizontal finned tubes, the water entering the bottom and leaving at the top. The hot gases are drawn to the top of the economizer through the duct shown in phantom at the left and then down through the economizer by means of the fan at the extreme right. The economizer is equipped with a soot blower, one of the problems encountered in economizers being due to the accumulation of soot and water that condenses from the combustion gases onto cool tubes. The cinders and soot are cleaned from the hopper bottom.

Air Heaters.—Another type of apparatus for reducing the stack losses is the air preheater or air economizer, which may be installed independently or in series with water economizers These preheat the air supply to the furnace and thus increase to this extent the temperatures obtained in the furnace. Air heaters are built in three types—plate, tube, and regenerative

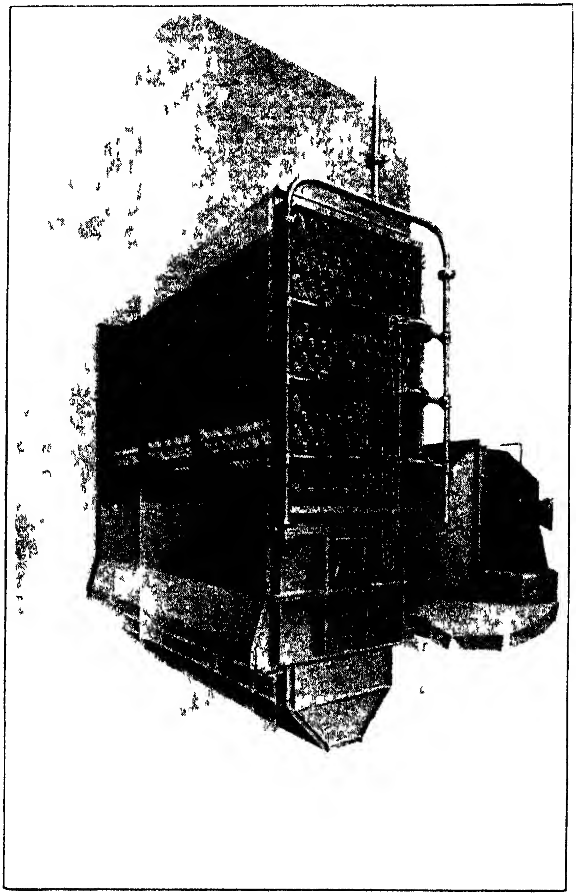


FIG 192 — Foster Wheeler economizer

Figure 193 shows a Foster air heater of the plate type, consisting of four interchangeable sections placed one above the other The hot gases enter at the top and pass down vertically through the heater between the plates forming the sides of the air passages The air enters the near side of the lower section, passes across through this section, up the connecting duct at the back to the second section, and follows a zigzag path through the entire heater, leaving on the near side at the top. This arrangement makes a counterflow heat-transfer apparatus. This heater is equipped with a soot blower.

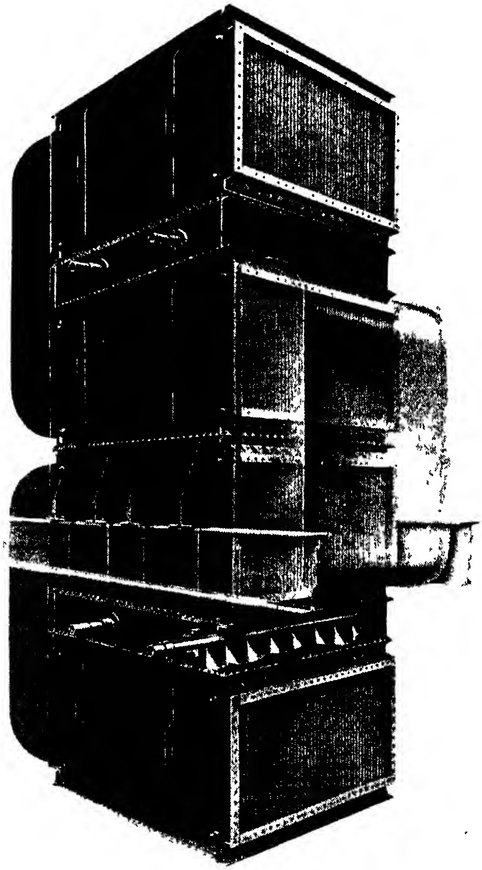


FIG. 193.—Foster Wheeler plate-type sectional air heater.

In the tube type the general arrangement is to have the gases drawn through steel tubes, perhaps 3 in. in diameter, across which the combustion air passes in counterflow fashion on its way to the furnace. The boiler of Fig. 201 is equipped with this type of heater but has the air flow through the tubes.

Figure 194 shows a regenerative-type air heater, in which a rotor, driven at about 3 rpm by a motor, is caused to pass momentarily through the stream of hot gases and then through the stream of combustion air. The rotor consists of a large number of thin steel plates, corrugated so as to increase their surface. The plates are alternately heated by the hot gases and cooled by the air, thus transferring the heat from gases to air.

Tests on air heaters have reported added efficiencies of 3 to 5 per cent. The limiting factor in the amount of air heating seems

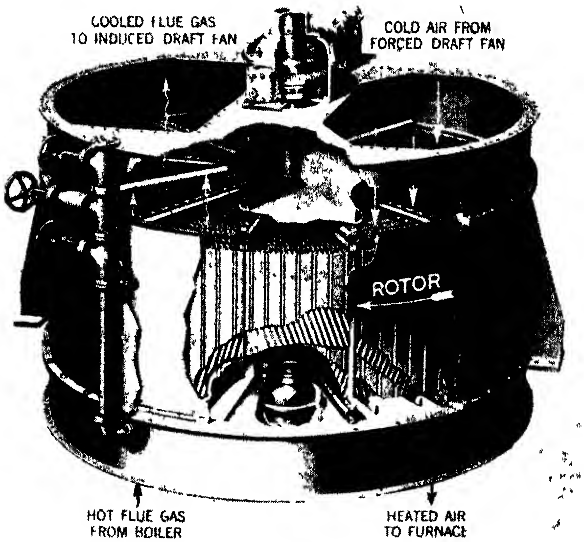


FIG 194.—Ljungstrom regenerative air preheater.

to be the higher furnace temperatures resulting from higher temperature of the combustion-air supply, since high furnace temperatures tend to increase brickwork maintenance.

In the early years of the development of steam boilers when hand firing was the rule, rates of combustion were low enough to permit of the use of refractory brick linings in the furnaces. However, with the coming of mechanical stokers, pulverized coal, and oil- and gas-fired boilers, frequently with air preheating, the accompanying high furnace temperatures caused trouble with the refractories. This resulted in the development of air-cooled and water-cooled furnace walls; the most usual form has the furnace entirely lined with water tubes, having extended surfaces welded or bolted to the tubes. Figure 191 shows the furnace side of a typical water wall of the welded-fin construction which presents an all-metal surface to the fire. This is backed by refractory tile and insulating brick, the headers at top and bottom being outside the furnace and connected to the water-circulation system. Water-tube screens are also used in some installations at the bottom of the furnace, the tubes being spaced far enough apart to allow the ash to fall to the ash hopper below. In other installations the hopper-shaped bottom is lined with water tubes.

A recent development in furnace bottoms is the *slag-tap* floor which is used for pulverized firing of coal having a low-melting-point ash. The molten ash and added flux form a pool on the floor which is tapped off either continuously or periodically. The floor may consist of refractory tile or water wall covered with smooth cast-iron blocks. The boiler of Fig. 202 has a water-cooled, slag-tap floor.

179. American Types of Water-tube Boilers.—As a consequence of the efforts of designers to produce more economical steam boilers, many different types of water-tube boilers have appeared on the market, each having some points of advantage. These designs may, for convenience, be classified as below, and shown diagrammatically in Figs. 195 and 196:

A. Straight Tubes.
- 1. Longitudinal drums.
  - *a*. Drums and tubes parallel, both inclined, (*a*), Fig. 195.
  - *b*. Drums horizontal, tubes inclined, (*b*), Fig. 195.
- 2. Cross drums, inclined tubes, (*c*), Fig. 195.
- 3. Vertical drums and tubes, (*d*), Fig. 195.
- 4. Semivertical, (*e*), Fig. 195.

B. Curved tubes, cross drums.
- 1. Two drum, (*f*), Fig. 195.
- 2. Three drum, (*a*), (*b*), (*c*), Fig. 196.
- 3. Four drum, (*d*), (*e*), Fig. 196.
- 4. Multidrum, (*f*), Fig. 196.

180. Steam-generating Units.—The modern steam-generating unit is more than a boiler. It is a coordinated and balanced assembly of boiler proper, superheater, economizer, air heater, furnace, fuel-burning equipment and setting. The correct proportioning of the various elements becomes an intricate problem in which theory and experience find expression in many and various designs. As would be expected, the boiler proper, which consists largely of convection surface, has become a

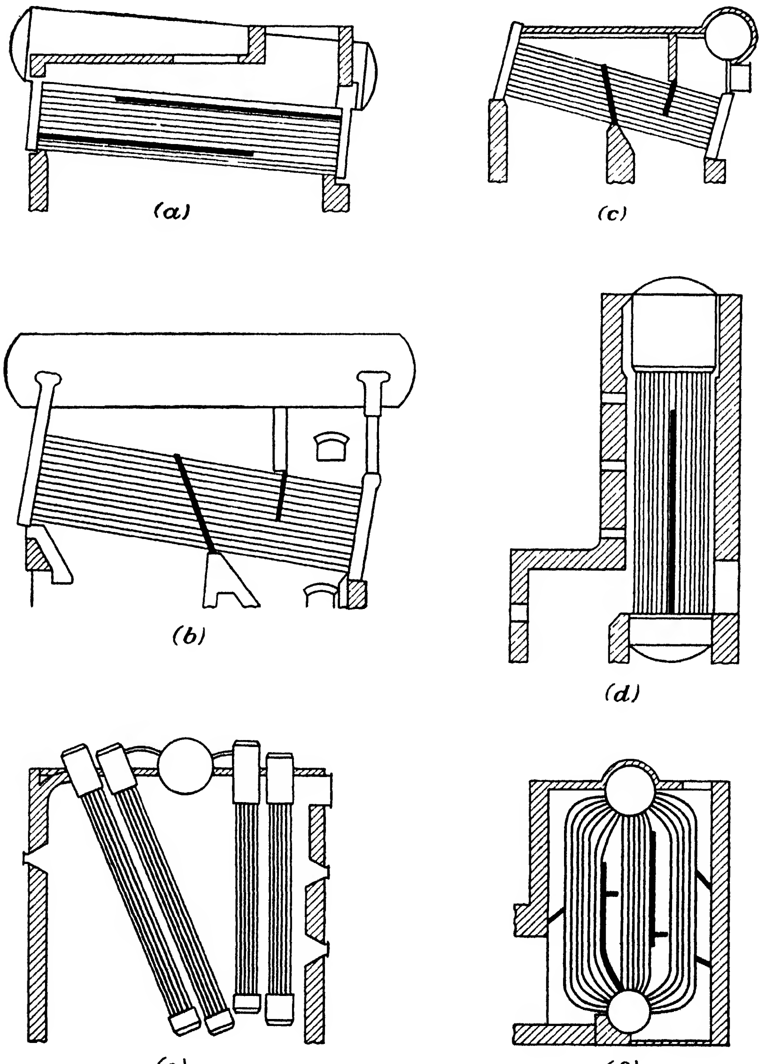


FIG. 195.—Water-tube boiler types.

smaller part of the unit, while the radiant surface has been increased. In all cases, however, the complete steam-generating unit is designed as a whole with the result that boilers of very high capacities and efficiencies above 85 per cent are common.

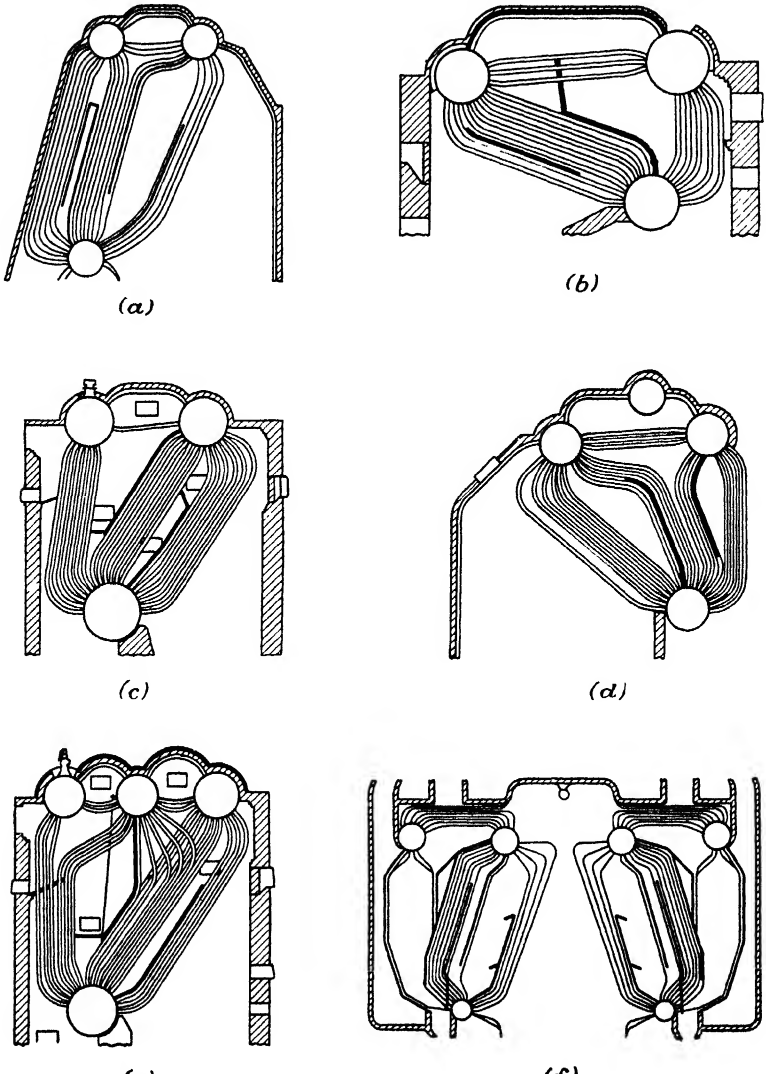


FIG. 196.—Water-tube boiler types.

Figure 197 shows a perspective section of a modern self-contained steam generator which is built in sizes up to 20,000 lb steam per hour The features of this unit are (1) complete factory assembly in the smaller sizes, (2) water-cooled furnace

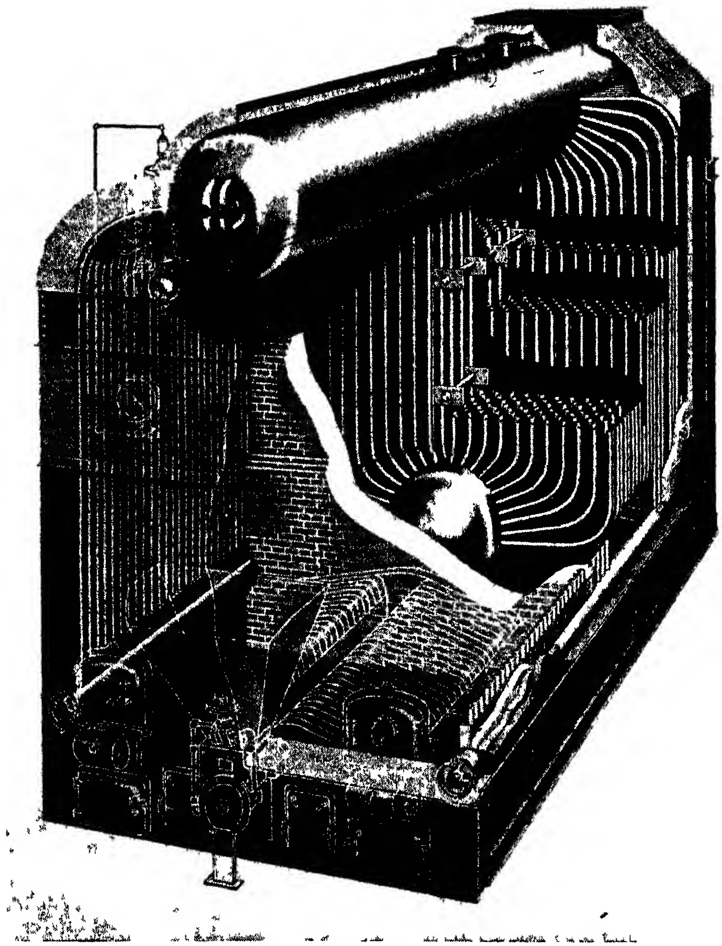


Fig 197 Keeler steam generator

walls and ceiling, (3) no brickwork except bridge wall and front wall, (4) entire unit insulated and steel encased, (5) simplicity

Figure 198 illustrates a standardized unit of larger size, 20,000 to 250,000 lb steam per hour. This consists of the boiler proper which is of the two-cross-drum type, convection superheater, water walls, and top and bottom furnace-screen tubes which completely inclose the furnace, all surrounded by a relatively light insulation and steel casing. Baffling is arranged for three-pass gas flow parallel to the tubes.

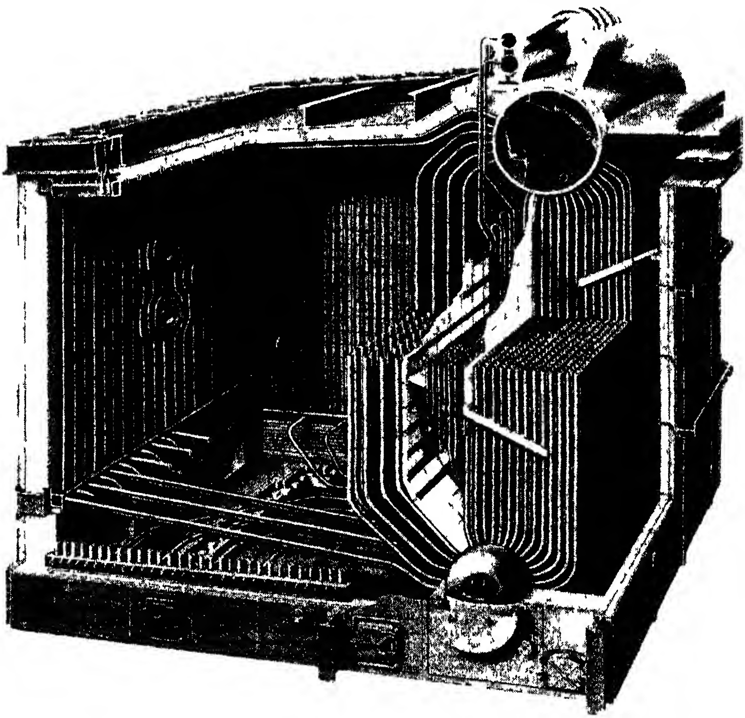
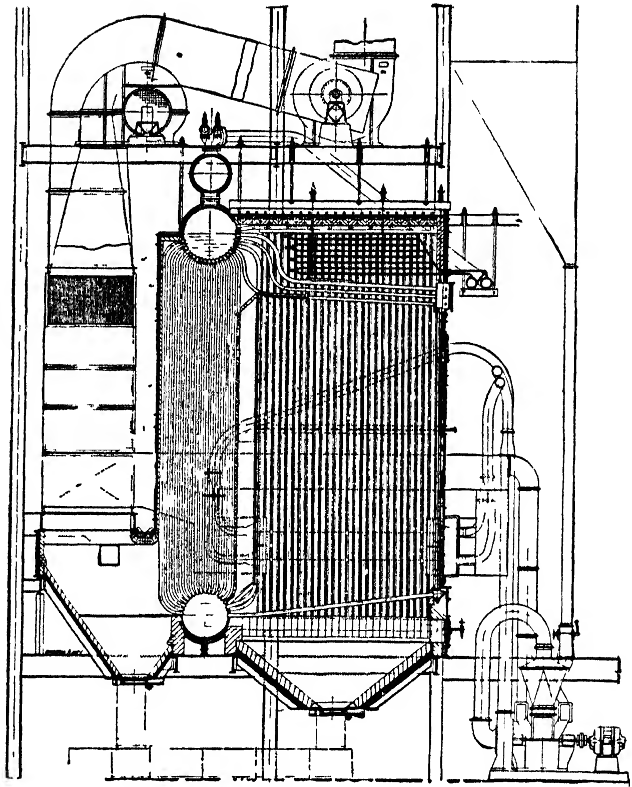


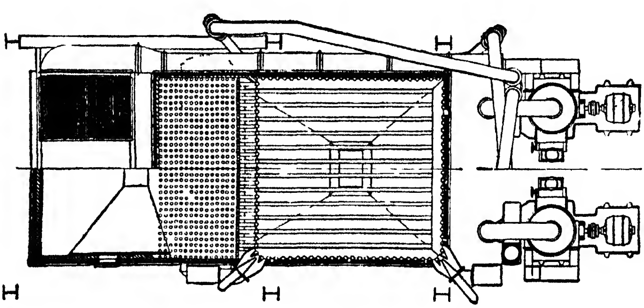
Fig. 198.—A Combustion Engineering steam generator.

The installation of Figure 199 is a recent single-pass vertical boiler. The combustion gases from the corner pulverized fuel burners pass through the top screen, over the superheater, down along the vertical boiler tubes and then out vertically through the plate-type air heater.

Figure 200 shows a section through a modern 1,400 lb per sq in. boiler plant equipped with water walls, water screen, four-pass boiler, convection superheater, air heater, water-sealed ashpit, and pulverized coal firing.



(*a*) Sectional elevation.



(*b*) Sectional plan through furnace.

FIG. 199.—Single-pass boiler or "steam generator." (*Combustion Engineering Corporation.*)

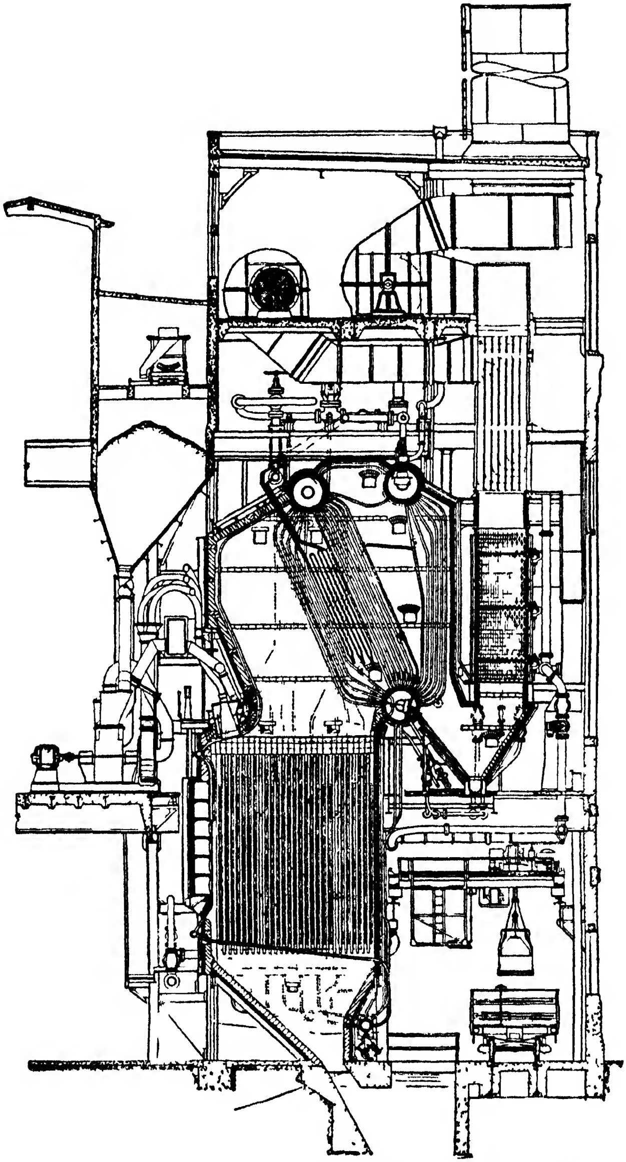


FIG. 200.—High-pressure boiler plant of the Kansas City Power & Light Co. 1,400 lb per sq in (*Ladd semi-vertical boilers.*)

Figure 201 shows a typical four-drum, Riley, steam-generating unit of 300,000 lb per hr capacity at 746 lb pressure. This pulverized-coal-fired unit has a water-cooled furnace and hopper bottom, convection and radiant superheater, integral economizer

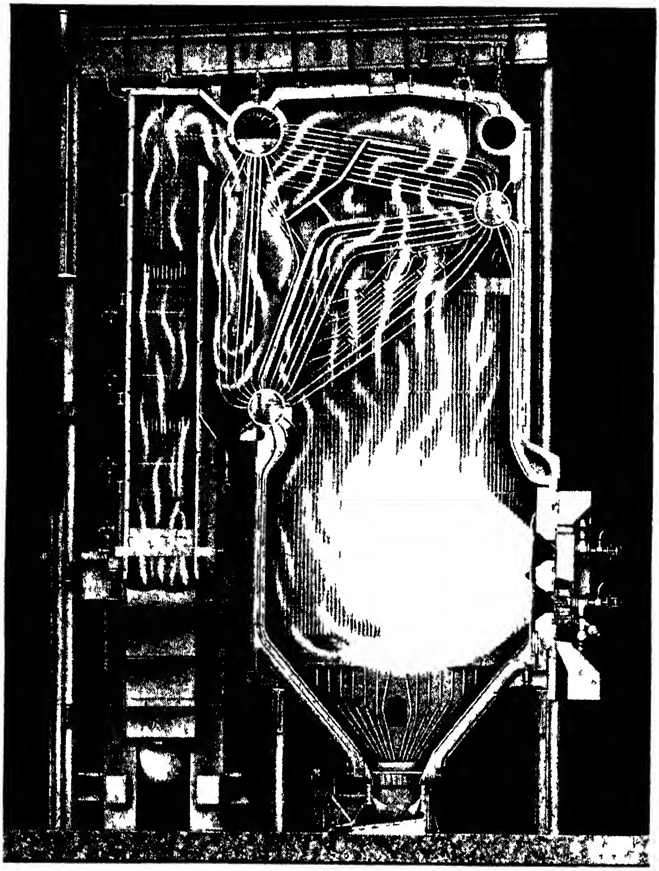


FIG. 201 —A Riley steam-generating unit

and a three-pass, tubular air preheater. The baffling is arranged for flow of gases across the tubes.

An industrial steam generator is shown in Fig. 202. The unit consists of a two-drum boiler of relatively small surface, completely water-cooled furnace with water-cooled continuous slag-tap bottom, convection superheater, separate economizer, and

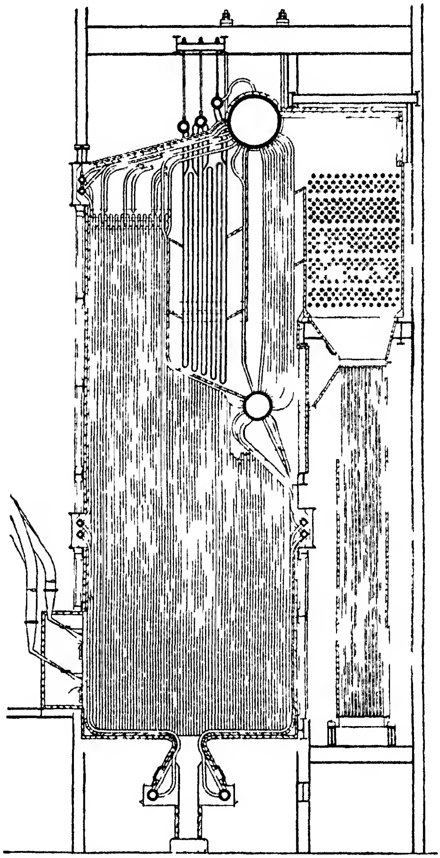


FIG 202 — A Foster Wheeler industrial steam generator

tubular air preheater The baffling is for longitudinal gas flow.

CHAPTER XVIII

BOILER FUELS AND FURNACES

181. Origin of Coal.[1]—It is generally accepted that coal is of vegetable origin and that it is the result of the transformation of the plant material in two stages. During the coal forming age, vegetable growth was very rapid and dead plant material accumulated in thick beds. As the deposits were laid down, they were acted upon by bacterial fermentation, which transformed the cellulose into peat. These peat deposits later became covered with earth or rock and, under the action of great pressure and temperatures, were very much compacted and devolatilized. The devolatilization consisted of a destructive distillation of the peat and driving off of water, carbon dioxide, carbon monoxide, and some of the lower hydrocarbons, thus leaving free carbon, various complex hydrocarbons and other compounds, water, and noncombustible mineral matter. As the process continued over longer periods of time, more of the gases were evolved and driven off, so that, generally, the greater the geological age of the deposit, the less will be the proportion of the volatile constituents and the greater will be the proportion of free or "fixed" carbon.

The ash content of coal was formed from the mineral content of the original plant material or by the direct admixture of sand or rock or by precipitates from mineral-bearing waters which filtered through the beds.

182. Analysis of Coal.—From the foregoing article it is seen that coals might be analyzed by separating the four constituent parts, moisture, volatile substances, fixed carbon, and ash. In the laboratory the process consists of heating pulverized samples of coal in the following manner:[2]

1. The *moisture content* is determined by heating a weighed sample for 1 hr in an open crucible in an oven at 104 to 110°C.

[1] *Bur. Mines, Bull.* 38, 1913.

[2] *Bur. Mines, Tech. Paper,* 8, 1929.

at standard barometer. The sample is then cooled in a desiccator and weighed again, the loss being the moisture.

2. The *volatile matter* is determined by heating a second weighed sample for exactly 7 min in a covered crucible in an electric furnace at 930 to 970°C. The sample is then cooled and weighed, the loss being moisture plus volatile matter.

3. The *ash* content is determined by heating the dried sample from the moisture determination in an open crucible in an electric furnace with good air circulation at 700 to 750°C., the crucible being placed in the cold furnace and the temperature of the two raised together. This drives out the volatile and burns all combustible matter, the process being continued until there is no further loss in weight. The residue is the ash.

This analysis of the coal into four parts is called the *proximate analysis*. This should not be confused with "approximate."

For some calculations it is desirable to know the properties of a coal on a "moisture-free" basis. For example, if it is desired to state the ash content in moisture-free coal if the proximate analysis is given, we should have

$$A_d = \frac{A_m}{1 - M}$$

in which A_d = fraction of ash in dry coal.
A_m = fraction of ash in moist coal.
M = fraction of moisture in moist coal.

On the other hand, it is sometimes desirable to express some property of the coal on a "moist basis" or "as-received basis," when the property on the "dry basis" and the moisture content of the moist coal are known. For example, if we have the heating value of the coal, which is always determined on the dry basis, we may express it on the as-received basis, thus:

$$(\text{Btu})_m = (1 - M)(\text{Btu})_d$$

in which $(\text{Btu})_m$ = heating value per pound moist coal.
$(\text{Btu})_d$ = heating value per pound dry coal.

It is assumed, of course, in this expression, that the water is inert and does not take heat for its evaporation. The correction for this heat quantity is discussed later.

A second analysis which is very useful in the solution of combustion problems is the **ultimate analysis.** This consists of the determination of the elementary substances in the fuel, the elements determined being carbon, hydrogen, oxygen, nitrogen, and sulphur. The ultimate analysis is always stated in terms of dry fuel, and, since ash is always a part of dry fuel, the percentages of the five elements named do not add up to 100. The ultimate analysis is very complicated and must be determined by an experienced chemist.

An abbreviated analysis, which consists of the determination of "total carbon" only, is very useful, as was shown in Chap. V. This test is made by a special "total-carbon" apparatus. In some cases it is important to know the sulphur content, and this is determined in a relatively simple manner by washing out the sulphurous and sulphuric acids resulting from the combustion of the coal in the bomb calorimeter when making the heat-value determination.

183. Commercial Classification of Coals.—The classification of coal into "ranks" is rather indefinite, the characteristics overlapping and merging gradually into each other. The practical method of ranking coals is according to their physical appearance, their combustion characteristics, and location of the deposits. The term "grade" applied to coal refers to the proportions of sulphur and ash and should not be confused with "rank." Thus, a high-grade coal would be one relatively free of sulphur and low in ash content.

The general properties of coals are given in the following paragraphs.

Anthracite, or "hard coal," is nearly all fixed carbon (except for ash and moisture). It has a deep black color, a shiny, metallic luster, few cracks; it is very compact, its specific gravity being 1.3 to 1.8. Anthracite kindles with difficulty and does not swell when burning but simply becomes red hot and then wastes away with very little flame or smoke.

Semianthracite is not quite so hard or so lustrous as anthracite. Its specific gravity is about 1.4. It can be distinguished by the fact that, when freshly broken, it will soot the hand. This coal contains more volatile matter than anthracite and, consequently, ignites more easily and burns with more flame.

Semibituminous coal is still softer and lacks most of the bright metallic luster but resembles anthracite more than bituminous coal.

Bituminous coals are commonly called "soft coals" and comprise a large and more varied group than coals of the other ranks. The specific gravity varies from 1.2 to 1.4, the color from dark brown to pitch black, and the hardness from that of semibituminous to lignite. Generally these coals are brittle with a glassy or greasy luster. Bituminous coals have a high-volatile content and burn with long flames, which are apt to be smoky.

There are two distinct types of bituminous coals, designated as *coking coals* and *noncoking*, or *free-burning*, *coals*. Coking coals are rich in hydrocarbons and are, therefore, valuable in gas manufacture. They fuse and swell when heated and form coke after the volatile is driven off. Noncoking coals do not fuse or swell and burn much more freely in the furnace.

Another kind of bituminous coal, which is very useful for gas making on account of its large hydrocarbon content, is *cannel* coal so called because it burns with a long candle-like flame. Cannel coal is found in relatively small deposits.

Subbituminous coal is frequently called "black lignite" and is not easily distinguishable from either lignite or bituminous coals. Its characteristics are its brownish-black color and its tendency to slack on exposure to the weather or even to the air.

Lignite is coal in the early stages of formation from the peat. The specific-gravity range is 1.2 to 1.23. In the beds it contains a very large percentage of moisture, often as much as 50 per cent. The color ranges from brown to black. It burns with a smoky flame. Lignite is very friable and does not stand handling in transportation. It is a low-grade coal, being high in ash, but is cheap and abundant.

Peat is found in large deposits in the United States but is not used to any appreciable extent. It is organic matter in the first stage of conversion to coal and, being found in bogs, contains 75 to 95 per cent of moisture. It is soft and porous and even if dried will reabsorb moisture from the air.

184. Analytical Classification of Coals.—The classification of coals by their physical characteristics, as outlined in the preceding article, is too uncertain and depends too much upon the personal element to be accurate. Many attempts have been made to rank coals on a scientific basis, some using the proximate analysis, some the ultimate, and others the combination of the two.

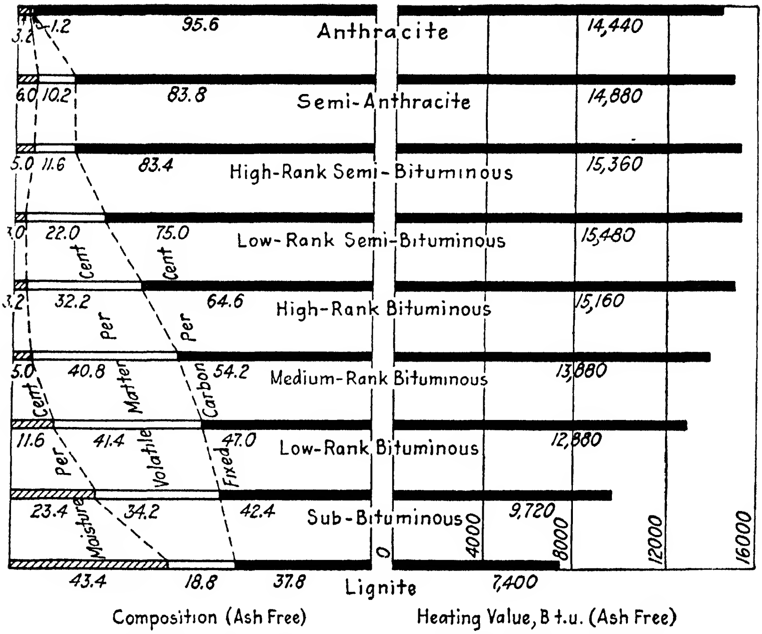


FIG. 203.— U. S. Geological Survey's classification of coals.

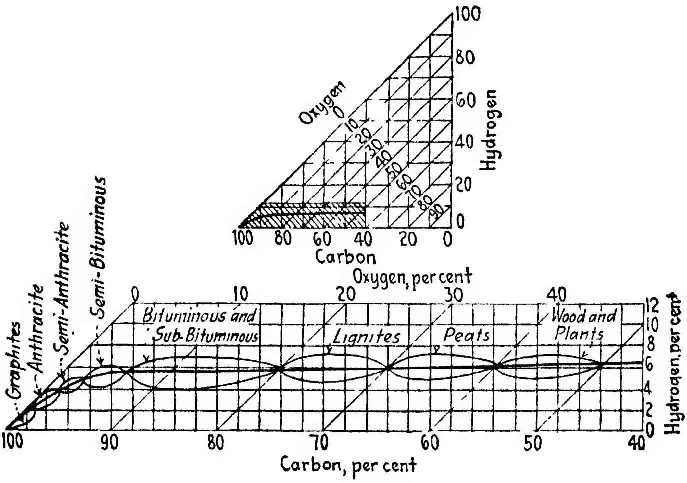


FIG. 204.—Ralston's classification of coals.

A classification of the U. S. Geological Survey based on the proximate analyses of typical coals is shown by the graph of Fig. 203. These figures indicate the approximate variation among the various ranks Since the ash content is a matter of grade rather than of rank, the analyses are on the ash-free basis.

A second method, devised by O. C. Ralston, is based on the ultimate analyis. The principle of this method is to plot the percentages of carbon, hydrogen, and oxygen in the sum of these three elements on a trilinear chart, as shown in Fig. 204. Since

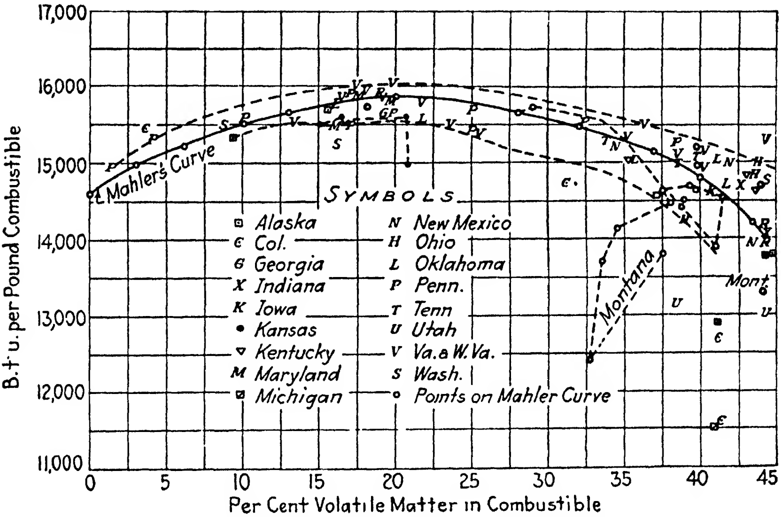


FIG. 205.—Relation of heating value to per cent volatile matter in combustible. (*After William Kent.*)

the sum of the distances from any point inside a triangle to the three sides is always the same, these three percentages may be plotted on the three scales, as shown. Figure 204 shows a group of analyses made by the Bureau of Mines and plotted on such a chart. Since the hydrogen percentage is always less than 10, only the bottom of the chart of Fig. 204 is used.

185. Estimating Heating Value from Proximate Analysis.—Figure 205 is a graph by William Kent, showing the relation between the heating value of coals per pound of combustible (volatile plus fixed carbon) and the percentage of volatile matter in the combustible. This chart was plotted for 155 typical analyses chosen from 3,000 collected by the Bureau of Mines.

It will be noted that, in general, coals from the Eastern fields lie in a fairly narrow band, but Western coals, which are high in volatile and oxygen and nitrogen, do not follow any apparent law. The coals of Montana, for example, seem very erratic. The heavy line represents the data collected by Mahler for European coals.

186. Distribution of American Coal Deposits.—The coal fields of the United States are shown in the map of Fig. 206. The available reserve supply of the various kinds of coal is given in Table XIV

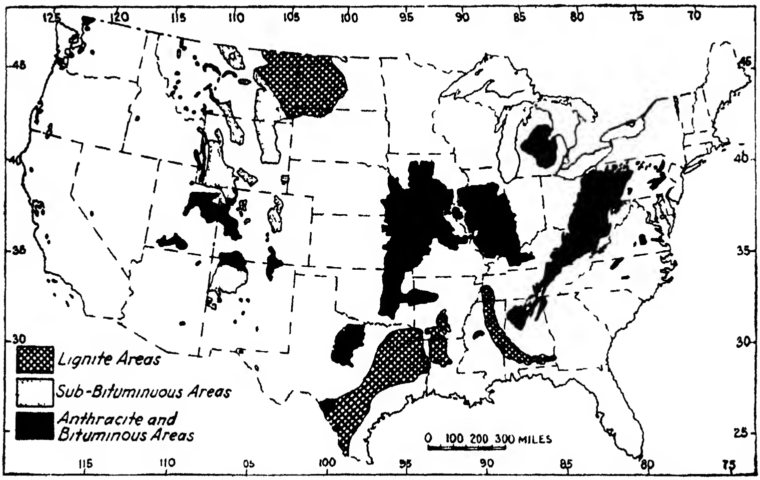


FIG. 206—Coal fields of the United States. (*After M. R. Campbell.*)

187. Typical American Coals.—Table XV[1] gives data of a number of typical coals.

188. Wood Fuels.—In certain sections of the country, especially the Pacific Coast, the use of wood for steam-raising fuel is of considerable importance. The heating value of most woods that are used for fuel is about 8,500 Btu per lb of combustible, the soft, resinous woods being slightly higher. Green woods contain 30 to 50 per cent of moisture, which is reduced to 18 or 20 per cent after drying for a year. Woods dried in the summer will reabsorb moisture from the air during the rainy season.

For hand-fired furnaces, wood is generally cut in cordwood lengths (4 ft). Sawmill refuse is *hogged,* that is, chopped into

[1] Complied from Bur. Mines, *Bull* 22 and others.

TABLE XIV.—FUEL RESERVES OF THE UNITED STATES

Fields	Area, sq. miles	Reserve, short tons
The anthracite fields:		
1. Eastern Pennsylvania	480	16,000,000,000
2. Colorado, New Mexico	29	?
Bituminous fields of the better grade:		
3. Atlantic Coast triassic fields (Virginia, North Carolina)	210	199,000,000
4. Appalachian Fields (Pa., Ohio, Md., Va., W. Va., Ky., Tenn., Ala., and Ga.)	69,755	530,000,000,000
5. Eastern interior fields (Ind., Ill., and western Ky.)	47,000	318,000,000,000
6. Northern interior field (Mich.)	11,000	11,900,000,000
7. Western interior fields (Ia., Neb., Kan., Mo., Ark., Okla., and parts of Texas)	74,900	187,500,000,000
Lignite and bituminous fields (chiefly low-grade bituminous and lignite coals):		
8. Gulf Coast lignitic field (Ark. and Texas)	2,100	20,000,000,000
9. Great Plains and Rocky Mountain fields (N. Dak., S. Dak., Mont., Wyo., Ida., Colo., N. Mex., Utah)	126,022	1,969,000,000,000
10. Pacific Coast fields (Wash., Ore., Cal.)	1,900	21,800,000,000
11. Alaska	1,210	?
Total	....	3,074,399,000,000

small bits by means of a hogging machine or macerator. In such hogged fuel and sawdust the moisture content may run as high as 60 per cent.

Refuse from planing mills and woodworking plants contains a much smaller moisture content, because the material has been either kiln-dried or air-dried.

189. Bagasse.—The refuse of sugar cane after the extraction of the juice has a good heating value and is used for boiler fuel in the industry that produces it. This fuel, called "bagasse," contains 40 to 55 per cent of moisture and has a heating value of about 8300 Btu per lb dry, about the same as other woody fibers. As with wet wood and lignite, a large portion of the available heat must be used to evaporate the high moisture content.

190. Oil and Gas Fuels.—Fuel oil for boiler firing may be, and usually is, of the lower grades which are not only cheap but high

TABLE XV.—ANALYSES OF TYPICAL AMERICAN COALS

Local name	Proximate analysis				Ultimate analysis					Heating value, Btu per lb as recd.
	Moisture	Volatile matter	Fixed carbon	Ash	Sulphur	Hydrogen	Carbon	Nitrogen	Oxygen	
Alabama:										
Dolomite	3.16	25.40	67.75	3.69	0.56	5.05	82.28	1.36	7.06	14,606
Littleton	2.53	26.94	59.48	11.05	0.79	4.80	74.44	1.59	7.33	13,286
Alaska:										
Mantanuska R	1.72	24.36	58.97	14.95	0.46	4.46	70.78	1.42	7.93	12,585
Port Graham	19.96	38.73	32.46	8.85	0.52	0.52	49.53	0.92	34.37	8,793
Arizona:										
Oraibi	9.88	32.64	46.86	10.62	1.12	5.42	62.00	1.13	19.71	10,800
Arkansas:										
Paris	2.77	14.69	73.47	9.07	2.79	4.02	78.71	1.46	3.95	13,774
Greenwood	3.21	14.84	72.66	9.29	3.12	3.75	78.37	1.52	3.95	13,588
California:										
Stone Canon	6.95	46.69	40.13	6.23	4.17	6.28	66.01	1.17	16.14	12,447
Colorado:										
Newcastle	4.45	42.05	49.56	3.94	0.44	5.43	72.57	1.72	15.90	13,129
Pikeview	26.20	29.67	37.67	6.46	0.30	6.13	49.36	0.66	37.09	8,552
Georgia:										
Menlo	3.80	15.88	65.83	14.49	1.27	4.32	70.59	1.09	8.24	12,790
Idaho:										
Haden	11.45	37.24	47.01	4.30	0.54	5.94	68.09	1.40	19.73	12,094
Illinois:										
Harrisburg	6.01	32.37	54.32	7.30	1.66	5.27	71.63	1.34	12.80	12,793
Auburn	16.00	32.41	37.82	13.77	4.05	5.55	53.89	0.91	21.83	9,940
Carterville	9.18	27.30	55.40	8.12	0.90	5.10	68.45	1.14	16.29	12,015
Centralia	9.95	34.76	42.06	13.23	3.87	5.25	59.64	1.04	16.97	10,960
Indiana:										
Linton	13.58	32.07	46.20	8.15	0.91	5.65	63.53	1.42	20.34	11,419
Brazil	16.91	26.85	38.87	17.37	1.89	5.48	52.97	1.01	21.28	9,524
Dugger	13.48	32.51	48.38	5.63	1.09	5.94	66.01	1.49	19.84	11,788
Iowa:										
Laddsdale	8.24	30.74	45.02	16.00	5.03	4.81	59.82	0.94	13.40	11,027
Chariton	15.39	30.49	41.49	12.63	3.19	5.74	55.81	1.14	21.49	10,242
Kansas:										
Fuller	4.85	33.53	52.52	9.10	4.95	5.08	71.20	1.24	8.43	12,942
Lansing	11.10	35.51	40.69	12.70	3.99	5.30	60.72	1.13	16.16	11,065
Kentucky:										
Flambeau	2.36	48.40	38.75	10.49	1.20	6.47	71.98	1.16	8.70	13,770
McHenry	9.89	35.94	43.36	10.81	3.64	5.37	62.27	1.33	16.58	11,392
Maryland:										
Lord	2.26	16.05	75.86	5.83	0.79	4.68	82.45	1.73	4.52	14,483
Eckhart	2.70	14.50	74.00	8.80	1.00	4.44	79.21	1.69	4.86	13,910
Michigan:										
Saginaw	11.91	31.50	49.75	6.84	1.24	5.84	66.56	1.19	18.33	11,781
Missouri:										
Kirksville	15.98	38.15	37.18	8.69	4.12	5.90	59.09	0.94	21.26	10,798
Hamilton	10.99	35.00	41.37	12.64	4.81	5.43	60.40	1.16	15.56	11,093
Montana:										
Bear Creek	9.67	35.92	46.39	8.02	1.64	5.52	61.66	1.48	21.68	10,832
Miles	29.13	25.33	30.51	15.03	0.55	5.60	40.09	0.54	38.19	6,662
New Mexico:										
Baton	2.12	36.06	50.22	11.60	0.64	4.94	69.96	1.33	11.53	12,965
Blackrock	14.69	34.93	41.56	8.82	0.79	5.82	60.93	1.12	22.52	10,809
North Dakota:										
Williston	36.78	28.16	29.97	5.09	0.48	6.93	41.87	0.69	44.94	7,204
Leith	36.18	29.77	25.35	8.70	0.68	6.76	39.45	0.59	43.82	6,700
Ohio:										
Amsterdam	3.50	37.98	51.08	7.44	3.09	5.43	73.39	1.46	9.19	13,286
Wellston	7.71	38.32	42.02	11.95	4.61	5.41	62.49	1.11	14.43	11,515
Oklahoma:										
McCurtain	2.70	21.07	69.88	6.35	0.77	4.46	81.33	1.67	5.42	14,098
Lehigh	7.07	36.41	45.68	10.84	3.64	5.13	64.38	1.44	14.57	11,468

TABLE XV.—ANALYSES OF TYPICAL AMERICAN COALS (*Continued*)

Local name	Proximate analysis				Ultimate analysis					Heating value, Btu per lb as recd.
	Moisture	Volatile matter	Fixed carbon	Ash	Sulphur	Hydrogen	Carbon	Nitrogen	Oxygen	
Oregon:										
Beaver Hill	16.10	31.10	39.63	13.17	0.81	5.53	51.07	1.19	28.23	9,031
Pennsylvania:										
Carroltown Road	0.93	23.10	69.29	6.68	1.30	4.81	81.64	1.26	4.31	14,485
Dunmore	3.43	6.79	78.25	11.53	0.46	2.52	78.85	0.77	5.87	12,782
Windber	2.40	13.50	77.80	6.31	1.26	4.44	82.62	1.31	4.06	14,370
Smoke Run	3.73	20.29	68.41	7.57	1.33	4.86	78.92	1.22	6.10	13,970
Rhode Island:										
Cranston	4.54	3.01	78.69	13.76	0.87	0.46	82.39	0.12	1.75	11,624
Portsmouth	22.92	2.78	58.37	15.93	0.10	2.84	58.46	0.18	22.49	8,528
South Dakota:										
Lodgepole	39.16	24.68	27.81	8.35	2.22	6.60	38.02	0.53	44.28	6,307
Tennessee:										
Lafollette	2.92	32.04	58.23	6.81	1.14	5.19	74.95	1.62	10.29	13,514
Dayton	1.76	27.86	49.57	20.81	0.49	4.51	66.24	1.19	6.76	11,666
Texas:										
Hoyt	33.71	29.25	29.76	7.28	0.53	6.79	42.52	0.79	42.09	7,348
Crockett	34.70	32.23	21.87	11.20	0.79	6.93	39.25	0.72	41.11	7,056
Utah:										
Emery	3.93	40.92	49.22	5.93	0.39	5.52	73.02	1.25	13.89	12,965
Coalville	14.20	36.00	44.80	5.00	1.41	5.79	61.40	1.09	25.31	10,630
Virginia:										
Pocahontas No. 3	3.50	15.50	76.80	4.20	0.73	4.77	83.36	1.08	5.86	14,630
Clinchfield	2.12	33.75	57.68	6.45	0.65	5.44	78.59	1.53	7.34	14,135
Montgomery Co.	1.88	12.34	66.81	18.97	0.67	3.54	72.33	0.85	3.64	12,157
Washington:										
Roslyn	3.89	37.00	46.49	12.62	0.37	5.58	68.55	1.31	11.57	12,434
Centralia	25.08	32.25	34.02	8.65	0.82	6.37	47.26	0.91	35.99	8,170
West Virginia:										
Stonewall	3.02	16.06	78.75	2.17	0.80	5.02	85.02	1.40	5.50	15,001
Roderfield	2.32	16.76	69.80	11.12	1.78	4.35	77.46	1.27	4.02	13,514
Coaldale	3.43	14.58	77.89	4.10	0.67	4.79	83.79	1.06	5.59	14,602
Wyoming:										
Kirby	15.86	33.01	47.39	3.74	0.59	6.06	62.03	1.29	26.29	10,984
Monarch	23.88	34.33	38.44	3.35	0.38	6.29	54.07	1.14	34.77	9,335

in heat value. Commercial fuel oils are classified in six grades according to viscosity, the navy specifications for the three lowest being Bunker A, Bunker B, and Bunker C. Of these Bunker C is the cheapest and is the grade usually used as boiler fuel. It has a viscosity of not over 300 sec Saybolt Furol at 122°F., a density of 5 to 14° A.P.I.,[1] and a heating value of about 18,500 Btu per lb. It should not contain more than 2 per cent water or more than 0.25 per cent sediment. Fuel oil possesses several advantages over coal which must be balanced against the somewhat higher cost of oil on a Btu basis. For example, weight is about one-third less and volume about one-half less than coal for the same heat content. Ease of handling, flexibility

[1] Degrees A.P.I. $= \left[\frac{141.5}{(\text{sp. gr. at } 60°/60°\text{F.})}\right] - 131.5.$

in firing, freedom of deterioration in storage, freedom from smoke and ash, and high efficiency are in favor of oil firing.

Natural gas is used for boiler firing chiefly in the regions in or near the gas wells although long gas lines are making this fuel increasingly available to centers of population. The characteristics of natural gas are given in Chap. IV. Tests[1] indicate that the inherent efficiency of gas firing is slightly below that of fuel oil.

191. Hand Firing.—When green coal is thrown on the fire, the heat distills off the volatile matter, which is consumed if the furnace temperature is high enough and the air supply sufficient, or passes out as smoke if conditions are not proper. For good results coal should be burned without smoke, and to accomplish this two general methods of firing have come into use.

The *alternate-spreading method* is to fire the coal in small amounts spread evenly over half of the grate at a time so that the entire fire is not cooled by the fresh coal. In this way, the volatile matter is exposed to the incandescent parts of the fuel bed and mixes with the clean flame from it and is consumed.

The *coking method* is to fire large quantities of fresh coal at the front of the grates on the *dead plate* just inside the fire doors. When the volatile has been driven off and consumed in passing over the incandescent fire to the rear, the resulting coke is pushed toward the bridge wall and a fresh charge fired in front.

In hand firing, three points should be watched particularly: (1) The ashpit should be bright at all times, indicating that the air spaces in the grate are not choked with ash or clinker; (2) holes must not be allowed to develop in the fuel bed through which air will enter the furnace without being used for combustion, thus cooling the boiler instead of heating it; (3) the furnace doors must be kept open as little as possible, because the entering cold air cools the furnace.

Figure 207 shows a type of hand-fired furnace especially adapted for use with high-volatile coals. The fire on the lower grate is fed by coal that falls through the upper grate, which consists of water tubes connected through crossheaders to the boiler shell. The green coal is fired on the upper grate, where the volatile matter is distilled off and, the draft being downward,

[1] For example, *Trans. A.S.M.E.*, Fuel and Steam Power, 54: 11, 1932.

is drawn through the coals on this grate to the fire below. The fire below serves to ignite any volatile which passes unburned through the upper grate. In this way smokeless combustion is achieved.

Shaking or dumping grates permit the ash and clinker that collect on them to be dumped into the ashpit without opening the firing doors or disturbing the fire to any great extent. With stationary grates it is necessary to scrape the fire aside, break up the clinker with a slice bar, and hoe it out of the firing door. The

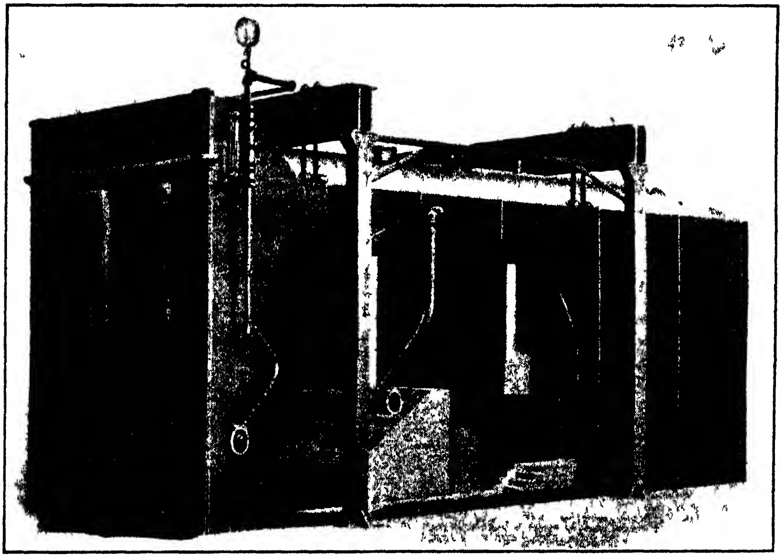


FIG 207 – Kewanee tubular boiler with smokeless down-draft furnace

fire is then replenished, and, when well under way, the other side is cleaned in the same manner.

Clinkering is caused by the fusing of the ash and takes place especially at those points where the fuel bed is disturbed so as to throw the ash up into the incandescent plane of the fire. Coals high in sulphur are generally apt to clinker. The thickness of the fuel bed for best performance varies with (1) the available draft, see Art. 198; (2) the kind and size of coal; and (3) the load on the boiler. A thick fire requires a stronger draft than a thin one, and fine coal a stronger draft than coarse coal. The greater the load the thinner should be the fire, in order to get the requisite quantity of air through the fuel bed, but the percentage of excess air increases as the fire becomes thinner. Holes are more apt to develop in a thin fire than in a thick one. Semibituminous coals are fired 10 to 14 in. thick and at 10 to 20-min. intervals; high-grade bituminous coals 4 to 6 in.; and subbituminous coals of the Western fields about 4 in.

192. Mechanical Stokers.—Hand firing is hard work and requires skill and some knowledge of combustion. Unfortunately, the supply of this high-class labor is limited, and the high steaming rates and the high economies of modern boilers are such that hand firing is no longer adequate. Even in small plants having a single boiler of 100 hp or less, "mechanical firemen" are being used to advantage.

The advantages resulting from stoker firing are

1. Higher efficiency due to more consistent proportioning of air to fuel. Since it is rarely necessary to open furnace doors, the entrance of cold air is very much less than for hand-fired furnaces, thus producing a higher carbon dioxide content and higher furnace temperatures.
2. Use of cheaper grades and smaller sizes of fuel.
3. Uniform and continuous firing instead of intermittent firing.
4. Saving in labor costs due to the fact that one fireman can fire greatly increased quantities of fuel with the aid of mechanical stokers. Even in small plants there is an indirect saving in labor cost, because the fireman is released for maintenance and supervision work which he could not do with hand firing.
5. High rates of steam output per boiler horsepower. This is due to the larger grate areas as well as to the ability of the stoker to burn larger quantities of fuel per square foot of grate surface.
6. Ability to care for sudden fluctuations in steam demand and still maintain constant pressure.
7. Smokeless combustion. It is possible to find on the market stokers suitable for every grade and rank of coal and which will operate with practically smokeless combustion.

Against these advantages should be balanced

1. Cost of stoker equipment, which imposes heavy investment and maintenance costs on the coal fired. With, however, the high wage levels existing in the United States and the mounting fuel costs, the quantity of coal fired per day must be very small not to be economically handled by stokers.
2. Operating cost. Stokers require power for their operation, and this power must be deducted from the gross boiler output.
3. The higher furnace temperatures increase the maintenance costs of furnace linings.

It should be pointed out that for most plants the decision does not lie between hand firing and stoker firing but between types of stokers for a particular installation or between stokers and pulverized fuel or oil. The problem, in each case, is one of economic selection, and the above advantages and disadvantages must be reduced to a common dollar basis before an intelligent choice can be made

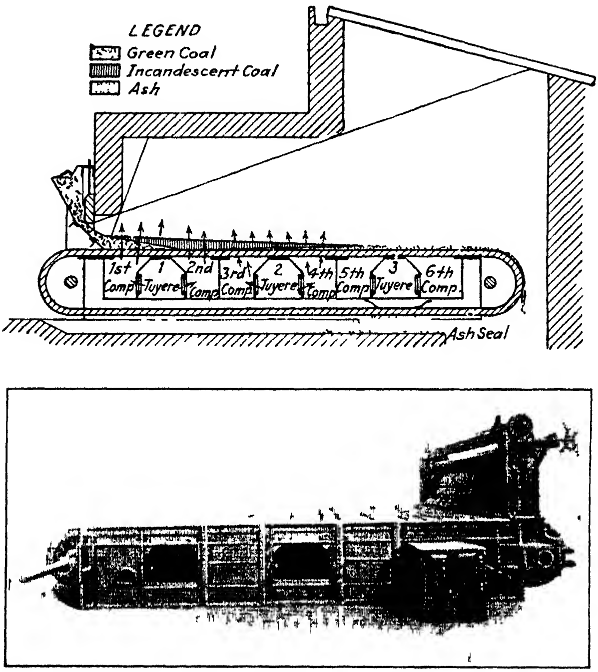


FIG 208 —Coxe traveling-grate stoker Upper view shows progressive combustion of the coal

Mechanical stokers are built in three general types: (1) traveling or chain grate, (2) overfeed, and (3) retort underfeed.

193. Chain-grate Stokers.—Chain-grate stokers consist of endless chains composed of short grate bars passing over drums or sprockets at the front and rear of the furnace. The coal is fed by means of a hopper to the front end of the grate and is there ignited as it passes under an ignition arch which reflects the heat of the hot part of the fire to the point of entrance of the fuel. Combustion progresses as the coal is moved toward the rear of the furnace and is completed when the rear of the furnace is reached. The ash and refuse are discharged by gravity as the grate makes the turn over the rear drum or sprocket.

Chain-grate stokers may be designed for use with natural draft or forced draft. Three points of control in their operation are (1) the thickness of the fuel bed, which is adjusted by a gate at the bottom of the coal hopper, (2) the speed of travel of the grate, which is adjusted by means of gears, pulleys, or driver

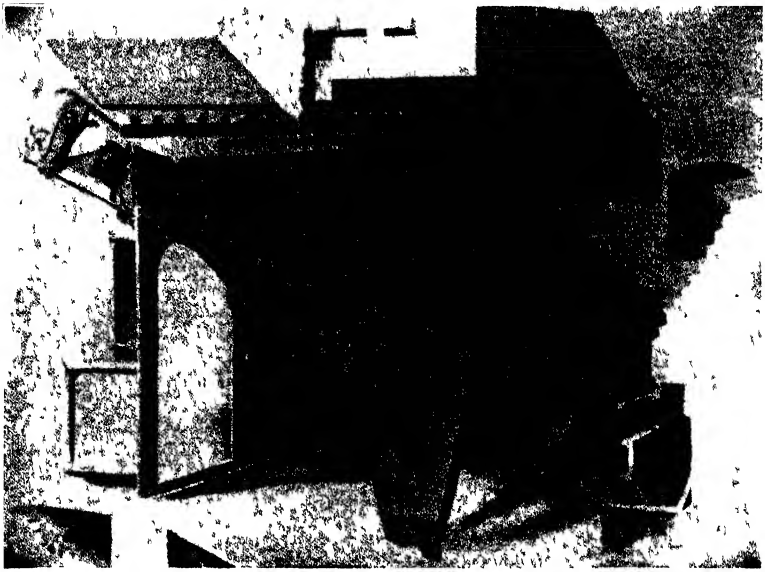


Fig. 209.—Westinghouse "new model" Roney stoker.

speed, and (3) the draft through the grate, which is adjusted by means of dampers or changes in fan speed.

Figure 208 shows a typical forced-draft, traveling-grate stoker. Chain-grate stokers are particularly adapted to noncoking coals, ranging from the high-volatile, high-ash, free-burning, clinkering coals to the smaller sizes of anthracite coals.

194. Overfeed Stokers.—There are two types of gravity overfeed stokers, those in which the coal is fed into the front of the furnace onto grates which incline toward the rear, and those in which the coal is fed along the entire length of each side of the furnace onto grates which incline downward toward the center in the form of a V. In the first class the grates are usually reciprocated, and the motion carries the fuel slowly to the rear, where it is discharged as ash after combustion is completed. In the second class the fuel is moved by rocking bars and is discharged as ash in the center of the furnace

The fuel is *coked* on a plate at the top, and the volatile which is driven off is ignited by passing over the burning carbon farther along the grate.

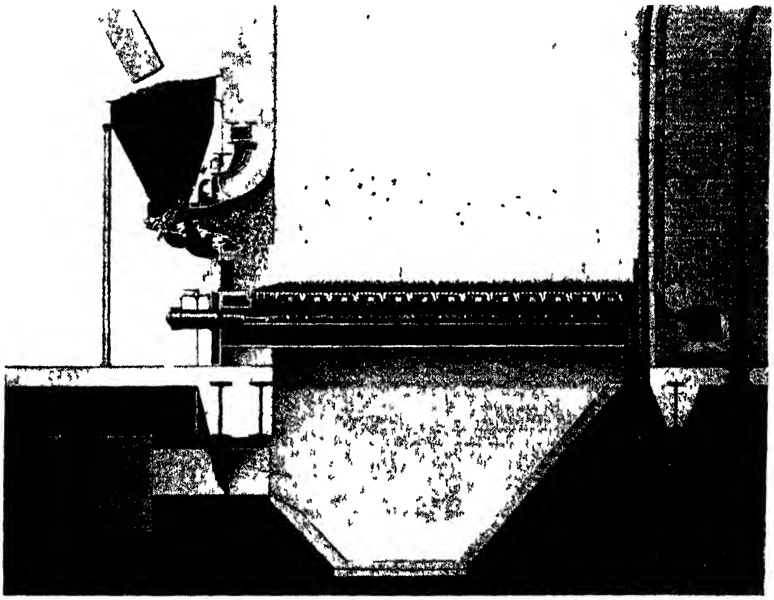


FIG 210 Detroit spreader-type stoker

Overfeed stokers are built in sizes from 50 to 600 boiler hp and for use with natural draft. They are best adapted to steady-load conditions.

In Fig. 209 is shown the Westinghouse-Roney front-feed, overfeed stokers.

The spreader type of stoker has found favor for the combustion of lignite and low-rank bituminous coals under boilers of moderate size. It consists of a hopper at the boiler front which feeds coal by gravity onto a plate inside the firebox, from which it is thrown to the fire in very small amounts at a time by means of an air jet or by mechanical means. In this way the fines are burned in suspension, and the coarser fuel is fed uniformly and thinly over the entire grate surface. Grates are of the rocking type and are dumped intermittently by the operator.

Figure 210 shows a spreader-type stoker using a mechanical spreader.

195. Retort Underfeed Stokers.—In underfeed stokers, the fuel is fed into the furnace at the front in a nearly horizontal direction by means of intermittent rams or plungers or by a continuous screw The coal is fed from hoppers at the stoker

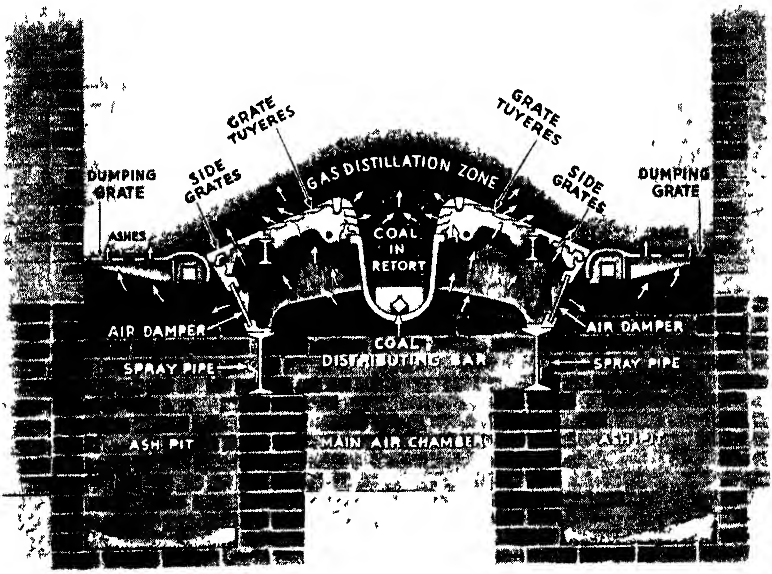


FIG 211 Detroit single-retort "Unistoker"

front. The rams force the coal into cast-iron *retorts*, the sides of which are perforated plates or blocks called *tuyeres* through which the air for combustion is forced by means of a blast fan As the coal enters the furnace from underneath, the heat distills off the volatile matter, which must pass up through the incandescent fuel bed with the air and is, thus, consumed.

In some underfeed stokers the retorts are sloping away from the ram, while in others the push of the ram alone forces the coal back. As combustion proceeds, the ash residue is shoved along with the coal and is finally dumped to the ashpit when combustion is complete.

Underfeed stokers are capable of handling very heavy overloads and are readily adjusted to fluctuating loads. The fuel bed is necessarily very thick, and this requires a high blast pressure Since the fuel bed is continually agitated, these stokers handle coking coals to very good advantage.

Figure 211 shows a cross-sectional view of a single-retort stoker in which the coal is pushed into the retort in the center in a direction perpendicular to this view. Figure 212 shows a single-retort stoker with screw feed to the retort. Single-retort stokers are made only in small and moderate capacities.

Figure 213 illustrates a multiple-retort stoker. Recently water cooling has been applied to multiple-retort stokers by means of

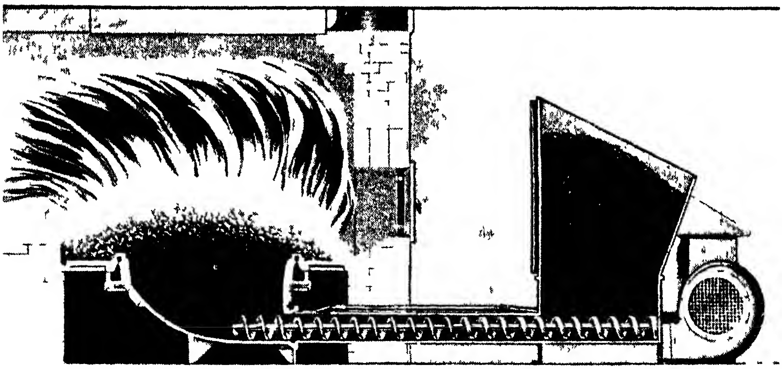


FIG 212 Iron Fireman screw-feed stoker

cooling tubes placed in recesses in the cast-iron tuyeres. Multiple-retort stokers have been built for capacities up to 24,000 sq ft of boiler surface, the limitation being the practical width which may be accommodated under the boiler.

196. Powdered-fuel Equipment.—If coal is pulverized very finely and blown into the furnace with the proper amount of air, it will burn with an intense flame similar to that of gas or oil This method firing coal possesses two inherent advantages: (1) that the quantity of excess air can be controlled and kept down to a very small percentage, thus producing high combustion temperatures and high efficiencies; (2) that the lower grades and ranks of coal may be burned with as good efficiency as the better coals. The use of pulverized fuel, however, involves three major problems: (1) pulverizing the fuel to the proper fineness, (2) maintaining brickwork under the high furnace temperature, and (3) taking care of the ash content of the coal.

Coal either may be pulverized in a central pulverizing plant and delivered to the boiler room ready for use, or it may be prepared as required for use by *unit pulverizers* in the boiler room.

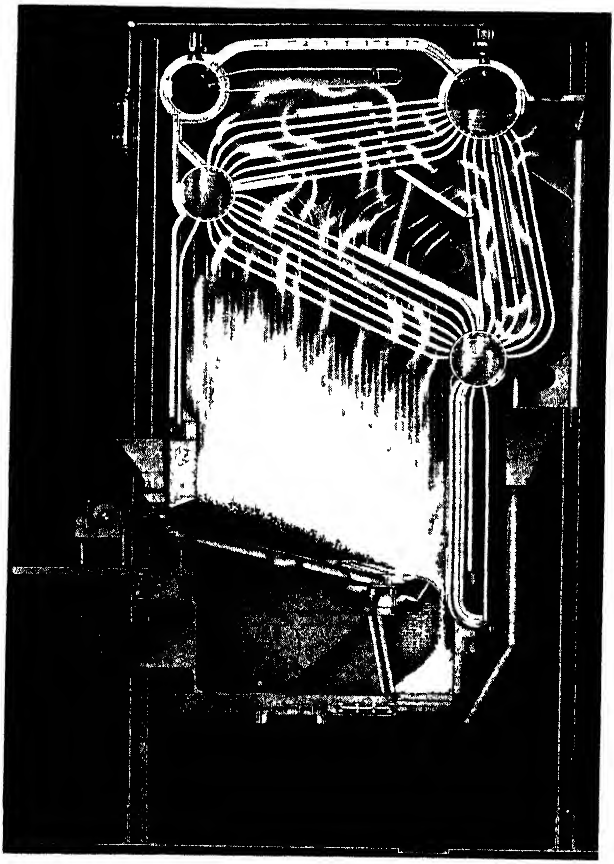


Fig. 213.—Riley Underfeed stoker under ring-flow type boiler.

The pulverizers themselves are of three types, *roller* mills, *ball* mills, and *hammer* mills.

Figure 214 shows a sectional view of a "bowl mill" in which the coal is ground between the whirling bowl and the rolls mounted on pivoted axes. Coal fed into the center is thrown to the sides of the bowl by centrifugal force where it is pulverized. At the same time the coal works its way up the inclined sides of the bowl to the rim where the fines are picked up by the upward air current in the annular space around the bowl and are carried to the classifier above. The coarser particles are deflected back into the bowl, and the oversize material from the classifier is returned to the bowl with the raw feed, while the prepared fines pass to the burner.

Figure 215 shows the so-called "ball-bearing" type of pulverizer in which the coal is ground between large balls and rotating

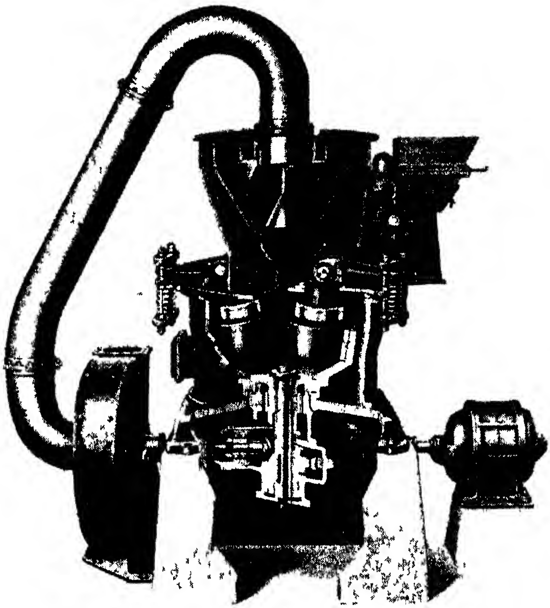


FIG 214.—Combustion Engineering "bowl-mill" pulverizer.

and stationary rings The rotating ring floats on, and is driven by, the vertical shaft and in turn propels the balls. A blast of air picks up the ground coal and classifies it, the fines being carried to the burner

In the ball mill of Fig. 216 the grinding is done between the balls and between the balls and the liners. A special air-actuated device controls the level of coal in the mill which is kept constant regardless of load. Preheated air drawn through the mill by an exhauster picks up the ground material and classifies it, the rate of air flow automatically governing the quantity of coal dust removed from the grinder.

Figure 217 shows a "Unipulvo"-unit mill of the hammer type. Fuel 1 is fed into the pulverizing chamber by the adjustable reciprocating feeder 2, the level of the fuel bed under the rotor being maintained automatically by the "feeler" bars 3. Tramp iron and other extraneous material accumulate in the space 4

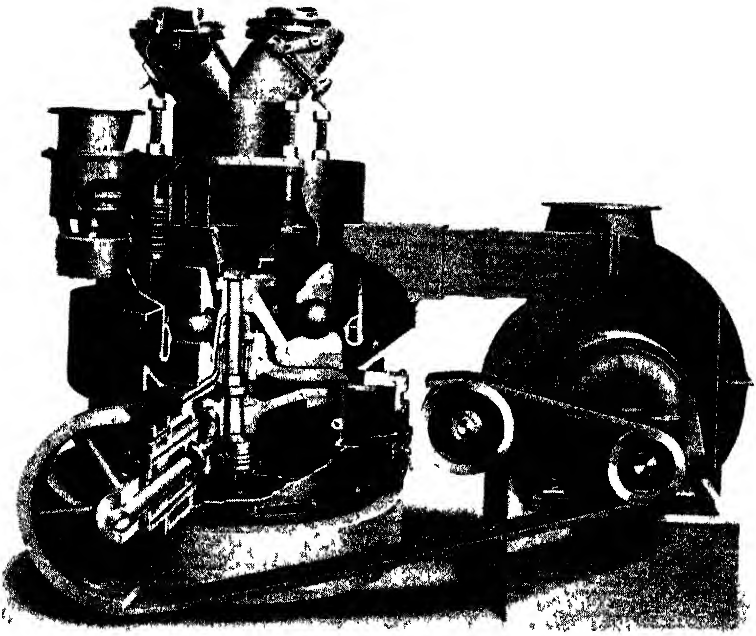


FIG. 215.—Babcock and Wilcox "ball-bearing" pulverizer

The rotor 5 having swinging hammers 6 revolves at 1,750 rpm, setting up a fan action which sweeps over the incoming fuel and around the fuel bed. The combustion air 8, which is preheated, is drawn into the pulverizing chamber by an induced-draft fan. The pulverized product passes to the separator 9, where expansion and the "spinner" 10 eliminate oversized particles, which return to the pulverizing chamber. The finely pulverized coal then passes from the separator through pipe 12 to the induced-draft fan and on to the burner spout.

Grindability of coal depends upon hardness and moisture content, the energy required for pulverizing varying from 5 to

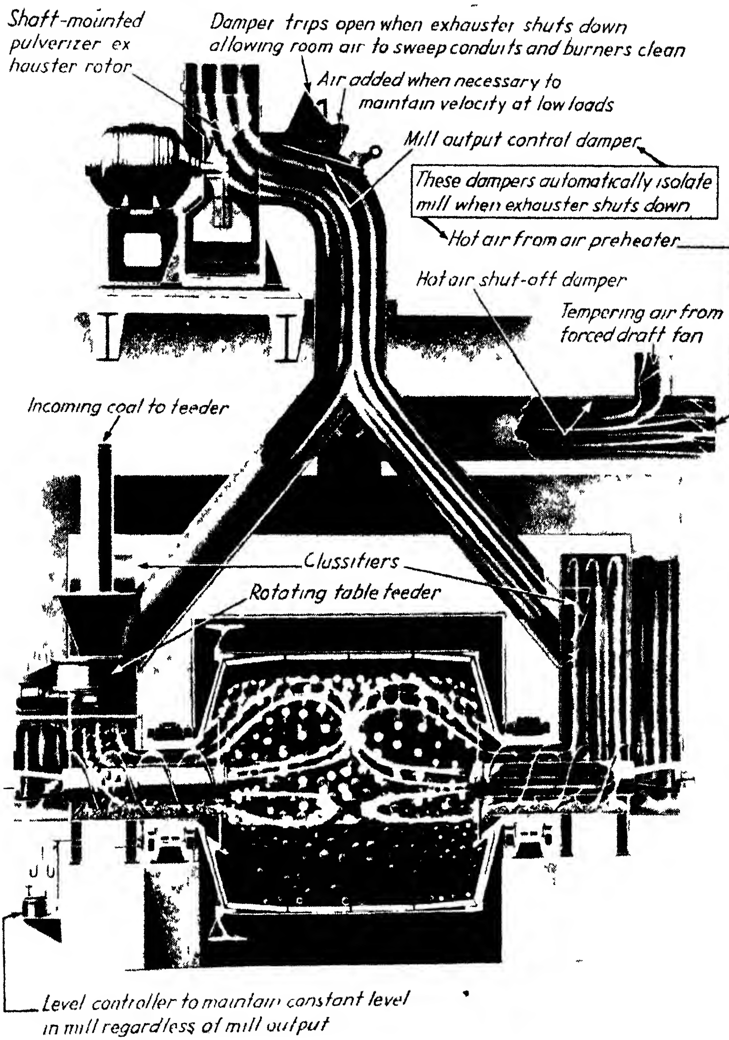


FIG. 216.—Foster Wheeler double classifier ball mill (Hardinge type).

35 kw-hr per ton. Moisture is reduced either by separate driers or by hot air or gases passed through the grinding mill. Hammer types of mills are adapted to grinding high-moisture coals.

Burners for pulverized fuel consist of a central pipe for introduction of the pulverized fuel, surrounded by an air space in

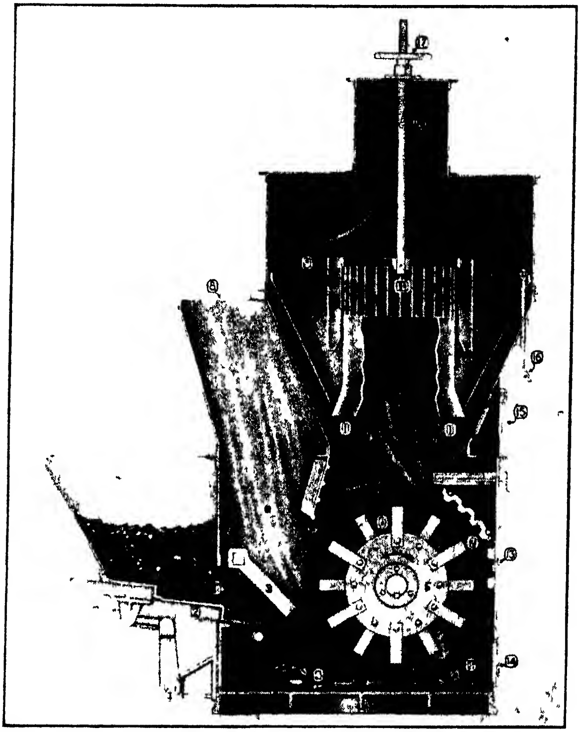


FIG. 217.—Unipulvo mill. (*Strong-Scott Manufacturing Co.*)

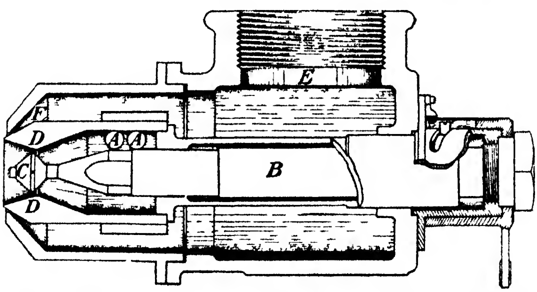


FIG. 218.—Hauck low-pressure air-atomizing burner.

which the air is caused to flow in swirling or turbulent paths by deflectors.

197. Oil and Gas Burners.—In the combustion of fuel oil two requirements are paramount, (1) the oil must be atomized, that is, broken up into very small droplets, (2) the oil fog must be intimately mixed with the correct amount of primary combustion air. Atomization is achieved by (1) air jets, (2) steam jets, or (3) mechanical devices. The heavier oils require heating to increase the fluidity before being delivered to the burner.

Figure 218 gives a sectional view of a low-pressure, air-atomizing type. Oil enters the central tube *B* and escapes through the radial holes *DD* of the double cone *C*. Here air at ½ to 2 lb pressure delivered to *E* enters the two-round tangential holes with a swirling motion through the venturi throat *AA*, thus breaking up the oil into a fine fog. At the tip of the burner the

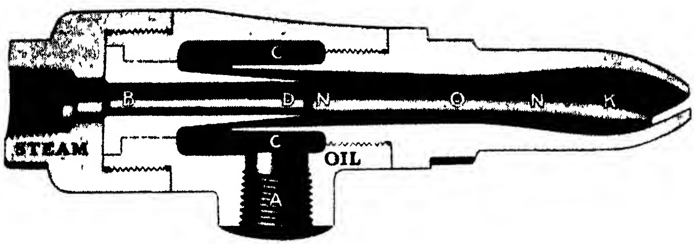


FIG. 219.—National Airoil steam atomizing burner.

primary mixture is met by a second body of air from the annular passage *F* which surrounds the jet. This secondary air can be controlled for accurate proportioning by moving the outer nose in or out.

A steam-atomizing burner is illustrated in Fig. 219. Steam enters at *B* forming a high-velocity jet in the venturi *O*. Oil under low pressure enters the burner at *A* and passes into chamber *C* from whence it is drawn in an annular film by the suction of the steam jet at *N*. Violent agitation in the chamber *K* atomizes the oil. Steam required for atomization may be expected to be 1 to 2 per cent of the boiler output.

Mechanical atomization for low-pressure boilers is sometimes by means of a spinning cup driven by an air impeller. For power-plant boilers mechanical atomizers of the type shown in Fig. 220 are used. Oil under pressure of from 200 to 300 lb and temperatures from 100 to 300°F. to insure fluidity is delivered to the burner-tip body, whence it passes through a ring of holes *B* in the distributor body to the chamber *C* ahead of the disk.

The disk has a series of tangential oil slots *P* which deliver the oil to the swirl chamber *E* and under the high pressure and velocity the oil issues from the tip orifice *F* in the form of a thin hollow cone. This burner is set in a circular register which admits air and distributes it to the flame.

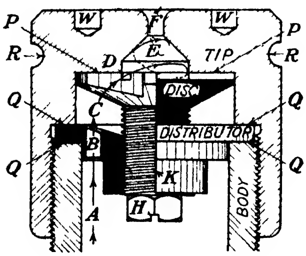


Fig 220.—National Airoil mechanical atomizing oil burner.

High-pressure gas burners for boiler furnaces are of the venturi type. Gas under pressure of from 2 to 20 lb, depending upon the calorific value of the gas, is delivered through a gas orifice spud to the throat of a venturi tube. This draws primary air in through adjustable orifices which mixes thoroughly with the gas. The amount of the primary air determines the length and color of the flame Figure 221 shows a venturi-type burner

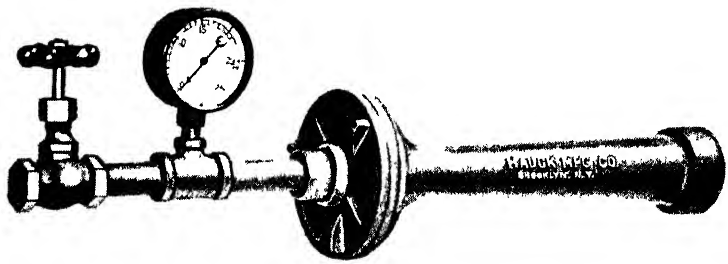


Fig. 221.—Hauck high-pressure gas burner of venturi type

Figure 222 shows a multijet gas burner for use with natural draft.

198. Chimneys.—Chimneys serve the double purpose of removing the combustion gases from the plant and producing the draft necessary for combustion of the fuel. The draft in a chimney is produced by the difference in density of the hot gases inside the chimney and of the air without.

Imagine a plate to be thrown across the lower entrance to a chimney (the ashpit door of Fig. 223) so as to close it *completely* but keep the temperature of the gases within the chimney the same. Then the difference in pressure on the two sides of the plate tending to hold the plate against the opening would be that

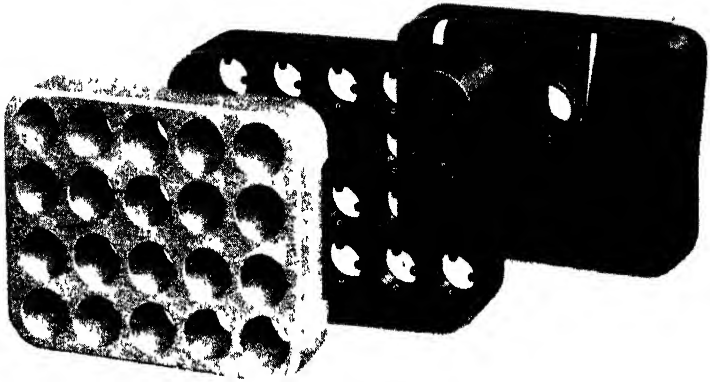


FIG. 222.—Mettler multijet, low-pressure gas burner.

due to the difference in the weights of the gas column within and an air column without of equal height.

Let T_o = absolute temperature of outside air.

T_i = absolute temperature of inside gases.

P = pressure of the atmosphere in pounds per square inch at the top of the stack.

H = height of chimney in feet measured above the ashpit doors.

d = draft pressure in pounds per square foot.

D = draft pressure in inches of water.

w_i = weight of chimney gases per cubic foot at temperature T_i.

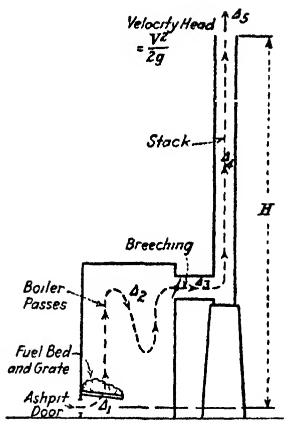


FIG. 223.—Distribution of draft potential in a boiler plant.

w_o = weight of outside air per cubic foot at temperature T_o.

Then the pressure on the chimney side of the plate in pounds per square foot equals

$$p_i = 144P + Hw_i$$

and the pressure on the atmospheric side of the plate in pounds per square foot equals

$$p_o = 144P + Hw_o$$

The slight difference between these two pressures is the draft, or

$$(p_o - p_i) = d = Hw_o - Hw_i \tag{119}$$

Now, from $pv = RT$ and $v = 1/w$, we have

$$w_o = \frac{p_o}{RT_o} \text{ and } w_i = \frac{p_i}{RT_i}$$

and we may assume that the chimney gases have the same R as the air outside, that is, 53.37.

Then, substituting these values of w_o and w_i in Eq. (119), we have

$$d = H\frac{p_o}{RT_o} - H\frac{p_i}{RT_i}$$

and, since p_i is very nearly equal to p_o, we may write

$$\begin{aligned} d &= \frac{Hp_o}{R}\left[\frac{1}{T_o} - \frac{1}{T_i}\right] \\ &= \frac{144HP}{53.37}\left[\frac{1}{T_o} - \frac{1}{T_i}\right] \end{aligned} \tag{120}$$

It is customary to measure drafts in inches of water column. A water column 1 in. high exerts a pressure of $62.4/12 = 5.2$ lb per sq ft. Therefore, dividing Eq. (120) by 5.2, we have, in inches of water,

$$D = 0.52HP\left[\frac{1}{T_0} - \frac{1}{T_i}\right] \tag{121}$$

This equation measures the total gas-moving force in the chimney, furnace, and boiler-pass system. This potential head is all absorbed in friction in the gas passages and in velocity head of the gases, following the ordinary laws of fluid flow. Figure 223 shows the five principal parts into which the total static draft may be considered as divided. Δ_1 is the draft-friction drop or loss through ashpit doors, grate, and fuel bed; Δ_2 is the draft drop through the boiler passes; Δ_3 is the draft drop through the breeching and any turns the gases may make; Δ_4 is the draft drop through the stack; Δ_5 is the draft equivalent of the velocity head of the escaping gases and equals $v^2/2g = h$, in which h is measured as feet of head of the chimney gases that would produce the velocity v.

The factors that influence the performance of a chimney are many and complicated. For example, the temperature inside the chimney depends upon (1) the rate of combustion, (2) the quantity of excess air, (3) the outside temperature, (4) the conductivity of the material of the chimney, (5) the height of the chimney, (6) the kind of boiler and arrangement of passes, (7) the number of boilers in service. At the same time the friction losses vary with (1) the thickness of the fuel bed, (2) the rate of combustion, (3) the quantity of excess air, (4) the kind of boiler and arrangement of passes, (5) the length and area of breeching and stack, (6) the material of chimney, (7) the number of boilers in service. Again the potential draft depends upon (1) the normal barometer and (2) weather conditions.

In addition to these variables, the net draft required to force the air through the fuel bed varies with (1) the kind of grate, (2) the kind and size of the fuel, (3) the rate of combustion.

In the choice of height and diameter, the correct procedure is to choose tentatively these dimensions from experience or by means of a formula such as Kent's

$$\text{Boiler hp} = 3.33E\sqrt{H} \tag{122}$$

which is an empirical formula based on experience. In Eq. (122) E equals the "effective" area of the stack in square feet and equals $(A - 0.6\sqrt{A})$, A being the actual area. Then, with the entire setup of the plant known, performance curves may be calculated and drawn out for different sets of conditions, and, if the calculated performance is not satisfactory, the dimensions should be changed to suit.

Figure 224 shows a set of characteristic curves for a particular small chimney. Curve A is the potential static draft calculated from actual chimney-temperature measurements for the average temperature throughout the height of the stack. Curve B is the draft head necessary for velocity of the stack gases, while the additional ordinates *BC* and *CD* are the friction losses in the gas passages and fuel bed, respectively. These curves were calculated for an outside temperature of 32°F., a barometer of 25 in., and with a single boiler in operation and a particular kind of coal. The curves show that the maximum load that could be carried with these conditions would be that at *E* at which curve *D* crosses curve *A* where the damper is wide open.

It should be pointed out that there would be several combinations of diameters and height that would give the same capacity. The choice among such combinations, then, becomes one of lowest cost.

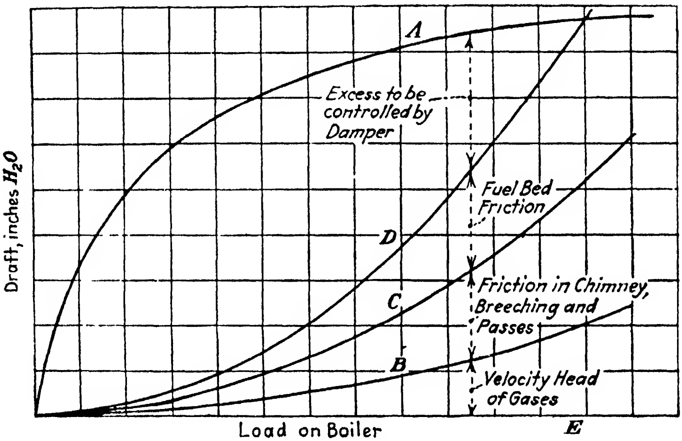


FIG. 224.—Draft in a boiler plant.

199. Mechanical Draft.—The curves of Fig. 224 show that it would be possible to increase the capacity of a plant either by raising the curve *A* by means of a suction fan at the base of the chimney or by forcing the air through the fuel bed by a blast fan (thus taking the burden of the ordinates between *C* and *D* off the chimney). The former arrangement is called *induced* draft; and the latter, *forced* draft. It is, also, possible to use a combination of forced and induced draft.

Practically all stoker-fired plants except the smaller capacities use forced draft. Since an air pressure in the furnace chamber would cause combustion gases to escape into the boiler room, it is customary to have a slight suction above the fire. This suction is kept slight—about 0.1 in. water—in order to prevent excessive infiltration of air into the setting. Such forced-draft fans are part of the stoker installation. The choice of a chimney height and diameter, then, is simplified by not having to deal with the loss through the fuel bed.

Induced-draft fans are usually necessary with air preheaters and economizers because of the additional friction loss imposed by these gas passages and because of the fact that these pieces of equipment reduce the chimney-gas temperature and, consequently, the draft

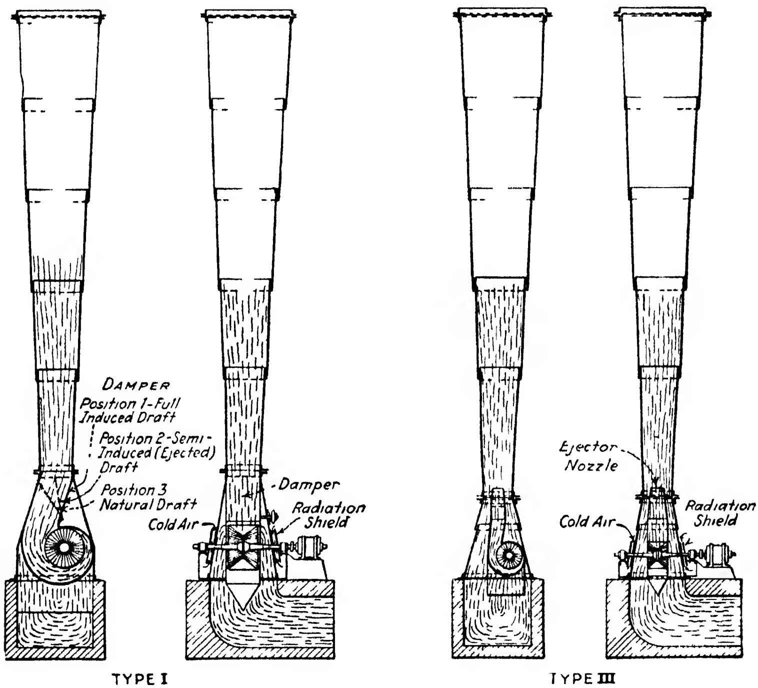


FIG. 225.—Prat-Daniel streamline stacks.

The choice of the relative amounts of chimney draft and induced draft to provide presents a nice problem in economic selection, the cost of the chimney, the value of the heat lost by the chimney gases, initial cost, and operating cost of induced-draft equipment being factors in the problem.

A form of induced-draft equipment that is gaining in favor is the *evasé* (vase-shaped) or venturi stack. Figure 225 shows two types of these stacks, as built by the Prat-Daniel Corporation. Type I uses full induced draft, and the fan handles the entire amount of chimney gas. Type III has a smaller fan and handles only a part of the gas, the remainder being moved by ejector action.

The gradually expanding sides of these stacks increase the area of the stream and, consequently, reduce the velocity so that at the mouth of the stack the gases have a very low velocity.

200. Automatic Combustion Control.—Since the demand for steam from most boilers is not constant but varies continuously, and often suddenly, it is impracticable even in small plants to rely upon manual control. While steam demand is the primary variable imposed upon the boiler, this is reflected promptly in variations in boiler pressure. Boiler pressure, then, can be conveniently made the primary variable in automatic control of the plant.

The dependent variables in control are:

1. *Feed-water Flow.*—This is kept proportionate to the steam flow leaving the boiler by regulators actuated by changes in water level (see Art. 209), and is usually not included in the combustion-control system.

2. *Fuel-Combustion Rate.*—To care for an increased steam flow, means must be provided for increasing the heat input to the furnace. This must be begun promptly, since some time lag is inevitable, especially with coal firing. With drop in demand the reverse action with equal promptness is required. This involves the fuel-feeding devices, whether coal, oil, or gas is fired.

3. *Air Flow.*—If combustion efficiency is to be maintained, air flow must follow fuel flow closely. This involves the forced-draft fans or dampers.

4. *Combustion Products Flow.*—As combustion is increased, a greater volume of flue gases must be eliminated. This involves chimney dampers and induced-draft apparatus.

In small plants a simple, three-point system, controlling (1) stoker speed, (2) fan speed, and (3) uptake damper opening in an all-on and all-off arrangement, has been found successful. Such a system is not properly called "automatic" control because considerable manual adjustment to the general load level is required, and this depends upon the experience of the operator. A modification of this system is to make the action of the regulator proportionate to the pressure change.

Also the equipment must be capable of making small adjustments in proportion to load changes instead of going to extreme positions. In most systems a master controller actuated by variations in steam pressure opens or closes a pilot valve which admits air or hydraulic pressure to the operating mechanism, or opens or closes electric circuits which operate the control devices.

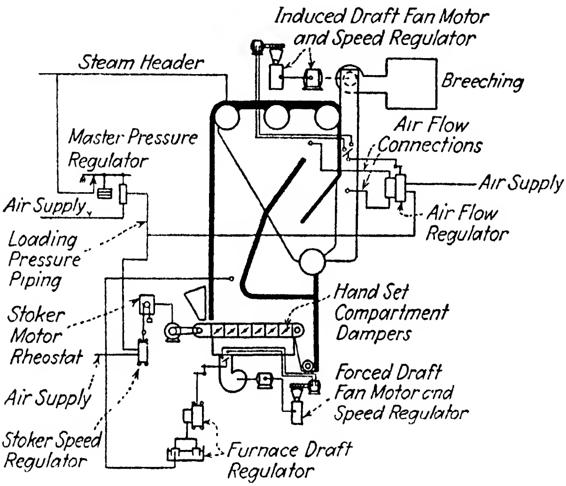


FIG. 226.—Typical combustion-control system.

Figure 226 shows diagrammatically the arrangement of a combustion-control system applied to a traveling grate stoker with electric drive on stoker and fans. A master regulator is connected to the steam header which controls the air supply sent to the stoker motor-speed regulator and to the air-flow regulator on the uptake damper. Air flow is also controlled by the furnace-draft regulator on the forced-feed fan damper, actuated by the pressure above the fire. If the air-flow damper and the uptake damper move to the limits of their rotation, they make contacts which change the speed of the fans. The air-flow regulator is actuated by the friction drop in the last boiler pass.

Problems

1. A coal gave the following proximate percentage analysis: moisture, 2.92; volatile, 32.04; fixed carbon, 58.23; ash, 6.81. Express these percentages on a "moisture-free" basis. How much water is mixed with each pound of dry coal?

2. A Pennsylvania coal gave the following proximate percentage analysis: moisture, 0.9; volatile, 21.59; fixed carbon, 68.49; ash, 9.02. Its heating value was 14,060 Btu per lb "as received." Calculate (*a*) the heating value on a "moisture-free" basis; (*b*) the weight of moisture held by each pound of dry coal.

3. A coal gives the following ultimate percentage analysis reported on an as-received basis: C, 58.16; H, 5.65; 0, 19.89; N, 1.04; S, 1.77; moisture 13.56. Heating value 10,771 Btu. Calculate (*a*) the ultimate analysis on a dry basis; (*b*) the heating value on a dry basis; (*c*) the moisture held by each pound of dry coal. (*d*) Check the heating value by Dulong's formula.

4. Given the following percentage data for a coal on the as-received basis: moisture, 11.35; volatile, 34.62; fixed carbon, 40.63; ash, 13.40. Heating value 10 733 Btu per lb. Calculate the heating value on (*a*) a moisture-free basis; (*b*) a moisture-and-ash-free basis.

5. A coal gave the following ultimate percentage analysis reported on an as-received basis: C, 78.85; H, 2.52; O_2, 5.87; N, 0.77; S, 0.46; moisture, 3.43. Heating value, 12,782 Btu per pound. Calculate (*a*) the ultimate analysis on a moisture-free basis; (*b*) the heating value per pound dry. (*c*) Compare this heating value with that obtained by using Dulong's formula

6. A lignite gave the following proximate percentage analysis: moisture, 35.96; volatile, 31.92; fixed carbon, 24.37; ash, 7.75. Heating value on as-received basis 7069 Btu per lb. Calculate the heating value on (*a*) the moisture-free basis; (*b*) the moisture-and-ash-free basis. (*c*) Determine the weight of moisture held by each pound of dry coal.

7. A chimney is 100 ft high. The normal barometer is 28.8 in. If the mean temperature inside the chimney is 500°F. and the outside temperature is 75°F., what is the potential draft at the base of the chimney in inches of water?

8. The largest chimney in America, at Anaconda, is 585 ft high and 60 ft in diameter inside. With furnace gases inside at a temperature of 300° and atmospheric temperature of 30° below zero, what is the potential draft at the base of the stack? Normal barometer at Anaconda is 24.5 in.

9. A chimney 225 ft high at sea level receives flue gases at a mean temperature of 450°F. The outside temperature ranges from 0 to 96°F., and the barometer ranges from 29.75 to 30.25 in. Calculate the potential draft with (*a*) a 30-in. barometer and outside temperatures of 0, 22.5, 45, 67.5, and 90°F.; (*b*) an outside temperature of 45°F. and barometric pressures of 29.75 30. and 30.25 in. (*c*) Plot two curves with drafts as ordinates against Fahrenheit temperatures and barometric heights as abscissas.

10. In a certain plant, the mean chimney temperatures at four rates of coal consumption are as follows:

Coal burned per hr., lbs....................	2,000	4,000	6,000	8,000
Mean chimney temperatures, deg. F........	475	495	535	580

The height of the chimney is 125 ft, the barometer 28 in., and the outside temperature 70°F. (*a*) Calculate the potential draft at each rate of combustion and (*b*) plot drafts against coal consumption.

CHAPTER XIX

BOILER ACCESSORIES AND AUXILIARIES

201. Definitions.—*Boiler accessories* might be defined as those pieces of apparatus or parts of a boiler which are necessary for safe and convenient operation, such as those listed in describing the elementary boiler of Art. 174. Some of those parts, called "accessories," are required by law, and by the A.S.M.E. Boiler Code, while others are for the purpose of improving performance.

Fig. 227.—Crosby pressure-gage mechanism, double-tube type.

Boiler auxiliaries are certain pieces of apparatus which are used in connection with a boiler and which contribute to the general performance of the boiler plant but are not a part of the boiler; for instance, boiler feeders, feed-water heaters, evaporators, and draft equipment. Superheaters and economizers, although a real part of the boiler, are, also, classed as auxiliaries.

202. Pressure Gage.—Figure 227 shows a type of gage that is almost universally used for steam-pressure measurement. The working element of this instrument is the Bourdon tube, which is a circular tube of oval cross section. The gage of Fig. 227 has a double-tube mechanism. The pressure inside the tubes causes them to tend to straighten out and, in so doing, moves the needle over the scale, acting through the lever and gear train. Since the stress-strain curve of the material of the tube varies with its temperature, it is important to prevent steam from entering it, and for this purpose a loop should always be placed in the connection to the steam pipe or boiler. This loop, or "siphon," traps water and thus keeps the steam out of the tube. The best gages have the tube, lever, and needle mechanism built as a unit, so that strains in the case will not disturb the calibration.

203. Water Columns.—The gage glass for showing the height of the water level in a boiler is connected to a small chamber, called a *water column*, which is, in turn, connected to the boiler. Figure 228 shows a water column with the gage glass attached, the glass in this case being inclined to permit of easy observation from the floor. In case of breakage of the glass, small check valves in the fittings at the ends close and prevent escape of steam and water. If the check valves fail, the two chain-operated valves may be closed from the floor

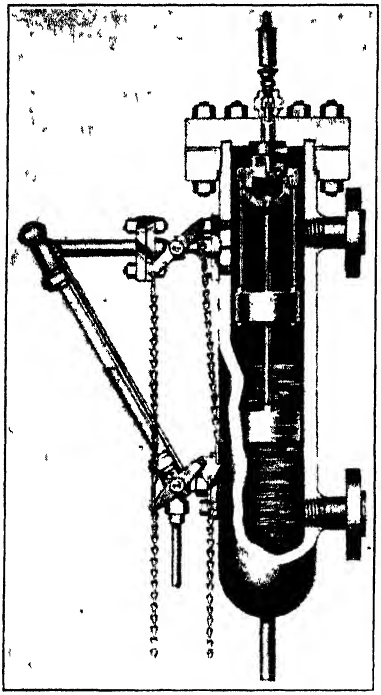


Fig. 228.—Yarway floatless hi-lo alarm water column.

This water column has a high- and low-water alarm whistle operated by the solid displacement weights. One objection to the use of the alarm is that the fireman becomes accustomed to depend upon it, and, if it fails, low water may occur without his knowing it. Frequent blowing down of the column by means of valves placed at the bottom ends of the two pipes projecting downward keeps the passages clear and shows the condition of the alarm.

In case of breakage of the glass, the water level is determined by the three *try cocks* on the side of the column, which may be seen in the boiler illustrations of Chap. XVII.

204. Feed-water Entrance.—The feed water should enter the boiler through a pipe long enough to raise the temperature of the feed water to very nearly that of the water in the boiler before being discharged from the pipe. In H.R.T. boilers the feed pipe may enter either at the front end or at the top, although the feed is sometimes introduced through the blowoff connection.

The feed should be introduced into the coolest part of the boiler and in such a way as to assist the circulation if possible The feed line must have a check valve, and there must be a stop valve between the check and the boiler.

205. Blowoffs.—The blowoff from the lowest part of the boiler requires a very carefully built valve because the sediment and scale will quickly cut out the seats of ordinary valves and cause them to leak Even a very small leak running continuously

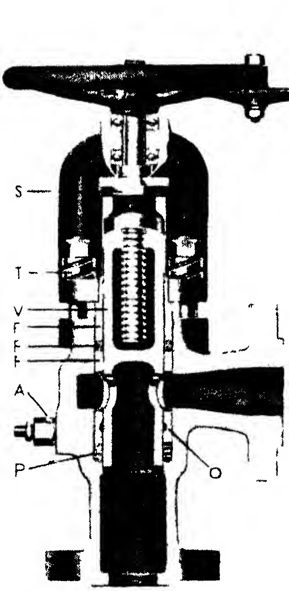


Fig. 229.—Yarnall-Waring blowoff valve.

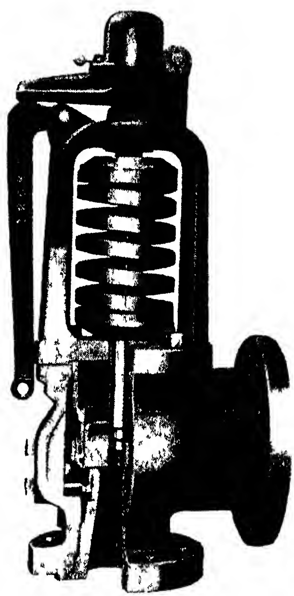


Fig. 230.—Crosby safety valve

represents a large waste of heat, and it will not remain small. Figure 229 shows one form of a successful blowoff valve. This valve has no seats but, in its closed position, is sealed by having the packing *P* compressed around it by the upper sleeve *F* when the hand wheel forces the collar *S* down against *F*.

206. Safety Valves.—Figure 230 shows a spring-loaded *pop* safety valve. When the pressure becomes sufficient to lift the valve off its seat against the pressure of the spring, the escaping steam will be deflected downward by the lips on the valve, and this will cause the valve to remain open with slightly less boiler pressure than was required to open it in the first place. Then, when the valve starts to close, it will close quickly. The lever at the left is for trying the valve. The inspector adjusts the valve by means of the sleeve nut at the top and then seals it with the seal at the top left.

The following requirements are quoted from the A.S.M.E. Boiler Construction Code (1937 edition).

P-269 *Safety-Valve Requirements.*—Each boiler shall have at least one safety valve, and if it has more than 500 sq ft of water-heating surface, or the generating capacity exceeds 2000 lb per hr, it shall have two or more safety valves . . .

P-270.—The safety-valve capacity for each boiler shall be such that the safety valve or valves will discharge all the steam that can be generated by the boiler without allowing the pressure to rise more than 6 per cent above the highest pressure at which any valve is set and in no case to more than 6 per cent above the maximum allowable working pressure. The maximum steaming capacity of a boiler shall be determined by the manufacturer and shall be based on the capacity of the fuel-burning equipment, on the air supply, draft, etc.

P-271.—One or more safety valves on the boiler proper shall be set at or below the maximum allowable working pressure. If additional valves are used the highest pressure setting shall not exceed the maximum allowable working pressure by more than 3 per cent. The complete range of pressure settings of all the valves on a boiler shall not exceed 10 per cent of the highest pressure to which any valve is set.

P-272.—All safety valves shall be so constructed that the failure of any part cannot obstruct the free and full discharge of steam from the valve. Safety valves shall be of the direct spring-loaded pop type, with seat inclined at any angle between 45 and 90 deg, inclusive, to the center line of the spindle. *The maximum capacity of a safety valve shall be determined by actual steam flow in the presence of authorized inspectors at a pressure of 3 per cent in excess of that at which the valve is set to blow,* . . . and credited with 90 per cent of the flow developed . . .

Dead-weight or weighted-lever safety valves shall not be used.

P-273.—Each safety valve shall be plainly marked by the manufacturer in such a way that the markings will not be obliterated in service. The marking may be stamped on the casing, or stamped or cast on a plate or plates securely fastened to the casing, and shall contain the following markings:

a. The trade-mark of the manufacturer.

b. Manufacturer's design or type number.

c. Size ______ in. Seat diameter ______ in.

d. Pres. _ _ lb.
e. B.D. _ _ _ lb.
(Blowdown—difference between the opening and closing pressure.)
f. Cap _ ____ _ _ lb per hr.
(In accordance with Pars. *P*-272 and *P*-281, and with the valve adjusted for the blowdown given in the preceding item.)
g. Capacity lift _ in.
(Capacity lift--distance the valve seat rises under the action of the steam when the valve is blowing under a pressure of 3 per cent above the set pressure.)
h. A.S.M.E. symbol.

207. Fusible Plugs.—To protect boilers carrying not over 250 lb. pressure from overheating in case of low water, plugs

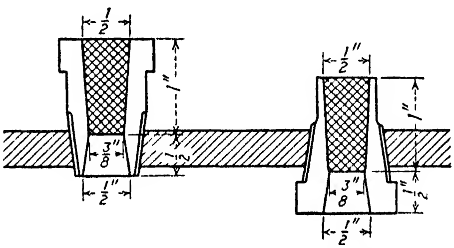


FIG. 231.—Typical fusible plugs (1937 *A.S.M.E. Code*.)

similar to those shown in Fig. 231 are installed in the direct passage of the hot gases just below the safe water level in the rear head of an H.R.T. boiler and in the bottom of the water and steam drum of a water-tube boiler. These plugs are filled with tin, which melts when not covered by the water, and the escaping steam warns the fireman.

The specifications of the A.S.M.E. Boiler Code require that the plugs be of general form of those shown in Fig. 231, the dimensions given being the minima. Tin of at least 99.3 per cent purity, with melting point between 400 and 500, must be used.

208. Automatic Stop Valves.—Whenever two or more boilers are operated in parallel on the same header, it is important that some means should be provided for automatically and quickly cutting a boiler off the header, in case one of its tubes should blow out or some other large leak should develop, in order to prevent steam from flowing into the defective boiler from the others. Figure 232 shows one form of such an automatic nonreturn stop valve, the left-hand flange of which is connected to the boiler nozzle and the other to the header. In case of a sudden reduction in pressure below the valve, the steam pressure in the dashpot above it would cause the valve to close.

When a boiler is to be cut into a header, the stem of the valve is screwed up, and then, when the pressure in this incoming boiler becomes slightly higher than that in the header, the valve will be

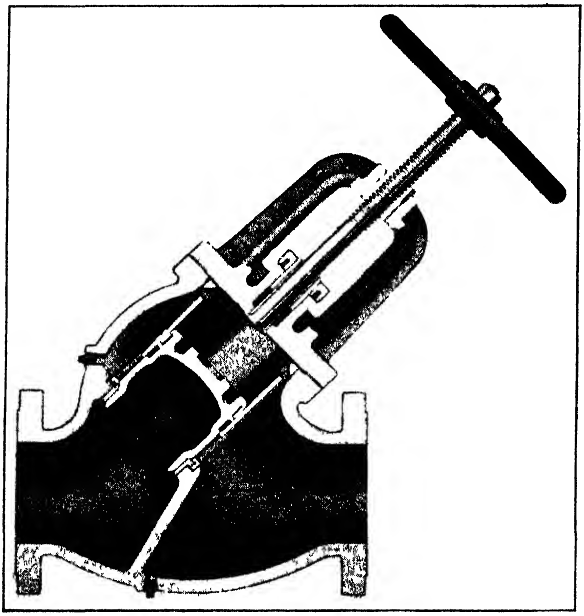


FIG 232.—Crane automatic stop-check valve.

forced up, and the boiler will deliver steam. In case a boiler is open for repairs or cleaning, it is impossible for anyone to turn steam into the boiler from the header, because the valve stem cannot lift the valve.

209. Feed-water Regulators.—With a varying demand for steam from a boiler, a constant water level cannot be maintained with the boiler feeder operating at a uniform rate. The reading of the gage glass, moreover, is not a true indication of the height of the water level, because violent ebullition at heavy loads tends to cause the glass level to be too high. Automatic control of the rate of feeding is, therefore, highly desirable. Automatic feed-water regulators are made in several different types, three of which are shown in Fig. 233. In type (*A*) the float chamber is connected to the steam-water drum by means of the two pipes at the right. If the water level falls, the float falls and raises the balanced valve in the feed line allowing increased flow to the boiler. Since the entire mechanism is under boiler pressure, the fit at the valve stem can be made loose and practically frictionless.

In type (*B*) the two pipes from the steam-water drum connect to a sloping tube, the upper end of which is filled with steam and

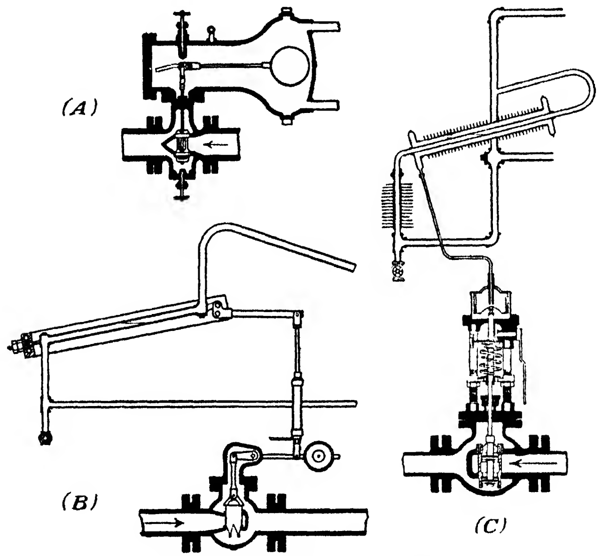


Fig. 233.- Feed-water regulator types.

the lower with water. This water, being separated from the boiler, becomes cool. Now, if the water level falls, the tube will contain more steam, and its average temperature will be higher. This increases the length of the tube which pushes against the short leg of the L-shaped lever, thus lowering the connecting link and opening the balanced valve in the feed line.

Type (*C*) has a similar sloping tube connected to the steam-water drum. Surrounding the slanting tube is a water jacket, and the bottom of the jacket is connected through the small pipe to the diaphragm-operated, balanced valve in the feed line, the water flowing in the direction of the arrow. If the water level is low, the steam in the upper end of the slanting tube will occupy a large portion of the length of the tube, and this will heat the jacket water, thus expanding it and causing it to open the diaphragm valve, allowing more water to flow into the boiler. With high water in the boiler, the reverse action will take place. The fins on the jacket are to cool the jacket water quickly when the water in the central tube rises.

Since the water under pressure in the feed line is not expansible, it is necessary to increase the speed of the feed pump to obtain

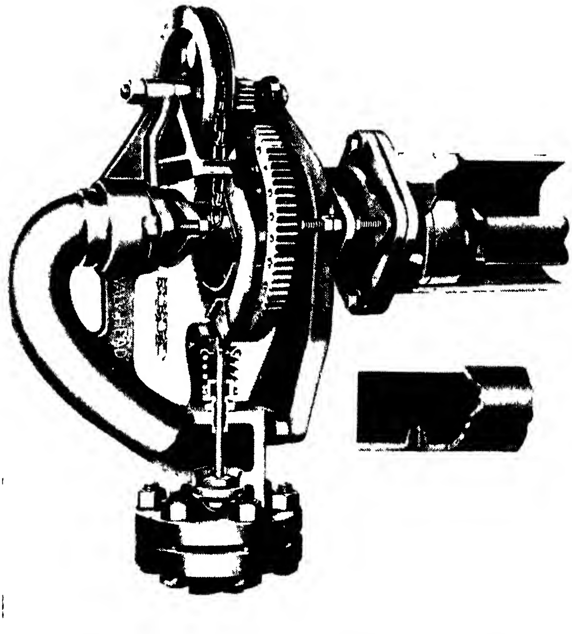


FIG. 234.—Diamond "Valv-in-Head" soot blower.

increased flow to the boiler. As the balanced valve is opened, a friction drop in the pipe line causes a slight drop in pressure at the pump, and this pressure drop works through a *governor* to increase the speed of the pump.

210. Soot Blowers.—The insulating value of carbon and dust deposits that collect on the fire side of boiler tubes is very high, even for very thin films. It is important, therefore, to remove this soot deposit frequently, and for this purpose *soot blowers* are used. For small boilers hand steam lances are used; but for large installations, the blowers are built in place as fixed equipment.

Figure 234 shows the operating principle of the Diamond soot blower for a water-tube boiler. The blower element consists of a steel pipe with nozzles (shown enlarged in insert) so spaced as to line up with the boiler tubes. When the chain is pulled, the gear is revolved, and the cam depresses the steam valve, thus admitting steam to the nozzle pipe. By revolving this pipe by a chain over the sheave wheel while the steam is blowing, the soot is blown from all the tubes within its range There are several blowers arranged at strategic points in the boiler passes, the number depending upon the size and type of boiler.

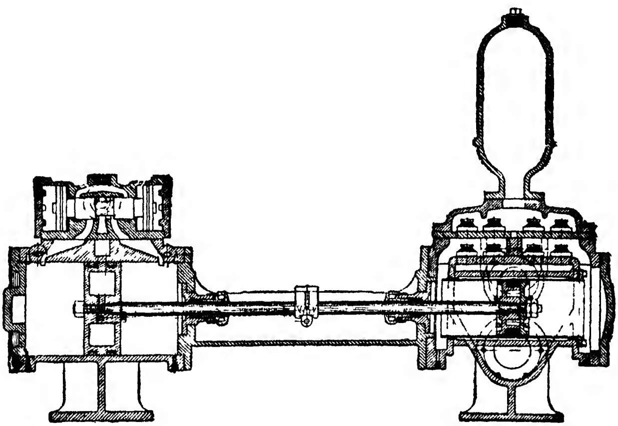


Fig 235 — American-Marsh simplex steam pump

211. Reciprocating Boiler-feed Pumps.—The feed water may be forced into a boiler by (1) gravity or hydrant pressure for low-pressure boilers, (2) direct-acting steam pumps, (3) motor-driven plunger pumps, (4) motor- or turbine-driven centrifugal pumps, or (5) injectors.

For small, high-pressure boilers, the direct-acting reciprocating pump is generally used. Such pumps may be either simplex, in which a single steam cylinder operates a single water cylinder, or duplex, in which there are two steam cylinders and two water cylinders. Figure 235 shows a typical simplex pump, the chief characteristic of which is the mechanism by which the piston is made to reverse at the end of the stroke. The dotted line behind the piston rod is a connecting rod which is fastened to a crosshead at the middle of the piston rod and to a rocker arm pendant from the valve chest. The motion of the rocker arm operates a small pilot valve which admits steam to one end of the piston valve in the steam chest, and the movement of this second valve admits steam to the main cylinder and exhausts the steam therefrom. The length of the stroke is controlled by an adjustment on the movement of the pilot valve.

Some simplex pumps have pilot valves operated by steam, controlled by the movement of the main piston. One disadvantage of the simplex pump is that it may be stalled near the end of the stroke.

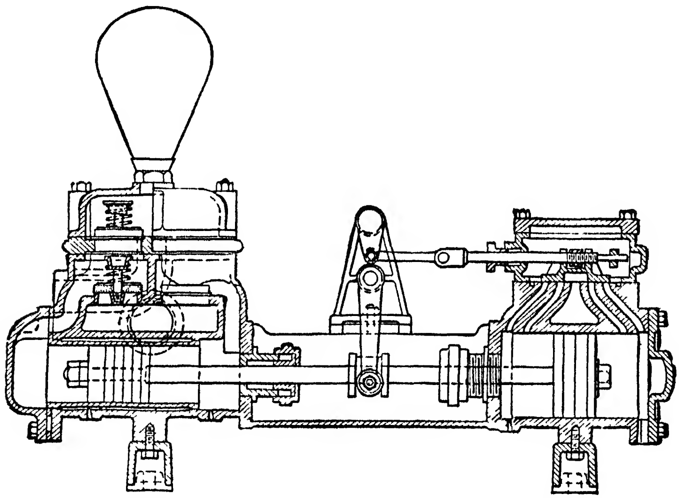


FIG. 236. Sectional view of duplex steam pump.

Figure 236 shows a section through a duplex pump, the operating principle of which is that the movement of the piston in one cylinder operates the valve of the other cylinder. The valves have considerable lost motion on their stems, which allows one piston to complete about half its stroke before it operates the valve of the other cylinder. To stop the piston before it reaches the cylinder head, a second steam port is arranged so as to trap steam at the end of the stroke.

Since the cylinder receives steam at full pressure for the entire stroke, the pressure of the exhaust is at high pressure and contains nearly its initial quantity of heat. For this reason, the steam consumption of direct-acting pumps is very high per horsepower-hour, from 100 to 300 lb (see Art. 119).

212. Centrifugal Feed Pumps.—Multistage centrifugal pumps are used for boiler feeding in the larger plants. Centrifugal pumps have the advantages of simplicity, compactness, absence of valves, and of delivering the feed without the pulsations or the pounding of direct-acting pumps, although the steam consumption is about the same as for direct-acting pumps and their initial cost is greater. These pumps are driven by small turbines or electric motors.

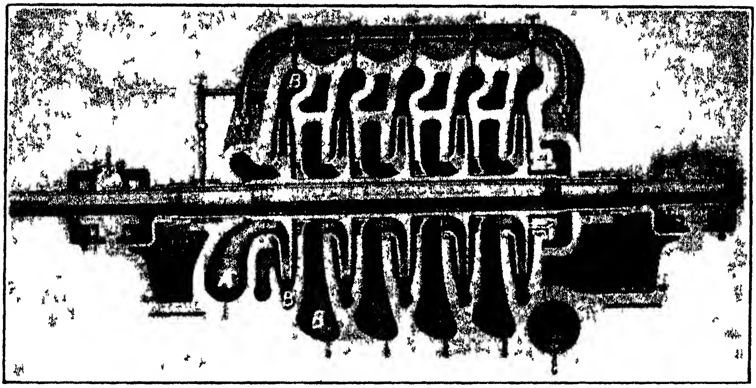


FIG. 237. Worthington five-stage centrifugal pump.

Figure 237 shows a sectional view of a Worthington five-stage centrifugal pump. In this pump the water is lifted through the suction inlet *A* by the first stage impeller and discharged into an annular ring *B* around this impeller at high velocity. Since the pump is to deliver at a high pressure, this velocity energy is transformed into pressure head by the passing of the water through diffusion passages with gradual increase in area, thus reducing the velocity. The second-stage suction is the delivery *B* from the first stage, and, likewise, for the succeeding stages.

213. The Steam Injector.—A very simple and efficient boiler feeder for small stationary boilers and for locomotive boilers is the injector, one form of which is shown in Fig. 238. The principle of operation of the injector is as follows:

When steam is admitted under boiler pressure through tube *R*, it passes through tube *S*, taking with it the air in chamber *F* and in the water-supply

pipe, and then escapes through the overflow valve P. The air being eliminated from F, a partial vacuum is created into which the supply water is forced by atmospheric pressure, and this water, meeting the steam jet at the top of tube S, is carried with it, partially condensing the steam.

When the proper proportions of steam and water are admitted, the mixture passes through tubes S and Y with high velocity. The flow through the tube Y creates a partial vacuum at the top of the tube which closes the ring disk T against the bottom of tube S. It also creates a partial vacuum in chamber G which closes the overflow valve P. As the mixture passes through S and Y with high velocity, the water more fully condenses the steam. In the diverging lower part of Y and in chamber O

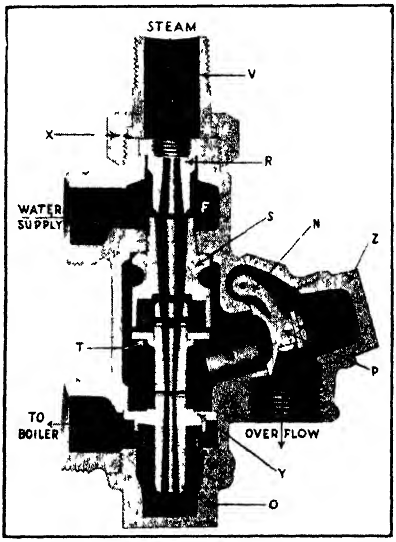


FIG. 238.—Penberthy automatic injector.

the velocity is largely converted to pressure, and the mixture, now practically all water, passes on into the boiler with comparatively low velocity and high pressure.

The injector possesses the disadvantage of being unable to handle warm water and, therefore, is used principally in small plants which use cold feed water. It is true that an injector is itself a feed-water heater, but it heats the water with steam drawn from the boiler, so that there is no saving or gain in the process. If hot water is to be fed, it must be taken from a tank above the injector.

214. Impurities in Boiler-feed Water.—Water is a powerful solvent, and in nature is seldom found free from impurities that are important in steam plants. The impurities most commonly found in waters available for boiler feed are

1. Suspended matter—silt, mud, or vegetable matter.
2. Bicarbonates of calcium and magnesium.
3. Sulphates of calcium and magnesium.
4. Chlorides of calcium and magnesium.
5. Carbonates of sodium and potassium.
6. Sulphates of sodium and potassium.
7. Chlorides of sodium and potassium
8. Air—oxygen and nitrogen.
9. Carbon dioxide.
10. Oil.

Hardness of water refers to the concentration of salts of calcium and magnesium. These react with soap to form a curd of insoluble calcium and magnesium soap. It is a common practice to reduce the various calcium and magnesium compounds to equivalent concentration of calcium carbonate, $CaCO_3$, to simplify water-treating calculations. *Temporary hardness* is due to bicarbonate of calcium and magnesium which may be removed by simple boiling. This is due to the following reversible reaction which moves toward the right with the application of heat

$$Ca(HCO_3)_2 \leftrightarrows CaCO_3 + CO_2 + H_2O \tag{A}$$

the calcium carbonate being precipitated and the carbon dioxide leaving as gas. *Permanent hardness* is due to sulphates which cannot be removed by simple boiling. However, the solubility of the sulphates is reduced by increasing the temperature as shown by the graph of Fig. 239.

The concentration of salts in water is expressed as "parts per million" (ppm) by weight or as "grains per gallon" (gpg) Grains per gallon multiplied by 17.12 gives parts per million.

215. Effects of Impurities in Boiler Water.—1. Suspended matter, vegetable substances, oil and high concentration of the soluble sodium and potassium salts may cause *foaming* due to minute bubbles of steam that do not break and subside. This may cause *priming*, that is, carrying over of water into the steam outlet of the boiler. Priming may cause serious damage to engines or fouling of superheaters, steam lines, and turbines.

2. Hardness in feed water causes *scale* to form on boiler surfaces thus reducing the heat-transfer rate of the steel. In general,

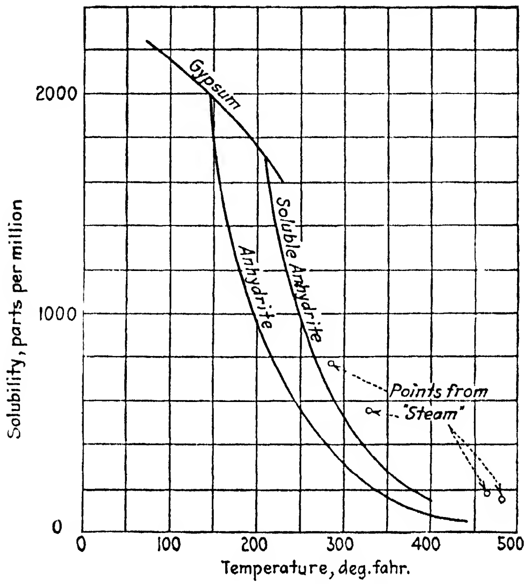


FIG. 239.—Solubility of calcium sulphate in water. (*R. E. Hall.*)

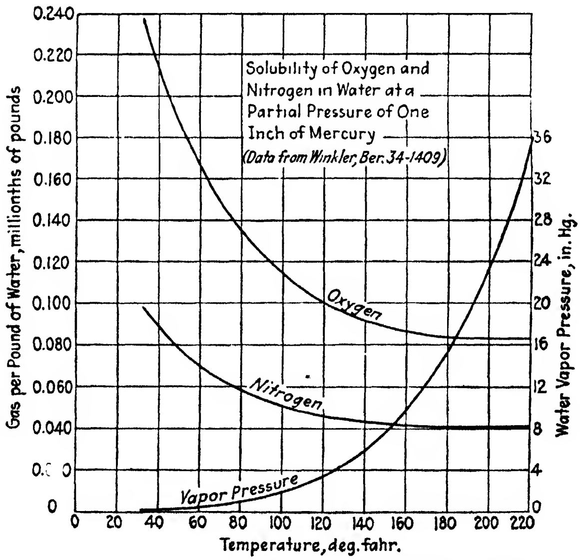


FIG. 240.—Solubility of oxygen and nitrogen in water.

carbonates form a soft scale while sulphates form a scale that cements firmly to the boiler. A thin but flintlike and hard-to-remove scale is formed by waters containing silicates. Scale in water-tube boilers is removed by means of a water-driven turbine cleaner run through from a drum or header. Scale on the water side of fire tubes is removed by a rotating hammer which jars the layer loose.

3. *Corrosion* in boilers may result from the presence of (1) free oxygen or (2) free acid. Natural water in contact with air is saturated with oxygen and nitrogen. Figure 240 shows the solubility of these gases in water at various temperatures for a partial gas pressure of 1 in. Hg. The following example illustrates the use of this chart:

Example A.—Find the weights of oxygen and nitrogen which water at 120°F. is capable of absorbing from the atmosphere at a barometric pressure of 28 in. Hg.

SOLUTION.—From the vapor-pressure curve of Fig. 240 the vapor pressure at 120°F. is 3.5 in. Hg. The partial pressure of the air in contact with the water surface at 120°F. is $(28 - 3.5) = 24.5$ in. Hg.

The partial pressures of the O_2 and N_2 are, respectively,

$$(24.5 \times 0.209) = 5.12 \text{ in. Hg and } (24.5 \times 0.791) = 19.36 \text{ in. Hg}$$

From the chart, Fig. 240, at 1-in. Hg partial pressure 1 lb water at 120° will dissolve (0.1×10^{-6}) lb O_2 and (0.046×10^{-6}) lb N_2. Then the weights of these gases absorbed per pound of water, under the conditions of the problem, will be

$$(0.1 \times 10^{-6})5.12 = 0.512 \times 10^{-6} \text{ lb } O_2$$

and

$$(0.046 \times 10^{-6})19.36 = 0.891 \times 10^{-6} \text{ lb } N_2$$

It has been found that, if feed water has a pH value above 9, it will have no tendency to corrode the boiler. The source of acidity in a boiler water may be (1) carbon dioxide which becomes H_2CO_3 in solution, (2) chemical treatment which lowers the pH value.

4. *Caustic embrittlement* is a term applied to the appearance of intercrystalline cracks in the boiler metal in the regions of concentrated stresses near riveted joints. It has been found to occur in boilers in which a low ratio of sodium sulphate to sodium carbonate concentration is present. The A.S.M.E. Boiler Code Committee has recommended the following ratios which have been found safe from caustic embrittlement:

Boiler Gage Pressure	Ratio of Na_2SO_4 to Equivalent Na_2CO_3
Up to 150	1:1
150 to 250	2:1
Over 250	3:1

Recent research[1] casts some doubt on the protection afforded by the maintenance of these ratios.

5. A high concentration of soluble salts within a boiler may result in the *carrying over* of droplets of the solution into the superheater and steam lines. When the water is evaporated from these droplets a dry powder remains which gradually builds up a scale deposit in pipes, valves, and turbine blading. Figure 241 shows such a deposit nearly 2 in. thick found in a 16-in. pipe carrying superheated steam.

FIG. 241.—Carry-over deposit in a superheated-steam main.

216. Methods of Feed-water Treatment.—The method used to condition boiler feed water will depend upon the nature of the impurities in the raw water, the quantity of make-up required, and the economic relationships in the plant. One or more of the following may be employed in a particular plant:

1. *Heating* to around 200°F. to remove air and bicarbonates and some of the sulphates.

2. *Chemical treatment* of feed water is accomplished by adding chemicals which precipitate the objectionable substances and leave in the water highly soluble but harmless salts. The lime-and-soda process, which is most commonly used, consists of the addition of slacked lime (calcium hydroxide) and soda ash (sodium carbonate) in calculated amounts and proportions, determined by the water analysis. The process may be hot or cold, the hot process being more rapid and requiring smaller excess of reagents. Figure 242 shows the details of the equipment for a continuous hot lime-soda softener. The raw water enters the

[1] A.R.E.A. Bulletin 404, June–July, 1938.

sedimentation chamber near the top where it comes in contact with the heating steam and the chemicals. The chemicals are introduced in proportion to the raw water flow by a proportioner controlled by an adjustable orifice. The treated, deaerated water is drawn off through the uptake funnel, the treated water chamber, and then is passed through the filter

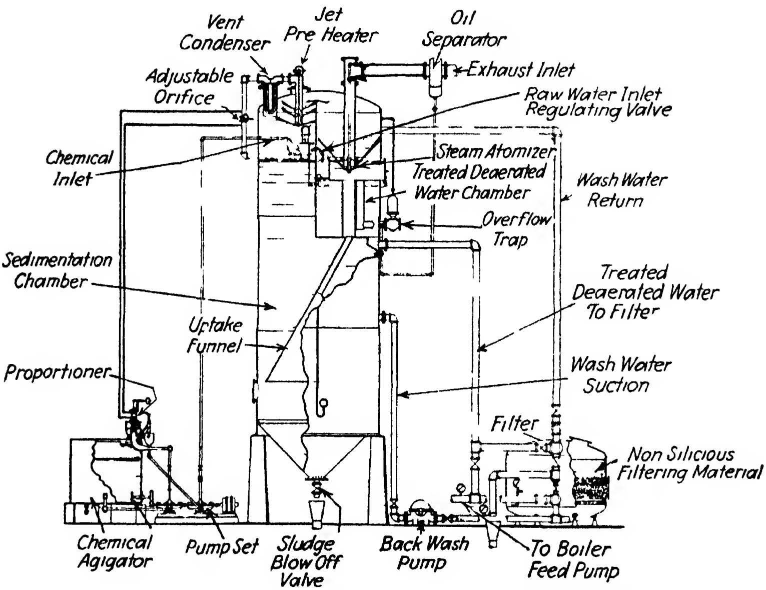


FIG 242 —Cochrane hot lime-soda, deaerating heater and softener

The type reactions for the lime-soda process are as follows

$$\underset{\text{Calcium bicarbonate}}{Ca(HCO_3)_2} + \underset{\text{Calcium hydroxide}}{Ca(OH)_2} \rightarrow \underset{\text{Calcium carbonate}}{2CaCO_3\downarrow} + \underset{\text{Water}}{2H_2O} \qquad (B)$$

$$\underset{\text{Calcium sulphate}}{Ca(SO_4)} + \underset{\text{Sodium carbonate}}{Na_2CO_3} \rightarrow \underset{\text{Calcium carbonate}}{CaCO_3} + \underset{\text{Sodium sulphate}}{Na_2SO_4} \qquad (C)$$

Other reagents sometimes used include caustic soda (sodium hydroxide), sodium aluminate, and trisodium phosphate.

3. *Zeolitic* water softening consists of passing the water supply through a tank containing a bed of "zeolite" which is a hydrated sodium aluminum silicate. The calcium and magnesium of the

soluble salts in the water exchange with sodium in the zeolite thus form calcium and magnesium zeolites, the soluble sodium carbonate, bicarbonate, and sulphate passing on into the boiler feed. The type reaction is

$$Na_2Z + CaSO_4 \rightarrow CaZ + Na_2SO_4 \qquad (D)$$

When the zeolite has lost its efficacy, it is regenerated with a solution of common salt, during which process the calcium zeolite is restored to the active sodium zeolite (Na_2Z).

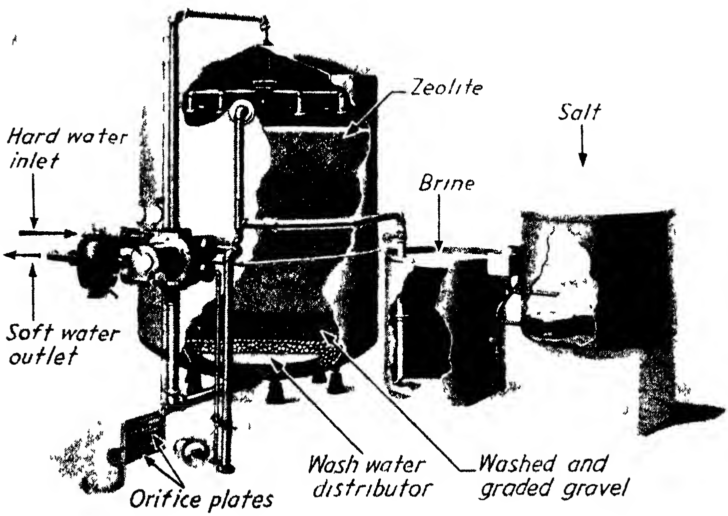


FIG. 243.—Permutit zeolitic down-flow softener

Figure 243 illustrates a typical zeolitic softener.

A disadvantage of all chemical treatments is that the calcium and magnesium alkalinity is simply changed to the equivalent sodium alkalinity and the total dissolved solid content is not reduced. Recently a hydrogen zeolite[1] has made its appearance which reduces the alkalinity and increases the sulphate-carbonate. This zeolite is regenerated with sulphuric acid.

Boiler "compounds" are frequently used for scale prevention in the smaller boiler plants. These compounds are introduced directly into the feed water and, to prove successful, must be

[1] *Trans. A.S.M.E.*, May, 1938, p. 315.

adapted to the analysis of the particular feed water. In general, the effect of these compounds is to form a sludge that is easily blown down or a soft scum that can be washed from the tubes with a water stream.

217. Steam Driers and Purifiers.—The ebullition inside a boiler, especially when carrying a heavy load, causes a mist of minute water drops to be carried up with the steam. In order to collect the steam so as to have it as free as possible from unevaporated water, it is common practice to fit the boiler with a so-called *dry pipe.* This consists of a horizontal pipe near the top of the shell, having a large number of small holes through which the steam must pass, the sudden change in direction being supposed to throw out the water.

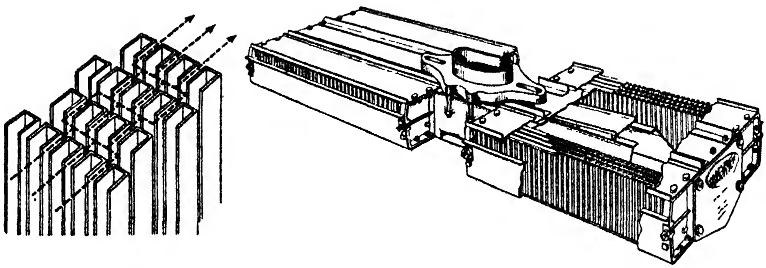


FIG. 244.—Tracyfier or steam purifier (*Andrews-Bradshaw Co.*)

These dry pipes have two disadvantages, namely, they do not dry the steam satisfactorily, and the water that is separated out falls back into the boiler and returns the dissolved salts to the boiler. As the water is boiled away, the concentration of salts in the boiler builds up, causing the bursting steam bubbles to throw an increasing quantity of water into the steam space. To reduce the concentration, a portion of the water is blown down at intervals through the blowoff valve and is replaced by fresh feed water. The amount of this blowdown is an important item in the economy of the plant.

A number of devices for cleaning and drying the steam as it leaves the boiler have been placed on the market. Figure 244 shows a "Tracyfier," which is placed inside the boiler drum in place of the dry pipe. The Tracyfier consists of a number of gutter-shaped baffles placed vertically and staggered, one behind the other, so that the steam is compelled to follow a tortuous course at low velocity among these baffles before it reaches the boiler nozzle. The baffles divide the steam into thin ribbons about $^{3}/_{16}$ in. thick and 7 in. wide, and the winding of these ribbons among the baffles removes the moisture with its impurities.

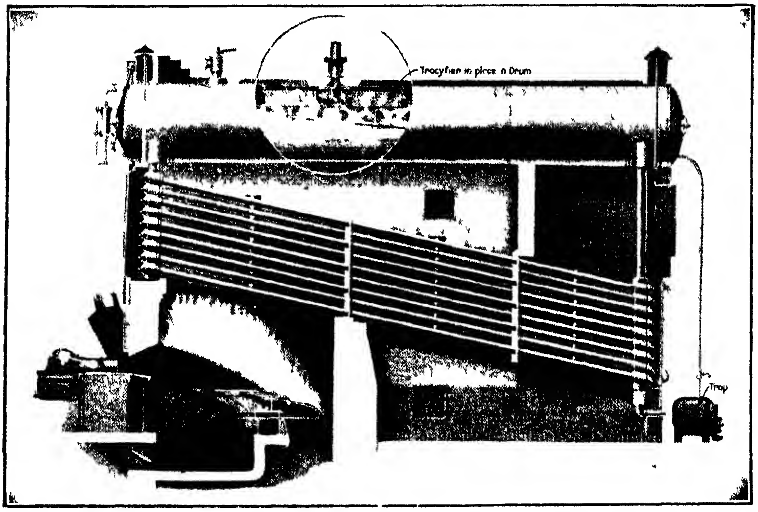


FIG. 245.—Tracyfier in place in steam drum of boiler. (Notice trap on floor at rear of setting.)

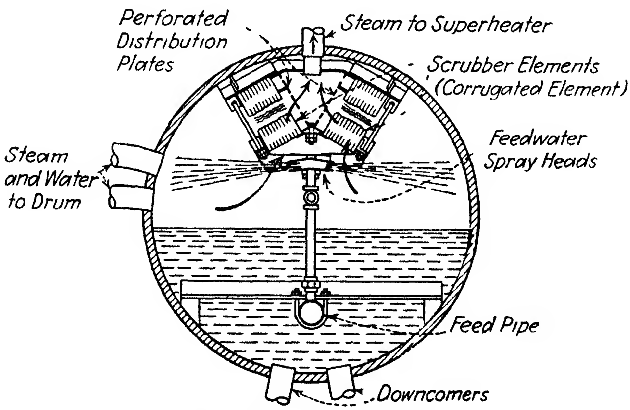


FIG. 246.—Steam washer.

This water is drained away from the boiler by means of a trap, as shown in Fig. 245, and is discharged to the sewer.

A spray-type steam washer is shown in Fig. 246. The wet steam entering the drum through the tubes at the left is caused to pass through a spray of the entering feed water. The feed water, having a lower concentration of salts than the droplets carried by the steam, dilutes the entrained water. The steam is then dried by the corrugated scrubber and the perforated screen before passing out of the drum at the top. The steam washer is plainly visible in the boiler of Fig. 198.

218. Feed-water Heating.—Before its introduction into the boiler, it is desirable to heat the feed water to as near the boiling point as possible, for several reasons: (1) Cold water may cause strains in the boiler shell; (2) the capacity of the boiler plant is increased if some heat is added outside the boiler; (3) if the feed is heated by waste heat, an improved efficiency for the plant results; (4) heaters may be used as water treaters to remove scale-forming solids or dissolved oxygen and carbon dioxide.

The heat used for feed-water heating may be supplied by exhaust steam from the auxiliaries in the plant, such as boiler-feed pumps, and, if so, the high steam rates of these auxiliaries cause no concern unless they supply too much exhaust steam for feed heating. Or, steam for feed heating may be obtained by bleeding the main turbine at one or more stages, as was pointed out in Art. 160. Noncondensing turbines or engines usually provide abundant exhaust steam for feed heating. In some cases live steam is used for this purpose. Feed-water heaters that utilize the heat of the escaping chimney gases are called *economizers*.

219. Open Feed-water Heaters.—Heaters in which the water comes into direct contact with the steam are called *open heaters*. Figure 247 shows a sectional view of a Swartwout open heater. The exhaust steam enters at the left side through the oil separator, in which the cylinder oil is thrown out and drained into the overflow trap. The steam then passes into the heater through the cut-out valve and there comes into contact with the feed water, which, entering at the top, trickles down from the baffle plates to the trays, one after another. As the water is heated, it gives up some of its mineral matter, which is less soluble in hot water than in cold, and deposits it as scale on the trays, from which it is removed periodically by opening up the heater. Open exhaust heaters are quite effective in removing the carbonates of calcium and magnesium but do not remove any great proportion of the sulphates. The heated feed falls to the bottom of the

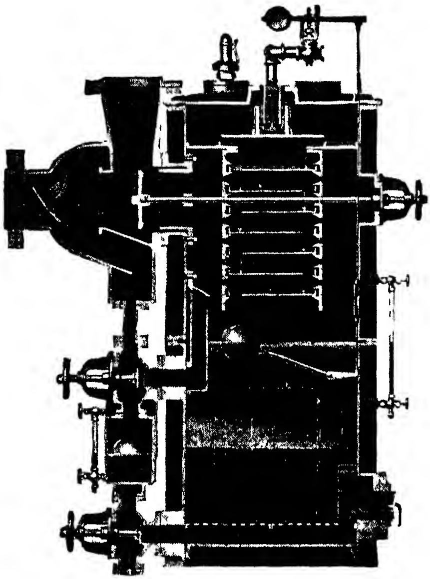


FIG. 247.—Swartwout open feed-water heater

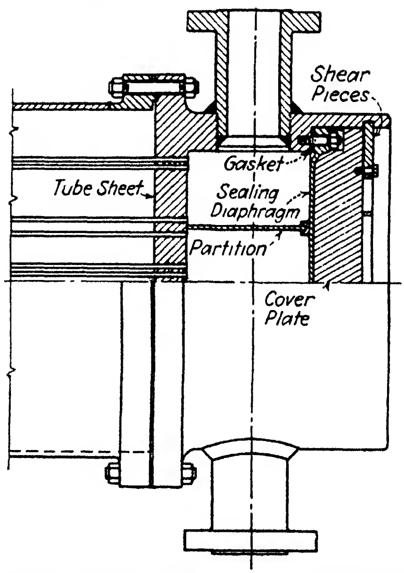


FIG. 248.—Construction details of Foster Wheeler high-pressure closed heater.

heater and passes through a filter on its way to the pump suction at the right.

The water level is maintained by a float-controlled valve on the water inlet and is safeguarded by the overflow at the left side. The steam pressure in the heater is limited by the relief valve on the top and, usually, by an atmospheric relief valve placed on the large pipe flange above the steam entrance.

Open feed-water heaters are always connected to the suction side of the boiler-feed pump, and, since a pump cannot lift hot water by suction, the heater must be placed so that the water level is above the pump.

220. Closed Feed-water Heaters.—Figure 248 shows a small section of a *closed type* of heater, in which the steam does not come into contact with the water. The steam enters through a pipe at the top (not shown) and, after being condensed by contact with the cold tubes, enters a float trap at the bottom, from which it is discharged by a float valve into the feed-water storage.

Closed heaters are connected to the discharge side of the feed pump, and, therefore, the water is under somewhat more than boiler pressure. The heater of Fig. 248 is designed for 800 lb working pressure within the tubes and uses steam bled from the main turbine for heating. The tubes in this heater are rolled into the end plates, the left end plate *H* being made floating so as to allow for tube expansion independently of the steel shell.

The closed heater is particularly adapted to extraction heating. Since these heaters are difficult to clean, the water should be nearly free from scale-forming impurities before passing into the heater.

221. Steam Accumulators.—Plants supplying steam to widely fluctuating industrial loads must be prepared for the sudden demands by either (1) boiler capacity sufficient to care for the maximum peak or (2) steam accumulators which receive steam at high pressure and deliver it at a lower pressure. The accumulator has the advantages, in those plants in which it is feasible, that (1) a smaller total investment for boiler, plus accumulator, is possible and (2) the boiler can operate at a steady and more efficient load.

The accumulator operates on the principle of storing the energy of high-pressure steam in a large quantity of water, at saturation temperature corresponding to the accumulator pressure, and then releasing this energy by reducing the pressure and allowing part of the water to flash into steam at the lower pressure. Figure 249 shows a sectional view of a Ruth's accumulator. It consists of a cylindrical shell with spherical ends mounted in a horizontal position, usually out doors, and is heavily insulated. It is kept about 9/10 full of water. When the pressure in the pipe O is above that of the process main P, valve V_2 acts as a reducing valve, and the excess boiler steam from N enters the accumulator through the nonreturn valve E. It passes into the water

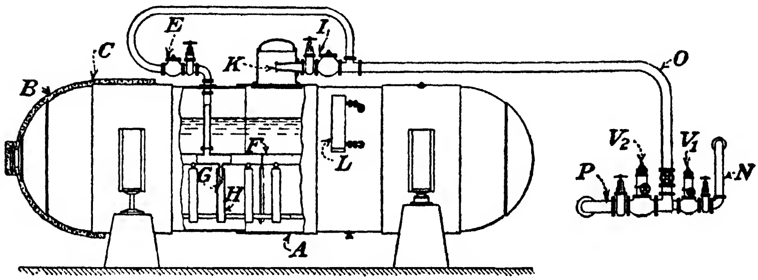


FIG. 249.—Ruth's steam accumulator. (*Foster Wheeler Corp.*)

through distributor pipe F and nozzles G, thus condensing and raising the temperature of the water. When the demand is increased so as to reduce the pressure in the pipe O and also in the accumulator, steam is released and delivered through the nozzle K and the nonreturn valve I. The special valve V_1 now throttles down so as to keep up the boiler pressure in main N while steam is drawn from O. On reduced loads the reverse action takes place. If, for example, the pressure in N is 150 lb and in P is 30 lb, the accumulator will operate between these two pressures.

Problems

1. Estimate, by means of Napier's rule, how much steam a safety valve having a nominal diameter of 4 in. with a 45-deg seat and 18-in. lift will discharge per hour from a boiler in which the pressure is 100 lb gage. Barometer 30 in.

2. A boiler is capable of evaporating 10,000 lb steam per hour when forced to its maximum. The boiler pressure is to be kept safely at 150 lb abs by a 45-deg seat safety valve. If the lift of the valve is assumed to be one thirty-second of the diameter, what should be the nominal diameter?

3. A 7-in. safety valve having a lift of 0.1 in. is applied to a boiler at sea level which is operated at a pressure of 225 lb gage. What is the approximate discharge rate of the valve in pounds per hour?

4. Raw water, assumed to be saturated with oxygen and nitrogen, is introduced into an open heater at 70°F. and its temperature raised to 200°F. The barometer is 29 in. How many standard cubic feet of these two gases will be removed from each pound of water by the heater?

5. How does the possible quantity of oxygen dissolved in water change when the pressure is changed from 100 lb per sq in. abs to 15 lb abs, the temperature remaining at 70°F.?

6. Condenser injection water at 60°F. is drawn from a spray pond, where it is saturated with oxygen and nitrogen. The barometer is 29 in. and the vacuum is 25 in. The water leaving the condenser has a temperature of 125°F. How many standard cubic feet of the two gases might be given up by each pound of water during its passage through the condenser?

7. The water in a boiler contains 30 grains of soluble salts per gallon. The quality of steam leaving the boiler nozzle is 99 per cent, the 1 per cent being assumed to have the same concentration as the boiler water. How many pounds of solids are carried over into the superheater annually per boiler horsepower?

CHAPTER XX

BOILER PERFORMANCE

222. Heat Given to Each Pound of Water.—Each pound of water fed into the boiler by the feed pump receives heat. This heat raises the water from the feed temperature to the boiling point and then evaporates it and, in some boilers, superheats it. How much heat it receives in the boiler depends upon three things: (1) its enthalpy when entering, (2) the pressure carried in the boiler, and (3) the condition of the steam as it passes out of the boiler, whether wet, dry, or superheated.

We have three cases to consider:

Case 1.—*For wet steam* the enthalpy increase of each pound of water while in the boiler would be

$$h_f + xh_{fg} - (t_w - 32)$$

in which h_f = the enthalpy of the liquid at the boiler pressure p.
x = the quality of steam leaving the boiler.
h_{fg} = the enthalpy of evaporation of steam at boiler pressure.
t_w = the temperature of the feed water as it enters the boiler.

We are justified, in practice, in assuming that $(t_w - 32)$ equals the enthalpy of 1 lb water at normal feed temperatures, because the specific heat of water is very nearly unity for the usual range of temperatures found in boiler practice, say, from 50 to 200°F.

Case 2.—*For dry steam*, x becomes unity, and we may express the enthalpy increase by

$$h_g - (t_w - 32)$$

where h_g represents the tabular enthalpy of dry saturated steam at boiler pressure $(h_f + h_{fg})$.

Case 3.—*For superheated steam* we have

$$h_g + c_p(t_s - t_1) - (t_w - 32)$$

in which c_p = the mean specific heat of superheated steam over the range $t_s - t_1$.

t_s = the final temperature of the steam.

t_1 = the temperature of saturation at boiler pressure p.

Of course, $h_g + c_p(t_s - t_1)$ might be obtained from the superheated-steam table, Table IX, for the pressure p and superheat $(t_s - t_1)$, and it is usually simpler so to obtain it.

The enthalpy increase of the water for these three cases is illustrated in Fig. 250.

223. Boiler Ratings.—The number of pounds of water a boiler will evaporate per hour under standard conditions determines its capacity, and the unit of boiler capacity is the "boiler horse-

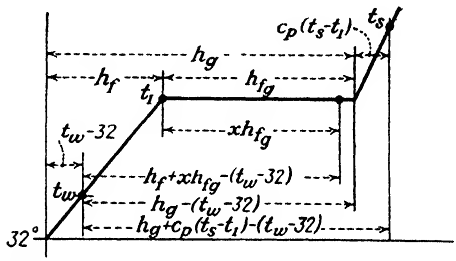


FIG. 250.—Enthalpy increase of water in a boiler

power." This was fixed in 1876 as "the evaporation of 34.5 lb water per hour from a feed temperature of 212° into dry saturated steam at 212°." If we use the new A.S.M.E. tables as standard, this means the absorption by the water in the boiler of 34.5 × 970.3, or 33,472 B.t.u. per hour.

If we consider that an engine horsepower-hour is the performance of 2,543 heat units of work, Art. 50, there is a very definite relation between a boiler horsepower and an engine horsepower, but the relationship has no importance in engineering. Both figures are completely arbitrary units of measure, and, although they originally had a close relation to each other, it has long since been lost by the advance in steam machinery.

In the early years of steam-boiler practice it was found that under good operating conditions an average square foot of heating surface of a fire-tube boiler could evaporate a little less than 3 lb. equivalent water per hour, and a water-tube boiler, about 3½ lb per sq. ft per hr. This means that about 12 sq ft fire-tube boiler surface and 10 sq ft water-tube surface were required to evaporate 34.5 lb equivalent water per hour. Manufacturers have, therefore, been accustomed to rate their boilers on the basis of dimensions at 10 sq ft per boiler hp for water-tube boilers and 12 sq ft for fire-tube boilers. Boilers that are used for low-pressure heating purposes are not rated in horsepower but upon the amount of radiation surface that they will supply with steam.

Under modern methods of firing with stokers or pulverized coal or oil, the amount of water that may be evaporated per square foot of heating surface has been increased many fold, even as much as ten times the original $3\frac{1}{2}$ lb, but the boiler is still rated on the basis of 10 (or 12) sq ft per hp, and the actual performance expressed as a percentage of rating. This forcing of a boiler is not detrimental to it in the same sense that overloading a generator may damage it, because, so long as the boiler is transmitting the heat to the water and is evaporating the water, it is not injured. There is much greater danger of injury to the furnace lining than to the boiler itself.

Example.—A water-tube boiler having 5,000 sq ft heating surface evaporates the equivalent of 40,000 lb water "from and at 212°." What horsepower is it developing and what percentage of its rated horsepower?

SOLUTION:

a. 40,000 ÷ 34.5 = 1,160 boiler hp is being developed

b. 5,000 ÷ 10 = 500 rated boiler hp.

Rating being developed equals 1,160/500 = 232 per cent

It should be pointed out that rating in terms of boiler horsepower based on heating surface has little significance in connection with modern, high-capacity, steam generators since these include, not only the conventional boiler heating surface, but also economizer and water-wall surface. A more satisfactory practice for the manufacturer, and the one commonly used, is to guarantee the generation of a definite number of pounds of steam per hour under specified conditions, basing his guarantee on the heat-absorbing capacity of boiler proper, superheater, economizer, water walls, and even the air preheater. Guarantee ratings in terms of "million Btu per hr" have been suggested as still more definite.

224. Boiler Economy.—When 1 lb fuel is burned on the grate, the heat generated will evaporate some of the water in the boiler, and the number of pounds of water evaporated for each pound of fuel is called the "economy" of the boiler and furnace. "Economy" may be expressed in terms of "actual water" evaporated under boiler conditions or "equivalent water" evaporated "from and at 212°" and is then called "actual evaporation" and "equivalent evaporation." The fuel may be considered on the basis of each pound of moist coal as fired, of dry coal, or of combustible. Thus we may have six combinations representing economy, and the particular one desired must be definitely stated:

1 Actual evaporation per pound of moist coal as fired.
2 Actual evaporation per pound of dry coal.
3 Actual evaporation per pound of combustible
4 Equivalent evaporation per pound of moist coal as fired.
5. Equivalent evaporation per pound of dry coal.
6. Equivalent evaporation per pound of combustible.

225. Boiler Efficiency.– As indicated in Art. 73, the efficiency of any piece of energy apparatus is output divided by input, and. since the function of a boiler is to transform the potential heat energy of the coal into heat energy in the steam, its efficiency may be expressed in the same manner. It must be remembered that the input and output are Btu's and not pounds of coal and pounds of steam.

Then, if we consider the input as the heating value of 1 lb coal, we have

$$E = \frac{\text{evaporation per pound coal} \times \text{Btu's given to each pound of water}}{\text{Btu content per pound of coal}} \tag{123}$$

as the general expression for efficiency. But, since the evaporation may be expressed in six different ways, as pointed out in Art. 214, it is evident that evaporation in the numerator must refer to the coal in the same condition as the pound of coal in the denominator, that is, "as fired," "moisture free," or "moisture and ash free;" and also Btu's given to each pound of water must refer to the appropriate pound of water, whether actual or equivalent.

$$E = \frac{\text{actual evaporation per pound of coal as fired} \times \text{actual heat received per pound of water}}{\text{Btu content per pound of coal as fired}} \tag{124}$$

or

$$E = \frac{\text{actual evaporation per pound dry coal} \times \text{actual heat received per pound of water}}{\text{Btu content per pound of dry coal}} \tag{125}$$

and, likewise, for the other expressions for economy. The efficiency is a single, definite quantity, and all these equations or methods will give the same value of the efficiency.

226. Heat Losses from Boilers.—The fact that the efficiency of a boiler is less than 100 per cent implies that some of the potential heat of the coal is dissipated and not given to the water in the boiler. The heat thus dissipated is lost in several ways:

1. Heat carried away as sensible heat of dry chimney gases ($=L_1$).
2. Heat carried away by moisture in the fuel ($=L_2$).
3. Heat carried away by moisture formed by burning H_2 ($=L_3$).
4. Heat carried away by moisture in air supply ($=L_4$).
5. Heat loss due to carbon in ashpit refuse ($=L_5$).
6. Heat loss due to carbon monoxide in chimney gases ($=L_6$).
7. Radiation and other losses ($=L_7$).

It is usually most convenient to compute the amounts of these losses for each pound of dry coal fired, for the reason that efficiency is usually computed on the dry-coal basis, since the heat value of the coal is determined from a dry sample and is, therefore, expressed on a dry-coal basis.

1. *Heat Carried Away as Sensible Heat of Dry Chimney Gas.*—This is usually the largest loss and is frequently as much as all the other losses combined. To calculate its magnitude, we must have the analysis of the flue gas and the ultimate analysis of the coal or, at least, the total carbon content. From this we compute the weight of dry combustion products per pound of dry coal by one of the methods given in Art. 44. Then, if we know the temperatures of the escaping gas and of the boiler-room air, we may compute the heat lost as sensible heat of dry gases by

$$L_1 = W_{dg}c_{pm}(t_g - t_{br}) \tag{126}$$

in which W_{dg} = the weight of dry gas pound dry coal fired.
t_g = the temperature of the escaping gas.
t_{br} = the temperature of the boiler room.
c_{pm} = the mean specific heat at constant pressure of the dry gases for the range from the boiler-room temperature to the chimney.

Since about 80 per cent of the gas is nitrogen and the specific heats of oxygen and carbon dioxide are not greatly different from that of nitrogen, it is generally sufficiently accurate to assume that the average specific heat of all the gases is 0.24 for the usual temperatures of chimney gases. The error in making this assumption is much less than that due to sampling and analyzing the coal and flue gases.

It is obvious that this loss should be reduced to a minimum by reducing as far as possible the amount of excess air and, therefore, the weight of dry chimney gases by proper methods of firing and control of dampers and air supply and by keeping the setting airtight so as to prevent infiltration. Abnormally high chimney temperatures usually indicate a dirty boiler inside or out.

2. *Heat Carried Away by Moisture in the Fuel.*—This heat loss per pound of dry coal is

$$L_2 = \frac{M}{1 - M}[1{,}058 + 0.46t_g - (t_{br} - 32)] \qquad (127)$$

$$= \frac{M}{1 - M}[1{,}090 + 0.46t_g - t_{br}]$$

in which M = the amount of moisture from the proximate analysis per pound of coal as fired.

$\frac{M}{1 - M}$ = the weight of moisture per pound of dry coal.

The first two terms in the brackets, $1{,}058 + 0.46\,t_g$, are an expression for the enthalpy of superheated steam at very low vapor pressures, such as would be exerted by the water in flue gases. Figure 251 shows how this expression was obtained by plotting the values of heat content against temperatures, as found in Keenan and Keyes tables, and extrapolating to 0°F. The subtractive term $(t_{br} - 32)$ represents the heat in the moisture when it is introduced into the furnace at boiler-room temperature. In Fig. 87 is shown the line of Fig. 251 in its relation to the rest of the Btu-temperature diagram. This is the line

PBN, which, it will be noted, has a slope of 0.46 and intersects the 0°F. temperature line at 1,058 Btu.

Equation (127) is valid for temperatures between 100 and 600°F. which is the normal range for chimney gases. A simpler, and perhaps safer, procedure is to take the tabular value of

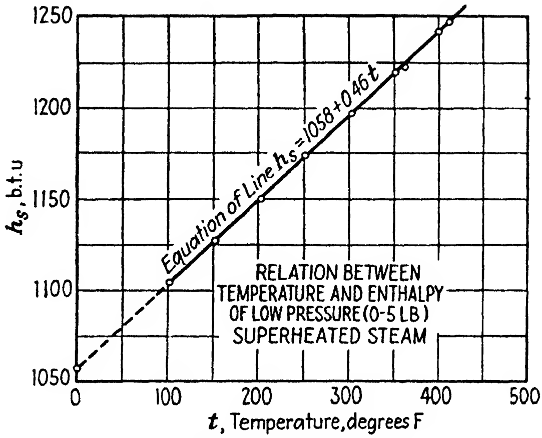


FIG. 251. Enthalpy of low-pressure steam.

enthalpy of superheated steam at low pressure (say 1 lb abs) for the chimney-gas temperatures from the steam table. Then

$$L_2 = \frac{M}{1 - M}[h - (t_{br} - 32)]$$

3. *Heat Carried Away by Moisture Formed by Burning Hydrogen.*—For computing this loss, we must have either the ultimate analysis of the coal or be able to calculate the amount of hydrogen from the total carbon and the Orsat analysis as in Art. 44. Then, since 1 lb hydrogen forms 9 lb water, the heat loss

$$L_1 = 9\frac{H}{100}[1{,}090 + 0.46t_g - t_{br}] \tag{128}$$

represents the heat loss due to this source, in which H is the weight of hydrogen per pound of dry coal and the quantity in the bracket is the same as for item L_2, above.

4. *Heat Carried Away by Moisture in Air Supply.*—The moisture in the air is conveniently determined by means of a wet- and dry-bulb thermometer and the tables of moisture content per cubic foot of air. From the specific volume of the

air for the temperature and pressure at the time of the boiler test and the weight of air used per pound of dry coal, we may compute the weight of moisture m in the air per pound of dry coal fired. Then the heat loss due to atmospheric moisture is

$$L_4 = m[0.46(t_g - t_{br})] \tag{129}$$

5. *Heat Loss Due to Carbon in Refuse.*—This loss may be computed in two different ways, depending upon the data available.

a. If the total weight of dry coal fired W_c, the total weight of refuse W_r, and the percentage of combustible in the refuse C_r, which is considered carbon, are known, the heat loss per pound of dry coal is

$$L_5 = \left(\frac{W_r}{W_c}\right)\left(\frac{C_r}{100}\right)14{,}600 \tag{130}$$

b. If the percentage of ash in the dry coal A_c and the percentage of combustible in the refuse C_r are known, we have, as the heat loss per pound of dry coal,

$$L_5 = \left(\frac{C_r}{100 - C_r}\right)\left(\frac{A_c}{100}\right)14{,}600 \tag{131}$$

If it is feasible to collect the cinders deposited in the boiler passes and combustion chamber, the loss thus represented may be found as in *a* or *b*, above, and added to the ashpit refuse. Otherwise, this loss becomes a part of the unaccounted-for loss of item 7.

6. *Heat Loss Due to Carbon Monoxide in the Flue Gas.*—Each molecule of carbon monoxide and of carbon dioxide accounts for one atom of carbon, and, since, according to Avogadro's principle, the number of molecules is proportional to the volumes of each gas, we have

$$L_6 = \left(\frac{CO}{CO + CO_2} \times C_B\right) \quad 10{,}050 \tag{132}$$

in which C_B is the weight of carbon actually burned, per pound of dry coal fired, and is equal to the total carbon per pound of dry coal C minus the carbon in the ashpit refuse, combustion chamber, and boiler passes as found for item 5, above.

7. *Radiation and Other Losses.*—This includes radiation from setting, exposed boiler surface and furnace, unconsumed hydrogen, cinders, and soot in breeching. This group of losses is obtained by subtraction of useful heat and known losses from the higher calorific value of the dry coal. This group of losses is commonly 2 or 3 per cent but ranges from a fraction of 1 per cent to 8 per cent.

The Heat Balance.—Table XVI shows a typical balance for a steam boiler.

TABLE XVI.—TYPICAL HEAT BALANCE[1]

	B.t.u.	Per cent
Heat absorbed by boiler	10,791	77.08
Loss due to moisture in coal	25	0.18
Loss due to moisture formed in burning H_2	642	4.58
Loss in dry chimney gases	1,616	11.54
Loss due to moisture in air	39	0.28
Loss due to incomplete combustion of C	227	1.62
Loss due to unconsumed C in ash	323	2.31
Radiation and unaccounted losses	337	2.41
Total	14,000	100 00

[1] From "Steam."

227. Boiler Trials.—Tests of boiler performance are made for one or more of the following specific purposes:

1. To compare different kinds of fuel.
2. To determine the efficiency for comparison with guaranteed performance.
3. To determine the causes of good or poor performance.
4. To compare different conditions of operation.
5. To determine capacity under prescribed conditions.

It is to be remembered that the entire steam-generating equipment must be considered as a unit and that it is impossible to separate the losses of one part from those of the other parts except in the matter of combustible dropping through the grates. For this reason the stoker and boiler manufacturers must know what apparatus is to be used with their own in order to make an intelligent guarantee as to efficiency or capacity.

In general, it is best to take sufficient data for the making of a complete heat balance because, from this, any of the above objects may be determined. The heat balance, also, serves as a check on the accuracy of the various parts of the test by showing up those items which are inordinately large or small.

Standardized rules for conducting boiler trials and the form of the report have been laid down by the Power Test Committee of the A.S.M.E., and this standard should be followed in order to

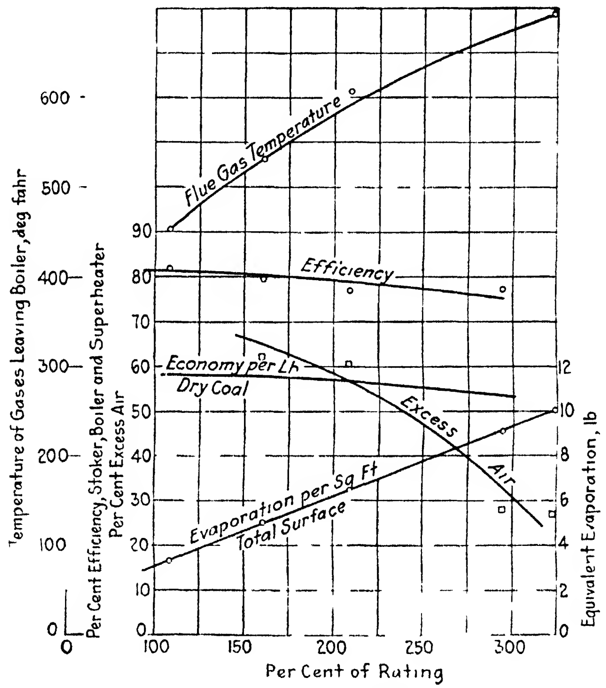


FIG. 252.—Performance curves for a steam boiler. (*Heine V-type at Moore's Park Station, Lansing, Mich., in Power.*)

insure that the results will be not only trustworthy but also comparable to other tests.

228. Variation of Efficiency with Load.—If we plot a curve with flue-gas temperatures as ordinates and load on the boiler as abscissas, we should find that the temperature rises very rapidly as the load is increased, because with greater load a greater quantity of fuel is fired and a greater quantity of flue gas must pass at greater velocity and thus does not have so much time to give up heat to the boiler. Such a curve for a Heine V-type boiler is shown in Fig. 252.

Assuming that the weight of flue gas per pound of fuel is constant for all loads, the flue-gas heat losses vary with the load. During the boiler trials of Fig. 252, however, the percentage of excess air decreased as the load increased, as is shown by the excess-air curve.

As the stoker is forced more and more, there is an increasing tendency to shove unburned carbon into the ashpit, and this loss increases with load.

Radiation from the setting and boiler surfaces is fairly constant over a wide range of loads.

The efficiency curve of Fig. 252 shows the variation of over-all efficiency with load. It will be noticed that the curve follows

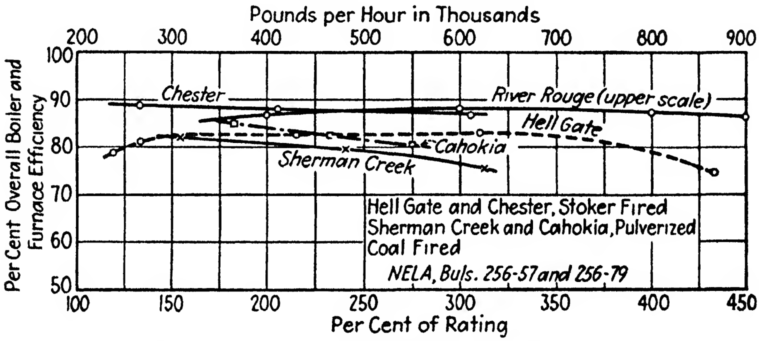


FIG. 253.—Performance curves for four large boiler plants.

the general form for efficiency of all kinds of energy apparatus, rising rapidly from zero at no load to near its maximum at a moderate load and, after going to the maximum, dropping again at high loads. It will be remembered that the point of high efficiency does not necessarily mean "full load," as in some apparatus.

Figure 253 shows the efficiency curves of four large boiler plants.

229. Boiler-room Records.—In addition to the occasional elaborate boiler trials discussed above, all modern boiler plants keep continuous records of the day-by-day performance of the equipment. These records not only enable the operating engineer to determine efficiency over any period of time but also serve as a means for checking up on each part of the plant, and in case of unsatisfactory performance the cause of the trouble can be easily found.

Since in most plants the quality of the coal and the factor of evaporation are fairly constant, a very good practical check on the plant performance is obtained by comparing the "economy" day by day or, better, shift by shift. Since the load on the boilers goes through rather definite daily cycles, we should expect the economy for the different shifts to be somewhat different. The measuring devices necessary for such daily checks are

Fig. 254.—Bailey coal meter.

1. *Coal-weighing lorries,* which weigh the coal as it is delivered to the stoker or pulverizer. In some plants counters on the stoker plungers give a fairly accurate check on the individual boilers, while in other plants coal-flow meters placed in the spouts leading to the stoker hopper are used. Figure 254 shows a Bailey coal meter.

A further refinement is obtained when provision is made for continuous sampling of the coal by which a composite daily, weekly, or monthly sample of the fuel is obtained. With this additional information the mean efficiency over any period may be calculated.

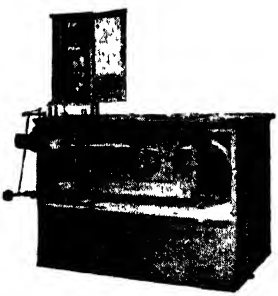


Fig. 255.—Yarway-Lea V-notch meter.

2. *Water Meters.*—For best results these are of either the V-notch type or the venturi type, although automatic weighing devices or displacement meters are sometimes used. Figure 255 shows a sectional view of a Yarway-Lea recording V-notch meter especially adapted for use with hot water. Since the flow over a triangular weir is a function of the head, the meter records by the float at the extreme left, which operates the indicating device in the cabinet at the top. These instruments usually indicate and draw curves of the flow rate in pounds per hour and also integrate the total pounds.

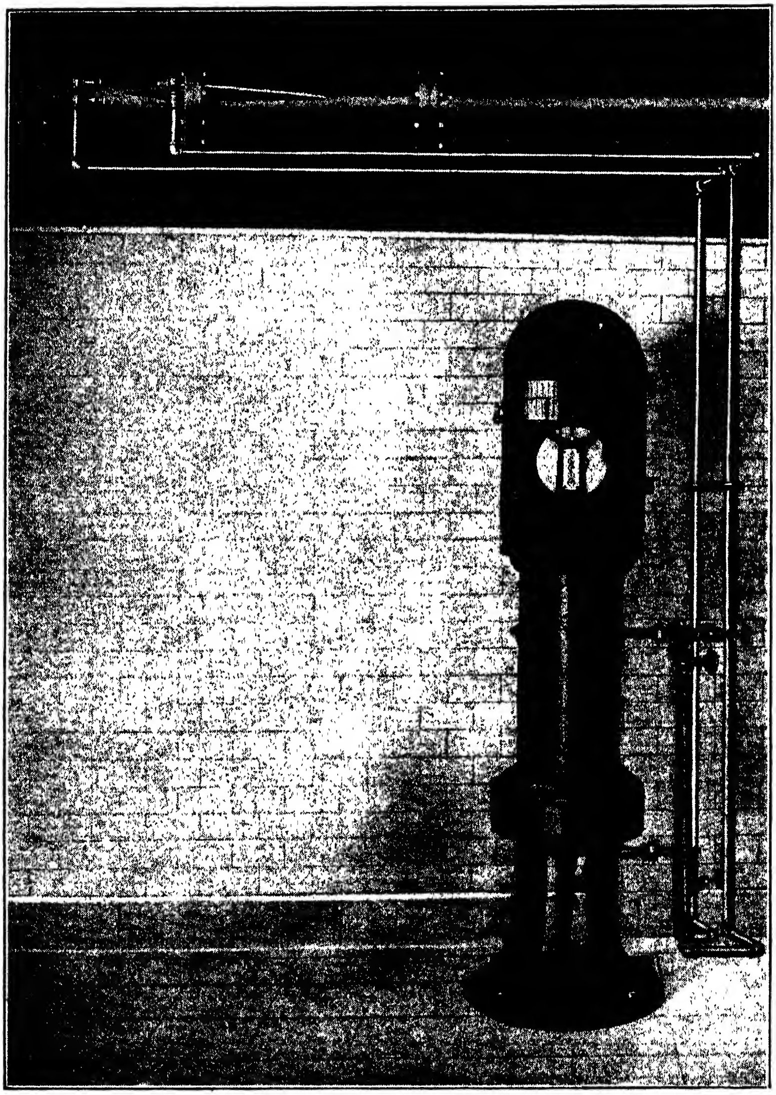


FIG. 256.—Simplex hot-water venturi meter.

Figure 256 shows the recording device of a Simplex venturi meter for hot water, the venturi tube being cut into the flow line and the recorder being placed at any convenient place in the boiler room. Some plants are equipped with weighing tanks for feed water by which the recording meters may be checked.

Some plants are equipped with permanent weighing tanks and scales that serve to check the calibration of the recording meters.

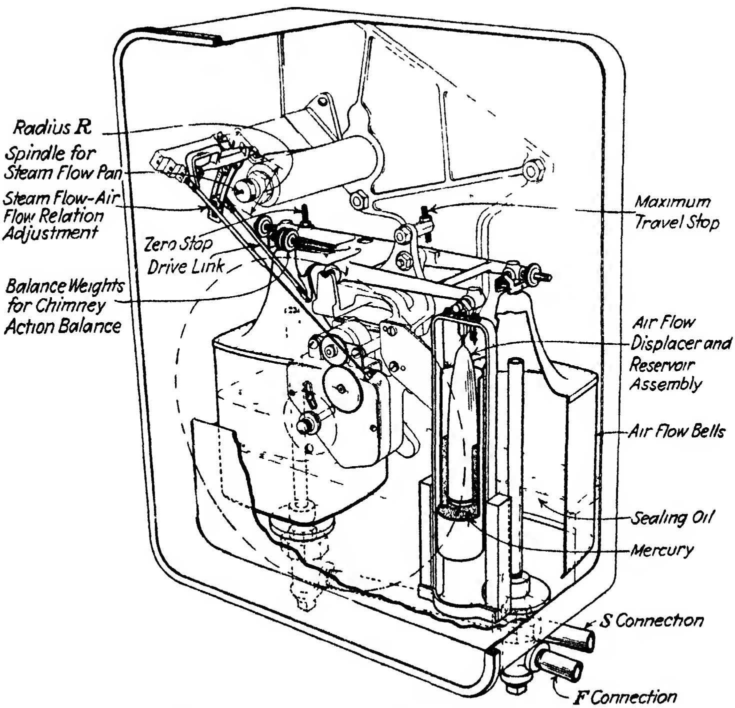


FIG. 257.—Bailey steam- and air-flow boiler meter.

3. *Steam-flow meters* are generally of the orifice type in which an orifice is placed in the steam line and the pressure drop is made to act through a mechanism to an indicating or recording meter.

4. *Recording pressure gages and thermometers* are employed in numbers in modern plants as a guide to operation and to furnish data for calculating the plant performance.

5. *Combination recorders.*—A type of boiler-room instrument that has found wide application is a combination instrument for recording two or more of the following: steam output; fuel

input; flue-gas flow; draft; and flue-gas temperature. Figure 257 gives a sectional view of such an instrument showing the "air-flow" recording mechanism. The two pipe connections at the bottom are made to the gas passes in such a way as to measure the friction loss between the two points in the setting, this friction loss being a measure of the flue-gas flow and also of the air flow to the furnace. This pressure differential, acting on the two oil-sealed bells, operates the air-flow pen. A similar mechanism in the same cabinet measures the pressure drop across an orifice in the steam outlet from the boiler. The two pens do not read directly in pounds per hour but are adjusted to each other so that for a given steam flow the air-flow line will coincide with the steam-flow line, when the optimum percentage of excess air is being used as determined by an Orsat trial. The operator of the boiler needs only to adjust drafts and dampers to keep the two lines together.

6. *Carbon Dioxide Recorders.*—For a systematic study and control of boiler-furnace operation the records of recording flue-gas thermometers and recording carbon dioxide instruments are of great practical value. Carbon dioxide recorders are built on three different principles: chemical, electrical, and mechanical. The chemical instruments operate by the absorption by either caustic solutions or dry cartridges, as in the Orsat apparatus. The absorption of the carbon dioxide from the flue-gas sample produces a reduction in pressure which moves the indicator over the scale.

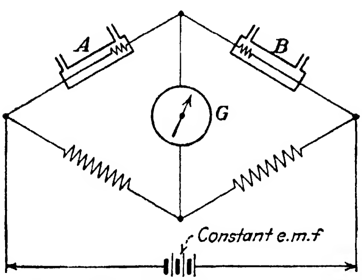


FIG. 258.—Principle of electric CO_2 recorder. (*Unbalanced Wheatstone bridge.*)

Electrical carbon dioxide instruments operate on the principle of the higher *thermal conductivity* of carbon dioxide than that of air. Figure 258 shows two platinum resistances placed in tubes A and B. Flue gases flow slowly through one tube, while a reference gas (air) passes at the same rate through the other. The resistances, being cooled by conduction of the gases in the tubes, attain equilibrium temperatures proportional to the thermal conductivity of the gases flowing and, in turn, attain resistances proportional to their temperatures. The resistances are connected in an unbalanced Wheatstone's-bridge circuit to a constant-voltage source, and the indications of the galvanometer are proportional to the resistances and, indirectly, to the percentage of carbon dioxide in the gas being analyzed.

The Ranarex mechanical carbon dioxide instrument operates through the difference in density of carbon dioxide and the other gases present. There are two chambers, each containing a motor-driven fan and, nearby, a stationary vaned disk acted upon by the blast of the fan. One chamber receives a flow of air and the other a flow of the gas being tested. The two vaned disks are connected together outside the chambers by levers, and the resultant movement of the two caused by the fans is proportional to the *difference* in densities of the two gases, which, in turn, is proportional to the percentage of carbon dioxide present.

Problems

1. What is the increase of enthalpy of each pound water fed into a boiler at 109°F. and converted into 98 per cent quality steam at 140 lb per sq in. abs?

2. A boiler receives feed water at 200°F. and converts it into steam at 600 lb abs and with a temperature of 600°F. What is the increase of enthalpy of the water?

3. What horsepower is being developed by a boiler that evaporates 10,000 lb water per hour from a feed temperature of 170°F. into 99 per cent quality steam at 200 lb per sq in. abs?

4. A boiler evaporates 8,000 lb water per hour from a feed temperature of 150°F. into steam at 280 lb per sq in. abs and 100° superheat. What boiler horsepower is being developed?

5. A 500-hp engine uses 20 lb dry steam per indicated horsepower-hour. Steam pressure 175 lb per sq in. abs and feed temperature 120°F. Allowing 5 per cent of the boiler steam for plant auxiliaries, what should be the boiler horsepower?

6. How many boiler horsepower will be required to supply steam for a 30,000-kw turbine at 10 lb steam per kilowatt-hour? Boiler pressure, 240 lb abs; superheat 240°F.; feed temperature 195°F.

7. A water-tube boiler having a heating surface of 3,750 sq ft evaporated 15,480 lb water in an hour, from a feed temperature of 162°F. Boiler pressure was 147 lb abs, and the steam quality at the boiler outlet was 99.5 per cent. What percentage of its rated horsepower was the boiler developing?

8. During an 8-hr shift, a boiler furnace is fired 22,420 lb moist coal containing 5.4 per cent of moisture. At the same time the boiler is fed 181,400 lb water at 141°F., which is evaporated into dry steam at 125 lb per sq in. abs. Calculate (*a*) the actual economy per pound of moist coal; (*b*) the equivalent economy per pound of moist coal; (*c*) the equivalent economy per pound of dry coal.

9. A 250-ihp engine uses 30 lb steam per indicated horsepower-hour. Feed temperature 125°F.; boiler pressure 140 lb per sq in. abs; superheat 50°. Boiler economy, 8 lb actual water per pound of coal. (*a*) How many pounds of coal will be required per hour? (*b*) What boiler horsepower will be developed?

10. A 10,000-sq ft, water-tube boiler evaporated 50,000 lb "equivalent" water per hour. What percentage of rating was it developing?

11. A boiler has an "equivalent" economy of 8 lb water per pound of coal as fired. Steam is generated at 160 lb abs and 100° superheat from a feed temperature of 120°F. Coal costs $3 per ton at the boiler room. Calculate the coal cost (*a*) per 1,000 lb steam generated; (*b*) per 1,000,000 Btu in steam.

12. A boiler is fed 10,000 lb water at 120°F. per hr. Boiler pressure 165 lb abs and steam quality 99.5 per cent At the same time the furnace is fired 1,250 lb moist coal having a heating value of 12,000 Btu per lb as fired. Calculate (*a*) the actual economy; (*b*) the equivalent economy; (*c*) the efficiency of boiler, furnace, and grate; (*d*) the boiler horsepower developed.

13. A boiler is said to have evaporated 12 lb water from a feed temperature of 90°F. into dry steam at 125 lb abs per pound dry coal containing 13,500 Btu per pound dry. (*a*) Is this possible? (*b*) Is it probable?

14. A boiler plant burns 100 tons coal per day of 12 hr. Coal contains 76 per cent C, 5 per cent H_2, and 5 per cent O_2. Efficiency of boiler, furnace, and grate is 71 per cent; feed temperature 85°F., steam pressure 149 lb per sq in. abs; quality 99 per cent. Calculate (*a*) the "equivalent" economy per pound of coal as fired; (*b*) the boiler horsepower being developed.

15. A 1,000-kw turbine uses 18 lb steam per kilowatt-hour. Steam condition 200 lb abs and 200° superheat. Feed temperature 180°F. Coal contains 11,000 Btu per pound as fired. Efficiency of boiler, furnace, and grate 75 per cent. Calculate (*a*) the boiler horsepower required to supply the necessary steam for full load on turbine; (*b*) the number of pounds of water fed to boiler per hour.

16. A boiler evaporated 8.4 lb equivalent water per pound of dry coal containing 13,600 Btu per lb, dry. Calculate the efficiency of boiler, furnace, and grate.

17. During a boiler trial lasting 24 hr, the following data were taken: temperature of feed water, 58°F.; steam pressure, 156.3 lb gage; quality of steam, 99 per cent; total weight of moist coal fired, 15.078 lb.; heating value of coal, 12,292 Btu per lb, dry; moisture in coal as fired, 7.5 per cent; total weight of water fed to boiler, 105,100 lb; barometer, 29.5 in.; boiler heating surface, 1,500 sq ft. Calculate (*a*) "equivalent" economy per pound of dry coal; (*b*) efficiency of boiler, furnace, and grate; (*c*) boiler horsepower developed; (*d*) percentage of rating developed.

18. A water-tube boiler having a heating surface of 5,080 sq ft was tested and the following data taken during a 16¼-hr run: total weight of coal as fired, 39,670 lb; heating value of the coal as fired, 12,000 Btu per lb; total water fed to boiler, 336,200 lb; steam condition, 119 lb gage and 136.5° superheat; feed-water temperature 50°F.; cost of coal, \$3.75 per ton at boiler room. Barometer, 30 in. Calculate (*a*) equivalent economy per pound of moist coal; (*b*) efficiency of boiler, furnace, and grate; (*c*) boiler horsepower developed; (*d*) percentage of rated horsepower developed; (*e*) coal cost of steam per 1,000 lb delivered to header; (*f*) coal cost per 1,000,000 Btu steam delivered to header.

19. A coal containing 75 per cent C, 5 per cent H_2, and 8 per cent O_2 on a dry basis is burned in a boiler furnace. The percentage flue-gas analysis is: CO_2, 4.3; O_2, 12.7; and N_2, 83 per cent. Temperature of the flue gases is 525°F., and that of the boiler room 70°F. Calculate the heat carried away per pound of dry coal by (*a*) the *dry* flue gases; (*b*) the water due to combustion of hydrogen.

20. A coal gave the following ultimate percentage analysis on a dry basis: C, 75.54; H_2, 4.97; O_2, 8 89; N_2, 1.96; S, 0.82. Heating value by calorimeter on dry basis 13,305 Btu per lb; moisture in coal as fired 5.41. During a boiler trial, the percentage flue-gas analysis by Orsat was as follows CO_2, 11.37; O_2, 7.40; CO, 0.13; N_2, 81.10. Temperature of flue gases 426°F. Temperature of boiler room 69°F. Calculate the Btu and percentage of heat value of each pound of the dry fuel carried away by (*a*) dry chimney gases; (*b*) moisture due to combustion of hydrogen; (*c*) moisture in coal; (*d*) incomplete combustion of carbon

21. A coal containing 11.41 per cent ash and 5.29 per cent moisture as fired has a heating value of 13,150 Btu per lb on a dry basis. During a boiler trial, the ashpit refuse contained 14.37 per cent combustible, and the flue gases had a temperature of 562°F., while the boiler-room temperature was 54°F. Calculate the percentages of the heat of the dry fuel lost (*a*) by moisture in the coal; (*b*) due to combustible in the ashpit refuse.

22. During a boiler trial, 36,350 lb coal containing 5 3 per cent of moisture were fired. At the same time, 5,620 lb dry ashpit refuse were collected, and this contained 14.4 per cent of combustible. Calculate the Btu lost per pound of dry coal fired due to combustible in the refuse.

23. A 6,140-sq ft water-tube boiler gave the following data during a 24-hr trial: barometer, 29.45 in.; steam pressure, 178.6 lb gage; steam temperature, 390°F.; feed-water temperature, 199°F.; temperature of gases leaving boiler, 515°F.; boiler-room temperature, 70°F.; fuel fired per hour, 2,899 lb; moisture in fuel as fired, 5.41 per cent; ash in fuel as fired, 7.55 per cent; ashpit refuse per hour, 207.3 lb; combustible in refuse, 8.7 per cent; actual water fed to boiler per hour, 26,999 lb. Percentage flue-gas analysis by Orsat: CO_2, 11.46; O_2, 7.73; CO, 0 09; N_2, 80.72. Ultimate analysis of coal on dry basis: C, 75.5; H_2, 5.0 O_2, 9; N_2, 2. Calorific value of fuel on dry basis 13,300 Btu per lb. (*a*) Make a complete heat balance for this boiler trial. (*b*) Calculate boiler horsepower developed.

24. A stoker-fired, longitudinal, horizontal-drum, water-tube boiler was subjected to a 24-hr test and the following data taken[1]: steam pressure, 127

[1] *Power*, 63: 399.

lb gage; moisture in steam, 2.02 per cent; feed-water temperature, 195°F.; temperature of exit gases, 468°F.; boiler-room temperature, 73°F.; barometer, 14.7 lb. The percentage analyses: CO_2 13.1, O_2 5.6, CO 0.2, N_2 81.1. Total coal as fired, 47,968 lb; total weight dry refuse, 3,148 lb; moisture in coal, 2.41 per cent; heating value of coal, 14,871 Btu per lb dry; combustible in dry refuse, 9.82 per cent. Ultimate percentage analysis of coal on dry basis: C, 84.76; H_2, 4.55; S, 0.91; O_2, 2.91; N_2, 1.59; ash, 5.28. Total water fed to boiler, 572,236 lb. (*a*) Make a complete heat balance for this boiler trial. (*b*) Calculate boiler horsepower developed.

25. A waste-heat boiler and superheater[1] receiving the hot gases from an oil-fired forging furnace was tested, and the following data were obtained: duration of test 7 hr; steam pressure, 140.9 lb gage; gas temperature, entering, 1348°F. and, leaving, 485°F.; feed-water temperature, 175°F.; steam temperature, 495°F.; total water evaporated, 53,485 lb; average carbon dioxide in waste gases, 9.06 per cent by volume; weight of waste gases per hour, 38,500 lb; fuel oil consumed in the forging furnace per hour, 152.7 gal.

Assume the fuel oil to consist of 84 per cent carbon and 15 per cent hydrogen and to have a specific gravity of 0.88. Barometer 29.3 in.

Calculate (*a*) Btu absorbed by each pound of water while in boiler and superheater; (*b*) equivalent evaporation per hour; (*c*) boiler horsepower developed by boiler and superheater; (*d*) percentage of heat of entering gases absorbed by boiler and superheater (consider dry gases to be nitrogen); (*e*) percentage of total heat of fuel absorbed by boiler and superheater; (*f*) heat remaining in escaping gases above 60°F.

26. A boiler, stoker, and superheater are guaranteed to give the following performance:

Equivalent evaporation per hour from and at 212°	25,550	38,250	51,000	63,750	76,500
Combined efficiency, per cent..................	76	78	76	72	68

Boiler pressure 300 lb gage; 125° superheat; barometer 29.8 in.; feed temperature 190°.

Calculate (*a*) the actual water fed to the boiler at each load; (*b*) boiler horsepower developed at each load. (*c*) Plot efficiency against boiler horsepower.

27. A boiler plant is fed an average of 67,157,000 lb water per month during a certain year, and the average actual evaporation per pound of coal as received is 8.7 lb. Heat value of the coal is 13,760 Btu per lb as received; boiler pressure, 165 lb abs without superheat, and the feed temperature is 175°F. Coal costs $2.50 per ton delivered at the plant. Calculate (*a*) the average efficiency for the year; (*b*) the average boiler horsepower developed; (*c*) the cost of coal for this year.

[1] *Power*, **63**: 116.

CHAPTER XXI

REFRIGERATION

230. Refrigeration.—By refrigeration is generally meant the removal of heat from a body or substance to such an extent as to leave it, or maintain it, at a lower temperature than that of its surroundings. Mechanical refrigeration means the removal of heat by mechanisms or mechanical devices.

In the real refrigerating plant decided advantages are obtained from the use of fluids that may be easily vaporized and liquefied at the temperatures involved in refrigeration work, instead of using those that remain a gas during the entire cycle. Air, because of its cheapness and other desirable characteristics, is the only fluid used to any considerable extent as the working substance in a gas-refrigeration cycle. The air cycle (first used in 1873) has now, however, been almost superseded by the "vapor cycles," because the latter gave better performance and used less expensive equipment occupying less space. A discussion of the air cycle is given because a clearer understanding of the vapor cycles will be thus obtained.

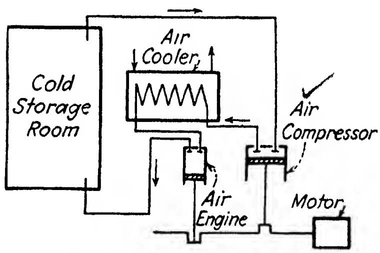


FIG. 259.—Flow diagram of open-cycle air-refrigeration system.

231. Air-refrigeration Cycles.—Two types of air cycles are used, the *open cycle* and the *closed cycle*.

Since air is nonpoisonous, it can be brought into direct contact with foodstuffs without detriment to them, and effective heat absorption is obtained by the direct contact of the open cycle. In this arrangement, as shown in Fig. 259, air at atmospheric pressure is taken from the cold-storage room into the compressor cylinder and compressed to a pressure which may be equal to 4 or 5 atm. The air is delivered to the air cooler where its temperature is materially reduced by the air flow through pipes cooled with water. The air then goes to the cylinder of the air engine.

In the air-engine cylinder it expands until its pressure becomes atmospheric, delivering some mechanical energy to the compressor and at the same time being cooled to a temperature in the order of 100°F. This cold air is exhausted from the air engine to the cold room where it absorbs heat.

The closed-cycle system differs from that of Fig. 259 in having the cooling air circulated through pipe coils in the cold-storage room, the lower pressure being about 50 lb per sq in. abs and the upper pressure about 200 lb. The pressures depend upon the inside and outside temperatures.

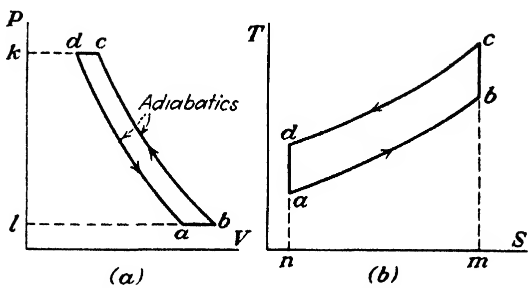


FIG. 260. Theoretical air-refrigeration cycle

Figure 260 gives the theoretical air-cycle diagrams, (*a*) on the *P-V* plane and (*b*) on the *T-S* plane.

On the *P-V* plane, area *lbckl* represents work done by the compressor upon the air during suction, compression, and delivery. Area *kdalk* represents work done by air in the engine during admission, expansion, delivery. Area *abcd* represents net work in foot-pounds, that is, the work equivalent of the energy which must be supplied externally to the system.

On the *T-S* plane, the amount of heat abstracted from air in the air-cooler is area *cdnm*. Amount absorbed by the air in passing through *CSR* = area *abmn*. Area *abcd* represents net work in Btu, that is, the heat equivalent of the energy which must be supplied externally to the system.

That external energy must be supplied for the operation of the system is evident from an examination of the theoretical diagrams. Since both expansion and compression take place over the same range of pressures and since the temperature of the air entering the compressor is higher than that leaving the expansion cylinder, it is evident that the power consumed by the compression process will exceed the power developed as a result of expansion in the air engine. This effect is apparent when it is remembered that greater volumes must be handled in the compression cylinder owing to the higher temperature level.

232. Performance of Refrigeration Machines.—The *coefficient of performance* of a refrigerating cycle is defined as *the ratio of the heat abstracted from the cold body to the energy expended to attain this result.*

The A.S.R.E. defines the energy unit of commercial refrigerating machines as the *standard ton of refrigeration,* equal to 288,000 Btu, equivalent to the latent heat of fusion of 2,000 lb ice. This unit involves no time element and is used to express capacity of refrigeration. The heat of fusion of ice is about 144 Btu per lb.

The *rate of refrigeration* is defined as the standard commercial ton of refrigeration or 288,000 Btu per 24 hr (12,000 Btu per hr or 200 Btu per min).

The *ice-making capacity* of a machine refers to the number of tons of ice the machine can make in 24 hr, as say, a 10-ton machine.

The above information may be more concisely expressed by means of equations.

Let Q_1 = the heat carried away by the cooling water in the condenser.

Q_2 = the heat absorbed from the cold room.

E_m = the heat equivalent of mechanical energy, the subscripts e, c, and r referring to expansion, compression, and required externally.

Then

$$Q_2 = WC_p(T_b - T_a)$$
$$Q_1 = WC_p(T_c - T_d)$$

in which W = weight of air circulated in pounds per min.

C_p = constant-pressure specific heat of air = 0.24.

Temperatures are in °F., absolute, and refer to points on diagrams as indicated by subscripts.

Net work required, $E_{mr} = Q_1 - Q_2 = WC_p[(T_c - T_d) - (T_b - T_a)]$, Btu.

Also,

$$Q_1 + E_{me} = Q_2 + E_{mc}$$
$$E_{me} + E_{mr} = E_{mc}$$
$$Q_1 = Q_2 + E_{mr}$$
$$\text{C.o.p.} = \frac{Q_2}{E_{mr}} = \frac{T_b - T_a}{T_c - T_d - (T_b - T_a)} = \frac{T_a}{T_d - T_a} = \frac{T_b}{T_c - T_b}$$

Example A.—In an ideal air-refrigeration machine the temperatures are as follows: entering compressor, 30°F., leaving compressor 272°F., entering engine, 60°F., leaving engine −112°F.

Compute: (*a*) The heat absorbed per pound of air.

(*b*) The heat rejected per pound of air.

(*c*) The coefficient of performance.

SOLUTION.—(*a*) $Q_2 = c_p(T_b - T_a)$
$= 0.24[(460 + 30) - (460 - 112)]$
$= 0.24(490 - 348) = 34.1$ Btu per lb

(*b*) $Q_1 = c_p(T_c - T_d)$
$= 0.24[(460 + 272) - (460 + 60)]$
$= 0.24(732 - 520)$
$= 50.9$ Btu per lb

(*c*) $\text{C.o.p.} = \dfrac{T_b}{T_c - T_b} = \dfrac{490}{732 - 490}$
$= 2.025$

The preceding problem illustrates the method of calculating the coefficient of performance of an ideal cycle involving the circulating of one pound of air. The actual air-refrigeration cycle differs in several respects from the ideal, namely,

a. The temperature of the air entering the compressor is usually somewhat below the cold-room temperature.

b. A pressure drop often accompanies the temperature drop in the cooling coils, so that p_d is somewhat less than p_c.

c. Owing to water jacketing of the compressor, the value of n during compression may be reduced to a value lower than k, as assumed in the ideal cycle.

In the design of air-refrigeration systems the basis of the calculations is the desired capacity and the temperature of the cold room and cooling water.

Total heat which must be removed from the cold room per minute,

$$Q_2 = WC_p(T_b - T_a)$$

in which Q_2 = total desired capacity in tons of refrigeration $\times$ 200; since 200 Btu per min = standard commercial ton of refrigeration.

Transposing we have

$$W = \frac{Q_2}{C_p(T_b - T_a)}$$

in which W = the weight of air to be circulated per minute since Q_2 is in Btu per min. If it is assumed that both expansion and compression follow the theoretical adiabatic, T_a and T_c may be calculated since, from Art. 71,

$$T_a = T_d\left(\frac{p_a}{p_d}\right)^{\frac{k-1}{k}} \text{ and } T_c = T_b\left(\frac{p_c}{p_b}\right)^{\frac{k-1}{k}}$$

Assuming complete expansion in the air-engine cylinder, the work area under line kda, Fig. 260, which is the work done in the expansion cylinder,

$$\begin{aligned}
E_{me} &= p_dv_d + \frac{p_dv_d - p_av_a}{k - 1} - p_av_a \\
&= \frac{p_dv_d - p_av_a}{k - 1} + (p_dv_d - p_av_a) \\
&= \frac{k}{k - 1}(p_dv_d - p_av_a) \\
&= \frac{k}{k - 1}WR(T_d - T_a) \text{ ft-lb} \\
&= WC_p(T_d - T_a) \text{ Btu}
\end{aligned}$$

and the net work required

$$\begin{aligned}
E_{mt} &= E_{mc} - E_{me} \\
&= WC_p[T_c - T_b - (T_d - T_a)]
\end{aligned}$$

as calculated above.

Example B.—In a 10-ton air refrigeration machine, the pressure at the end of compression is 75 lb per sq in. abs, at entrance to the expansion cylinder 65 lb per sq in. abs, and in the coils in the cold room 15 lb per sq in. abs. The temperature in the cold room is 30°F., and the air enters the expansion cylinder at 75°F. During compression n = 1.35, and expansion is adiabatic.

Determine.—(*a*) Net work required externally.

(*b*) The actual coefficient of performance.

SOLUTION:

$$T_a = T_d\left(\frac{p_a}{p_d}\right)^{\frac{k-1}{k}} = 535\left(\frac{15}{65}\right)^{\frac{1.4-1}{1.4}}$$
$$= 535(0.2305)^{0.286} = 535 \times 0.656$$
$$= 351°\text{F. abs}$$
$$T_c = T_b\left(\frac{p_c}{p_b}\right)^{\frac{n-1}{n}} = 490\left(\frac{75}{15}\right)^{\frac{1.35-1}{1.35}}$$
$$= 490(5)^{0.2595} = 490 \times 1.518 = 742°\text{F. abs}$$
$$W = \frac{Q_2}{c_p(T_b - T_a)} = \frac{10 \times 200}{0.24(490 - 351)}$$
$$= 59.9 \text{ lb per min}$$
$$E_{mc} = \frac{n}{n-1}WR(T_c - T_b)$$
$$E_{mc} = \frac{1.35}{1.35 - 1} \times 59.9 \times 53.36(742 - 490) = 3{,}100{,}000 \text{ ft-lb per min}$$
$$= 3980 \text{ Btu per min}$$
$$E_{me} = WC_p(T_d - T_a) = 59.9 \times .24(535 - 351) = 2645 \text{ Btu per min}$$
$$= 2{,}060{,}000 \text{ ft-lb per min}$$
$$E_{mr} = 3{,}100{,}000 - 3{,}060{,}000 = 1{,}040{,}000 \text{ ft-lb per min}$$

In other words (1,040,000/33,000) = 31.5 hp is required to operate the machine. This energy must be supplied from some external source such as a motor or engine.

$$\text{C.o.p.} = \frac{Q_2}{E_m} = \frac{10 \times 200}{1{,}040{,}000/778} = 1.495$$

233. Vapor-compression Refrigeration.– The use of the term "vapor cycle" means that the working substance is a vapor while passing through that very important part of the equipment which involves the expenditure of mechanical energy, that is, the compressor. However, in other parts of the cycle this working substance is a liquid or a mixture of vapor and liquid.

The basic equipment, as shown in Fig. 261, consists of four pieces: expansion valve, evaporator, compressor, and condenser. The liquid refrigerant at d flows through the expansion valve dropping in pressure from p_d to p_a at constant enthalpy. Heat is absorbed by the refrigerant in the evaporator at constant temperature and is evaporated. It is then compressed adiabatically to the higher pressure and condensed at constant pressure. Figure 262 (a) and (b) shows the theoretical cycle diagrammatically on the P-V and T-S planes, respectively.

ab is a constant pressure process taking place in the expansion coils while the mixture is absorbing heat.

bc is a process of passing vapor through compressor wherein its pressure and temperature are greatly increased so that it becomes a superheated vapor in state *c*.

cd represents constant pressure condensation in the condenser. The resultant liquid is next passed through the expansion valve and thus its pressure and temperature are greatly reduced, so that it enters the evaporator in the state shown at *a*; the process *da* being a throttling (constant enthalpy) path of expansion.

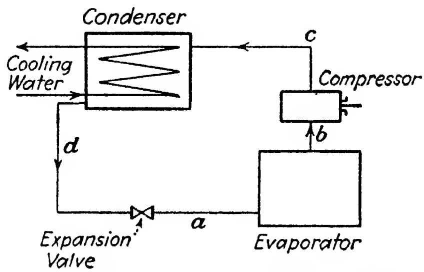


FIG. 261.—Flow diagram for vapor-compression refrigeration system.

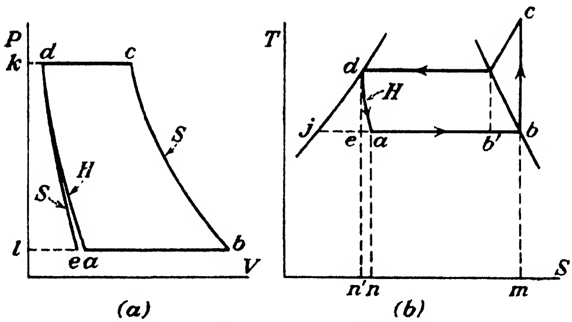


FIG. 262.—Theoretical vapor refrigeration cycle.

If a throttling process takes place from *d* to *a* without heat transfer, then h_a becomes known because it is equal to h_d. Line *da* represents the locus of successive states having the same heat content as *d*.

An examination of the *T-S* diagram will show that there is a loss of refrigeration effect due to the use of the expansion valve in preference to some device which would permit external work to be developed and used to furnish part of the power needed for compression. If the process of expansion had been adiabatic, the line *de* would have been followed in the figure, and the heat removed from the cold-storage room per pound of vapor would have been represented by area *n′ebm*. When the expansion valve is used, the heat removed per pound is represented by area *nabm*. The loss of refrigeration effect per pound of vapor may, therefore, be measured as the area *eann′*.

Further examination of the *T-S* diagram will show that, if an ideal engine had been used instead of a throttling valve, the adiabatic line *de* would have resulted, and the work obtainable in the engine would be the area *kdel*, which inspection shows is smaller than area *kdal*. This would indicate that the throttling valve delivers greater work than an ideal engine, which is impossible. Consequently, the area enclosed by the heavy lines *abcd* cannot represent the net work of the cycle.

$$h_d = h_a = h_{fa} + X_a h_{fga}$$

$$X_a = \frac{h_d - h_{fa}}{h_{fga}} = \text{quality of mixture entering the expansion valve}$$

in which h_d = enthalpy of liquid entering expansion valve.
h_{fa} = enthalpy of liquid at suction pressure.
h_{fga} = enthalpy of evaporation at suction pressure.

The above equation can be set up because the process *da* is one of constant enthalpy.

Let Q_1 = heat carried away by the cooling water in the condenser.
Q_2 = heat absorbed from the cold room (expansion coils).
E_{mc} = heat equivalent of the mechanical energy that must be supplied to operate the system, that is, supplied to the compressor.

$$\begin{aligned} E_{mc} &= Q_1 - Q_2 = \text{area } cdn'm - \text{area } abmn \\ &= (h_c - h_d) - (h_b - h_a) \end{aligned}$$

but

$$\begin{aligned} h_d &= h_a \\ E_{mc} &= h_c - h_b \end{aligned}$$

It will be realized in this development that $E_{mc} = E_{mr}$, since $E_{me} = 0$ with the expansion valve in use.

$$\text{C.o.p.} = \frac{Q_2}{E_{mc}} = \frac{h_b - h_a}{h_c - h_b}$$

in which h_b = enthalpy of vapor entering compressor.
h_a = enthalpy of mixture entering expansion coils.
h_c = enthalpy of vapor leaving compressor.
The weight of vapor flowing may be calculated from

$$W = \frac{Q_2}{h_b - h_a} = \frac{Q_2}{h_b - h_d}$$

In order to produce a commercial ton of refrigeration, the weight of vapor required per hour would be,

$$W = \frac{12{,}000}{h_b - h_a}$$

In the actual cycle the compression is not adiabatic because the cylinder is cooled and the compression line ends somewhere between C and the saturation line of the T-S diagram.

There are several vapors available for use in vapor-compression systems: ammonia, carbon dioxide, sulphur dioxide, water, methyl chloride, dichlorodifluoromethane (F-12) and dichlorotrifluoroethane (F-14) being the most common. Ammonia is used in most of the large refrigerating machines, being preferable for several reasons. It is desirable to use a refrigerant of which a small quantity can produce a large cooling effect. In order to have a small power equipment to keep the refrigerant in circulation through the system and to permit the use of a small cylinder in the compressor, the vapor of the refrigerant should have a small volume per pound, that is, the weight of 1 cu ft vapor should be large. This makes it possible to produce a given amount of refrigeration with a small compressor, as the size of the compressor and its power requirement depend on the volume of refrigerant circulated through the system. Ammonia possesses both these desirable properties, and this is the reason why it is most generally used in large refrigerating machines. Ammonia vapor is therefore selected for problem work in this discussion as its properties are conveniently available in table form. Water is frequently used for air conditioning in which the temperature range is small. Appendix D gives properties of the common refrigerants.

Figure 263 shows a flow diagram of a typical domestic compression refrigerator.

234. Dry versus Wet Compression.—When the compression process takes place wholly within the superheated region as from *b* to *c*, Fig. 262, the process is called *dry compression.* When the compression is begun before evaporation is complete and the vapor is wet during the entire compression, the process is called

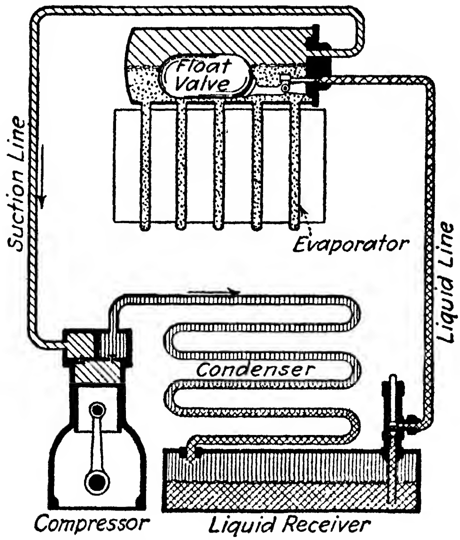


FIG. 263.—Typical vapor-compression domestic refrigerator.

wet compression. Dry compression possesses the following advantages:

1. Higher volumetric efficiency of the compressor.
2. Higher mechanical efficiency due to item 1.
3. Less danger of damage to compressor from presence of liquid.
4. Higher compressor speeds with safety, and thus a reduced first cost of motor and compressor.
5. Greater refrigerating effect per unit weight of fluid circulated through the system.

These advantages are sufficient to make it the type generally used.

Wet compression has the following advantages:

1. Less cooling water is required for the same amount of refrigeration than when high superheat results.
2. Easier to lubricate the cylinder when using ammonia, as a mixture of oil and ammonia clings to the cylinder walls.

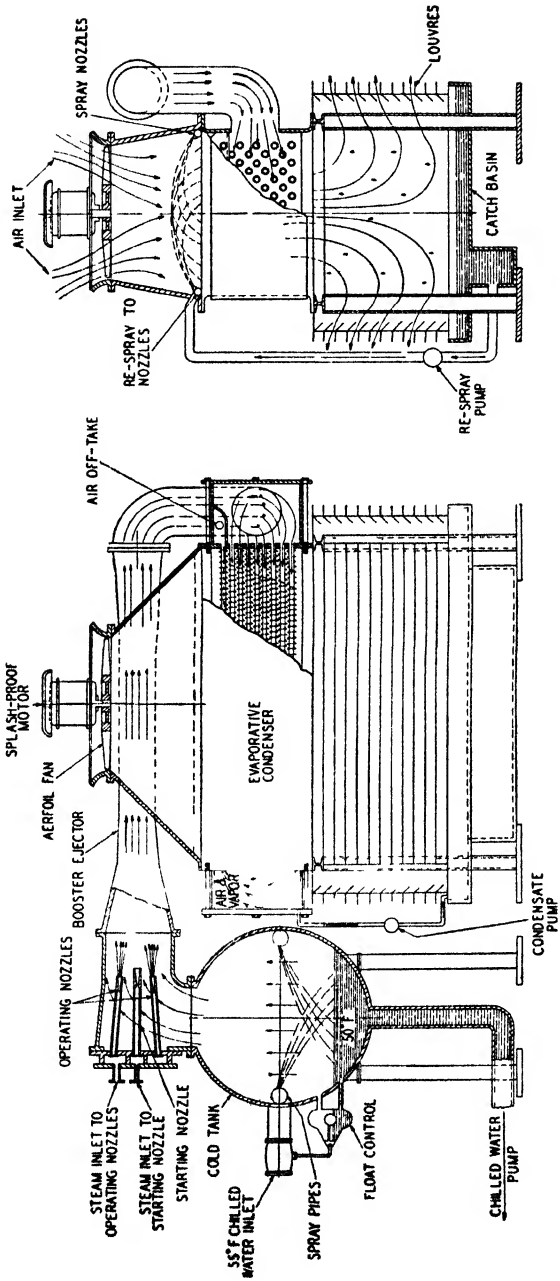


FIG 264.—Westinghouse water-vapor refrigeration system

The compressor is the most important piece of equipment in the refrigerating plant because (1) the performance and cost of operation are directly dependent upon it and (2) it is the mechanism that is subject to wear at vital points which may permit leakage of the refrigerant and thereby possibly cause injury to people.

235. Water-vapor Refrigeration.—The canvas water bag carried by desert travelers is a primitive but effective form of water-vapor refrigeration in which the evaporation from the surface of the bag cools the bag and its contents. The same principle is employed in commercial water-vapor refrigeration. Figure 264 shows a diagram of a modern system, the operation of which is exactly analogous to the mechanical-compressor system. Water in the cold tank is chilled by evaporation into the space above the water surface because of the low pressure maintained by the steam-jet ejectors at the top. The ejectors carry the vapor and the steam from the operating nozzles into the evaporative condenser. This, in turn, is cooled by means of water sprays over which a current of air is forced by the airfoil fan at the top. The chilled water is circulated by means of a centrifugal pump to the cooling chamber where the temperature is raised and is then returned to the cold tank where it is again cooled. Make-up water must be added to replace the evaporation in the cold tank where the level is maintained by a float valve.

The weight of water that flashes into vapor per pound of water entering the cold tank is that necessary to cool the remainder at constant enthalpy for the pound. That is,

$$h_{f1} = h_2 = h_{f2} + x_2 h_{fg2}$$

from which x_2 may be calculated, the subscripts 1 and 2 referring to the higher and lower temperatures, respectively. The volume of vapor handled by the first ejector is

$$V = x_2 V_{g2}$$

236. Absorption-refrigeration Systems.—In the ammonia-absorption process of refrigeration, heat is supplied directly to the refrigeration machine to accomplish the refrigerating effect. The condenser, expansion valve, and refrigeration coil are still a part of the equivalent and have the same functions as in the vapor-compression system. The essential difference between the compression and absorption systems is the replacement of the compressor by three pieces of apparatus, the absorber, the aqua ammonia pump, and the generator.

If the absorber, pump, and generator are to replace the compressor, the pressure and temperature of the ammonia vapor must be raised in passing through this equipment while flowing from the expansion coil to the condenser. The possibility of doing this depends upon the properties of a solution of ammonia in water or aqua ammonia. If the temperature of such a solution of a given concentration is increased at constant pressure, a temperature will be reached at which the water is unable to hold

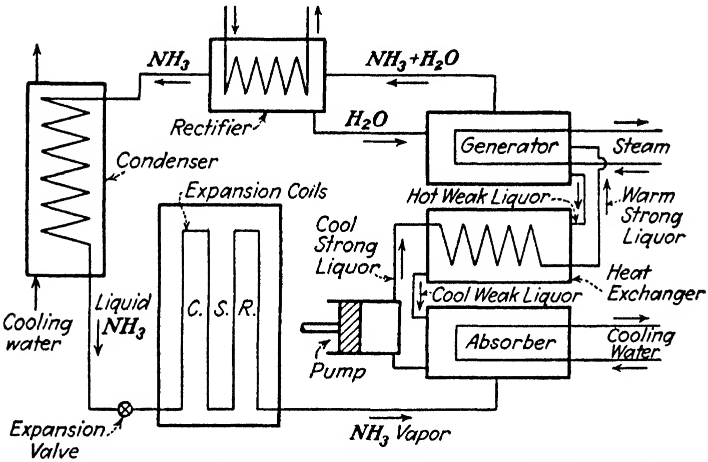


FIG. 265.—Flow diagram of ammonia absorption-refrigeration system.

as much ammonia in solution as at the lower temperature, and as a result ammonia vapor and a small amount of water vapor also will escape from the solution. In other words, water can hold increasing amounts of ammonia in solution either as the pressure on the solution increases or as the temperature of the solution decreases; that is, as the pressure increases, the solubility increases and, as the temperature rises, the ammonia is progressively driven out of solution.

Figure 265 shows the flow diagram for a commercial absorption system.

In the generator, heat is usually supplied to the aqua ammonia by circulating steam through coils, which are immersed in the solution. The original concentration of the ammonia in the water may be so chosen that, for the temperature of the solution as fixed by the temperature of the steam supplied, some of the ammonia vapor will be driven off from the solution and will pass to the condenser. This leaves in the generator a weak aqua ammonia solution, and, if the ammonia vapor is to be supplied steadily to the condenser, the generator must be constantly filled with a stronger solution furnished by the absorber and pump.

The absorber is arranged so that the weak solution, which has a greater density than the strong, may flow directly from the bottom of the generator to the absorber. The ammonia vapor as it comes from the expansion coil at a low pressure and temperature is brought into contact with this weak solution, which because of its low temperature is able to absorb the ammonia vapor, thereby increasing the ammonia concentration of the solution. Since heat is given off during this absorption of ammonia vapor, cold water must be circulated through the absorber in order to maintain a temperature which will be low enough to permit the absorption of the vapor by the weak solution. The pump then delivers the strong solution to the generator, and the cycle of the aqua ammonia is thereby completed. The weight of strong solution flowing from the absorber will be greater than the weight of weak solution flowing into it by the weight of ammonia which passes through the condenser and expansion valve; the weight of flow in the system is therefore balanced.

A similar balance must exist in the heat flow. The amount of heat Q_2 absorbed in the cold room will be similar in amount to that for the vapor-compression system. This heat together with the heat of solution of the ammonia is almost immediately removed by the cooling coils in the absorber. It might be stated at this point that, when vapor enters into solution, heat is generated from two sources. The first of these is due to the reduction of the vapor to the form of a liquid and the amount of heat generated is equal to the difference between the enthalpy of the original vapor and the enthalpy of the liquid ammonia at the temperature of the solution; the second is a chemical effect known as the heat of solution. The amount of heat removed by the cooling water in the absorber will be designated Q_A. In the generator an amount of heat Q_G enters the system; this amount is

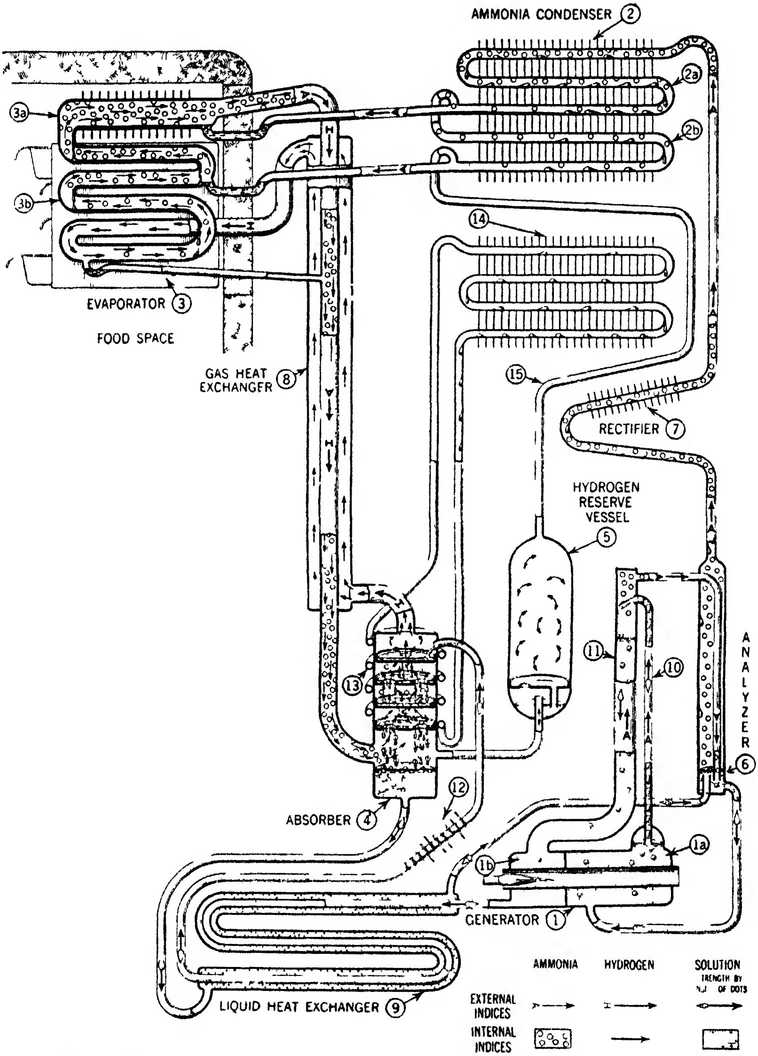


FIG. 266.—Flow diagram of Electrolux absorption-refrigeration unit (*Copyright by Servel, Inc. Used by permission*)

OPERATING PRINCIPLE Heat applied by flame in generator 1 expels ammonia from solution Ammonia vapor flows through cool analyzer 6 and rectifier 7 where water is condensed out, thence to condenser 2 Liquid ammonia flows by gravity through a trap to evaporator 3 and the vapor flows downward to absorber 4 Strong liquor then flows back to generator 1 Hydrogen atmosphere reduces partial pressure of ammonia. 13 and 14 constitute a secondary cooling system for the absorber using a volatile fluid Heat exchangers 8 and 9 conserve heat and cold equal to the sum of the heat of solution and the heat of vaporization of the ammonia as it leaves the solution. Then the ammonia vapor is condensed to a liquid in the condenser, an amount of heat Q_C being removed during the process.

Neglecting minor heat losses, the heat balance may be written thus:

$$\text{Heat added} = \text{heat removed}$$
$$Q_2 + Q_G = Q_A + Q_C$$

In addition to suitable pumps for circulation of the liquor two pieces of auxiliary equipment must be used with the system:

1. *Exchanger.*—The weak solution leaves the generator at high temperature and the strong solution leaves the absorber at low temperature, the solution being made to flow in opposite directions through an exchanger; the transfer of heat effected improves the performance.

2. *Rectifier.*—A small preliminary water-cooled condenser is interposed between the generator and condenser to cool the vapor so as to condense most of the water vapor, which is returned to the generator.

The absorption system has been applied very successfully to the Electrolux domestic refrigerator, a diagram of which is shown in Fig. 266.

A system analogous to the ammonia-absorption system is the silica gel adsorption system used chiefly for air cooling in air-conditioning systems. Silica gel is a very porous form of silicon dioxide having the power of adsorbing the refrigerant during one part of the cycle and of giving it off during the regenerating process which consists of heating. Since the silica gel is in the form of sand, it is not mobile in the apparatus and the process must be an intermittent one.

Problems

1. A refrigerating machine has a coefficient of performance of 5 and requires 40 hp to drive it. What is the capacity of the machine in tons of refrigeration?

2. If a large compressor requires 50 hp to drive it while producing 1 ton ice per hour, find the coefficient of performance if the ice left the plant at a temperature of 20°F. and the water entered at a temperature of 65°F Specific heat of ice equals 0.5.

3. Ammonia vapor has a pressure of 10 lb per sq in. abs and a temperature of 20°F. Find by aid of the National Bureau of Standards Mollier chart of Properties of Ammonia (*a*) the specific volume, enthalpy, and entropy of the vapor; (*b*) determine its degree of superheat and its mean value of c_p from saturated vapor.

4. (*a*) Calculate the rate of heat absorption from ammonia condensed at the rate of 200 lb per hr at the constant pressure of 153 lb per sq in. abs, the vapor entering the condenser at 180°F.; (*b*) find the temperature and enthalpy of the liquid condensate leaving the condenser; (*c*) if the liquid was now cooled at constant pressure to 50°F., find its enthalpy; (*d*) if the liquids in the two states in (*b*) and (*c*), respectively, now have their pressures reduced to 20 lb per sq in. abs, by passing through a perfectly insulated expansion valve, find the resulting qualities.

5. An ammonia compressor produces 5 standard tons refrigeration daily (24 hr) while operating with a suction pressure of 20 lb per sq in. abs and a discharge pressure of 140 lb per sq in. abs. Saturated liquid enters the expansion valve and saturated vapor enters the compressor. Determine: (*a*) the quality of the mixture entering the expansion coils; (*b*) the weight of ammonia that must be circulated per hour; (*c*) the actual coefficient of performance, if compression takes place adiabatically.

6. Ammonia enters a compressor with a pressure of 30 lb per sq in. abs and a temperature of 20°F. is compressed adiabatically to a discharge pressure of 125 lb per sq in. abs. Temperature of liquid entering the expansion valve is 60°F.; vapor leaves the expansion coils with a quality of 95 per cent. Two pounds of ammonia is circulated per hour. Determine: (*a*) temperature in the expansion coils; (*b*) actual coefficient of performance; (*c*) capacity of the machine in tons; (*d*) horsepower required to drive the compressor; (*e*) theoretical (Carnot) coefficient of performance; (*f*) efficiency of the machine.

7. In a 10-ton air-refrigeration machine the pressure and temperature at entrance to the expansion cylinder are 70 lb per sq in. abs and 70°F., respectively. Expansion takes place adiabatically in the cylinder to a pressure of 14 lb per sq in. abs maintained constant in the cold room. A temperature of 25°F. is to be kept constant in the cold room (at exit). Assuming adiabatic compression in the compressor to the expansion-cylinder pressure of 70 lb per sq in. abs, what will be the net work in Btu?

8. Ammonia enters a compressor with a pressure of 24 lb per sq in. abs. and a temperature of 10°F., compressed isentropically to a discharge pressure of 150 lb per sq in. abs. If the machine is to produce 10 standard tons of refrigeration daily (24 hr), find: (*a*) weight of ammonia that must be circulated per hour; (*b*) power delivered to ammonia by compressor; (*c*) the ideal Carnot coefficient of performance; (*d*) the actual coefficient of performance; (*e*) quality of mixture entering expansion coils.

9. An ammonia compression machine with dry compression operates between the pressures of 35 and 175 lb per sq in. abs. (*a*) What is the temperature in the expansion coils? (*b*) What is the coefficient of performance? (*c*) If 500 lb ammonia is circulated per hour, what is the capacity of the machine, and what horsepower is required? (*d*) If the compressor is single acting and makes 150 rpm what approximate cylinder volume is required?

10. Repeat Prob. 9 for wet compression.

APPENDIX A

COMBUSTION CALCULATION SHEET[1]

NOTE: Use red pencil for the fuel used, the given data, and calculation basis assumed. Use black pencil for the derived or calculated data.

AIR		
Subst.	Mols.	$\frac{20.9}{79.1} N_2$ Lb.
O_2		
N_2		
Air		

FURNACE OR ENGINE

FUEL: Basis assumed:			
Subst.	%	Mols.	Lb.
C			
H_2			
O_2			
N_2			
S			
Fuel	100.0		

Total weight in:

OXYGEN IN:
O_2 from fuel________Mols.
O_2 from air________Mols.
Total O_2 in________Mols

DRY COMBUSTION PRODUCTS Basis assumed:					
Dry Prod.	Mols.	Orsat %	Molal Sp. Ht.	Btu. Loss per °F	Lb.
CO_2					
O_2					
CO					
N_2					
Total					

SO_2 ________ Mols, ________

H_2O ________ Mols; ________

Total weight out·

OXYGEN OUT:
O_2 used for C________Mols
O_2 used for S________Mols
O_2 used for H_2________Mols
Excess O_2 out________Mols
Total O_2 out________Mols

1. Cu ft air per cu ft fuel in: ________ cu ft
2. Lb air per lb fuel in: ________ lb
3. Percentage excess air: ________ %
4. Temperature of entering substances, T_r, ________ °F.; leaving, T_g, ________ °F.
5. Heat losses per lb or per cu ft of fuel:
   (a) Due to dry gases: ________ Btu.
   (b) Due to CO: ________ Btu.
   (c) Due to H_2O from H_2: ________ Btu.

[1] For discussion of this chart see *Power*, 77: 600, 652.

APPENDIX B

STEAM TABLES

Table VII.—Properties of Saturated Steam (Temperatures)*

Temp. Fahr., t	Abs. press., lb. per sq. in., p	Specific volume			Enthalpy			Entropy			Temp., Fahr., t
		Sat. liquid, v_f	Evap., v_{fg}	Sat. vapor, v_g	Sat. liquid, h_f	Evap., h_{fg}	Sat. vapor, h_g	Sat. liquid, S_f	Evap., S_{fg}	Sat. vapor, S_g	
32	0.08854	0.01602	3306	3306	0.00	1075.8	1075.8	0.0000	2.1877	2.1877	32
35	0.09995	0.01602	2947	2947	3.02	1074.1	1077.1	0.0061	2.1709	2 1770	35
40	0.12170	0.01602	2444	2444	8.05	1071.3	1079.3	0.0162	2.1435	2.1597	40
45	0.14752	0.01602	2036.4	2036.4	13.06	1068.4	1081.5	0.0262	2.1167	2.1429	45
50	0.17811	0.01603	1703.2	1703.2	18.07	1065.6	1083.7	0.0361	2.0903	2.1264	50
60	0.2563	0.01604	1206.6	1206.7	28.06	1059.9	1088.0	0.0555	2.0393	2.0948	60
70	0.3631	0.01606	867.8	867.9	38.04	1054.3	1092.3	0.0745	1.9902	2.0647	70
80	0.5069	0.01608	633.1	633.1	48.02	1048.6	1096.6	0.0932	1.9428	2 0360	80
90	0.6982	0.01610	468.0	468.0	57.99	1042.9	1100.9	0.1115	1.8972	2.0087	90
100	0.9492	0.01613	350.3	350.4	67.97	1037.2	1105.2	0.1295	1.8531	1.9826	100
110	1.2748	0.01617	265.3	265.4	77.94	1031.6	1109.5	0.1471	1.8106	1.9577	110
120	1.6924	0.01620	203.25	203.27	87.92	1025.8	1113.7	0.1645	1.7694	1.9339	120
130	2.2225	0.01625	157.32	157.34	97.90	1020.0	1117.9	0.1816	1.7296	1.9112	130
140	2.8886	0.01629	122.99	123.01	107.89	1014.1	1122.0	0.1984	1.6910	1.8894	140
150	3.718	0.01634	97.06	97.07	117.89	1008.2	1126.1	0.2149	1.6537	1.8685	150
160	4.741	0.01639	77.27	77.29	127.89	1002.3	1130.2	0.2311	1.6174	1.8485	160
170	5.992	0.01645	62.04	62.06	137.90	996.3	1134.2	0.2472	1.5822	1.8293	170
180	7.510	0.01651	50.21	50.23	147.92	990.2	1138.1	0.2630	1.5480	1 8109	180
190	9.339	0.01657	40.94	40.96	157.95	984.1	1142.0	0.2785	1.5147	1.7932	190
200	11.526	0.01663	33.62	33.64	167.99	977.9	1145.9	0.2938	1.4824	1.7762	200
210	14.123	0.01670	27.80	27.82	178.05	971.6	1149.7	0.3090	1.4508	1.7598	210
212	14 696	0.01672	26.78	26.80	180.07	970.3	1150.4	0.3120	1.4446	1.7566	212
220	17.186	0.01677	23.13	23.15	188.13	965.2	1153.4	0.3229	1.4201	1 7440	220
230	20 780	0.01684	19.365	19.382	198.23	958.8	1157.0	0.3387	1.3901	1.7288	230
240	24.969	0.01692	16.306	16.323	208.34	952.2	1160.5	0.3531	1.3609	1.7140	240
250	29.825	0.01700	13.804	13.821	218.48	945.5	1164.0	0.3675	1.3323	1.6998	250
260	35.429	0.01709	11.746	11.763	228.64	938.7	1167.3	0.3817	1.3043	1.6860	260
270	41.858	0.01717	10.044	10.061	238.84	931.8	1170.6	0.3958	1.2769	1.6727	270
280	49.203	0.01726	8.628	8.645	249.06	924.7	1173.8	0.4096	1.2501	1.6597	280
290	57.556	0.01735	7.444	7.461	259.31	917.5	1176.8	0.4234	1.2238	1.6472	290
300	67.013	0.01745	6.449	6.466	269.59	910.1	1179.7	0.4369	1.1980	1.6350	300
310	77.68	0.01755	5.609	5.626	279.92	902.6	1182.5	0.4504	1.1727	1.6231	310
320	89.66	0.01765	4.896	4.914	290.28	894.9	1185.2	0.4637	1.1478	1.6115	320
330	103.06	0.01776	4.289	4.307	300.68	887.0	1187.7	0.4769	1.1233	1.6002	330
340	118.01	0.01787	3.770	3.788	311.13	879.0	1190.1	0.4900	1.0992	1.5891	340
350	134.63	0.01799	3.324	3.342	321.63	870.7	1192.3	0.5029	1.0754	1.5783	350
360	153.04	0.01811	2.939	2.957	332.18	862.2	1194.4	0.5158	1.0519	1.5677	360
370	173.37	0.01823	2.606	2.625	342.79	853.5	1196.3	0.5286	1.0287	1.5573	370
380	195.77	0.01836	2.317	2.335	353.45	844.6	1198.1	0.5413	1.0059	1.5471	380
390	220.37	0.01850	2.0651	2.0836	364.17	835.4	1199.6	0.5539	0.9832	1.5371	390
400	247.31	0.01864	1.8447	1.8633	374.97	826.0	1201.0	0.5664	0.9608	1.5272	400
410	276.75	0.01878	1.6512	1.6700	385.83	816.3	1202.1	0.5788	0.9386	1.5174	410
420	308.83	0.01894	1.4811	1.5000	396.77	806.3	1203.1	0.5912	0.9166	1.5078	420
430	343.72	0.01910	1.3308	1.3429	407.79	796.0	1203.8	0.6035	0.8947	1.4982	430
440	381 59	0.01926	1.1979	1.2171	418.90	785.4	1204.3	0.6158	0.8730	1.4887	440
450	422.6	0.0194	1.0799	1.0993	430.1	774.5	1204.6	0.6280	0.8513	1.4793	450
460	466.9	0.0196	0.9748	0.9944	441.4	763.2	1204.6	0.6402	0.8298	1.4700	460
470	514.7	0.0198	0.8811	0.9009	452.8	751.5	1204.3	0.6523	0.8083	1.4606	470
480	566.1	0.0200	0.7972	0.8172	464.4	739.4	1203.7	0.6645	0.7868	1.4513	480
490	621.4	0.0202	0.7221	0.7423	476.0	726.8	1202.8	0.6766	0.7653	1.4419	490
500	680.8	0.0204	0.6545	0.6749	487.8	713.9	1201.7	0.6887	0.7438	1.4325	500
520	812.4	0.0209	0.5385	0.5594	511.9	686.4	1198.2	0.7130	0.7006	1.4136	520
540	962.5	0.0215	0.4434	0.4649	536.6	656.6	1193.2	0.7374	0.6568	1.3942	540
560	1133.1	0.0221	0.3647	0.3868	562.2	624.2	1186.4	0.7621	0.6121	1.3742	560
580	1325.8	0.0228	0.2989	0.3217	588.9	588.4	1177.3	0.7872	0.5659	1.3532	580
600	1542.9	0.0236	0.2432	0.2668	617.0	548.5	1165.5	0.8131	0.5176	1.3307	600
620	1786.6	0.0247	0.1955	0.2201	646.7	503.6	1150.3	0.8398	0.4664	1.3062	620
640	2059.7	0.0260	0.1538	0.1798	678.6	452.0	1130.5	0.8679	0.4110	1.2789	640
660	2365.4	0.0278	0.1165	0.1442	714.2	390.2	1104.4	0.8987	0.3485	1.2472	660
680	2708.1	0.0305	0.0810	0.1115	757.3	309.9	1067.2	0.9351	0.2719	1.2071	680
700	3093.7	0.0369	0.0392	0.0761	823.3	172.1	995.4	0.9905	0.1484	1.1389	700
705.4	3206.2	0.0503	0	0.0503	902.7	0	902.7	1.0580	0	1.0580	705.4

* Abridged from "Thermodynamic Properties of Steam," by Joseph H. Keenan and Frederick G. Keyes, John Wiley & Sons, Inc., New York.

TABLE VIII.—PROPERTIES OF SATURATED STEAM (PRESSURES)*

Abs press, lb per sq in, p	Temp Fahr, t	Specific volume		Enthalpy			Entropy			Internal energy		Abs press, lb per sq in, p
		Sat liquid, v_f	Sat vapor, v_g	Sat liquid, h_f	Evap h_{fg}	Sat vapor, h_g	Sat liquid, s_f	Evap s_{fg}	Sat vapor s_g	Sat liquid, u_f	Sat vapor, u_g	
1.0	101.74	0.01614	333.6	69.70	1036.3	1106.0	0.1326	1.8456	1.9782	69.70	1044.3	1.0
2.0	126.08	0.01623	173.73	93.99	1022.2	1116.2	0.1749	1.7451	1.9200	93.98	1051.9	2.0
3.0	141.48	0.01630	118.71	109.37	1013.2	1122.6	0.2008	1.6855	1.8863	109.36	1056.7	3.0
4.0	152.97	0.01636	90.63	120.86	1006.4	1127.3	0.2198	1.6427	1.8625	120.85	1060.2	4.0
5.0	162.24	0.01640	73.52	130.13	1001.0	1131.1	0.2347	1.6094	1.8441	130.12	1063.1	5.0
6.0	170.06	0.01645	61.98	137.96	996.2	1134.2	0.2472	1.5820	1.8292	137.94	1065.4	6.0
7.0	176.85	0.01649	53.64	144.76	992.1	1136.9	0.2581	1.5586	1.8167	144.74	1067.4	7.0
8.0	182.86	0.01653	47.34	150.79	988.5	1139.3	0.2674	1.5383	1.8057	150.77	1069.2	8.0
9.0	188.28	0.01656	42.40	156.22	985.2	1141.4	0.2759	1.5203	1.7962	156.19	1070.8	9.0
10	193.21	0.01659	38.42	161.17	982.1	1143.3	0.2835	1.5041	1.7876	161.14	1072.2	10
14.696	212.00	0.01672	26.80	180.07	970.3	1150.4	0.3120	1.4446	1.7566	180.02	1077.5	14.696
15	213.03	0.01672	26.29	181.11	969.7	1150.8	0.3135	1.4415	1.7549	181.06	1077.8	15
20	227.96	0.01683	20.089	196.16	960.1	1156.3	0.3356	1.3962	1.7319	196.10	1081.9	20
25	240.07	0.01692	16.303	208.42	952.1	1160.6	0.3533	1.3606	1.7139	208.34	1085.1	25
30	250.33	0.01701	13.746	218.82	945.3	1164.1	0.3680	1.3313	1.6993	218.73	1087.8	30
35	259.28	0.01708	11.898	227.91	939.2	1167.1	0.3807	1.3063	1.6870	227.80	1090.1	35
40	267.25	0.01715	10.498	236.03	933.7	1169.7	0.3919	1.2844	1.6763	235.90	1092.0	40
45	274.44	0.01721	9.401	243.36	928.6	1172.0	0.4019	1.2650	1.6669	243.22	1093.7	45
50	281.01	0.01727	8.515	250.09	924.0	1174.1	0.4110	1.2474	1.6585	249.93	1095.3	50
55	287.07	0.01732	7.787	256.30	919.6	1175.9	0.4193	1.2316	1.6509	256.12	1096.7	55
60	292.71	0.01738	7.175	262.09	915.5	1177.6	0.4270	1.2168	1.6438	261.90	1097.9	60
65	297.97	0.01743	6.655	267.50	911.6	1179.1	0.4342	1.2032	1.6374	267.29	1099.1	65
70	302.92	0.01748	6.206	272.61	907.9	1180.6	0.4409	1.1906	1.6315	272.38	1100.2	70
75	307.60	0.01753	5.816	277.43	904.5	1181.9	0.4472	1.1787	1.6259	277.19	1101.2	75
80	312.03	0.01757	5.472	282.02	901.1	1183.1	0.4531	1.1676	1.6207	281.76	1102.1	80
85	316.25	0.01761	5.168	286.39	897.8	1184.2	0.4587	1.1571	1.6158	286.11	1102.9	85
90	320.27	0.01766	4.896	290.56	894.7	1185.3	0.4641	1.1471	1.6112	290.27	1103.7	90
95	324.12	0.01770	4.652	294.56	891.7	1186.2	0.4692	1.1376	1.6068	294.25	1104.5	95
100	327.81	0.01774	4.432	298.40	888.8	1187.2	0.4740	1.1286	1.6026	298.08	1105.2	100
110	334.77	0.01782	4.049	305.66	883.2	1188.9	0.4832	1.1117	1.5948	305.30	1106.5	110
120	341.25	0.01789	3.728	312.44	877.9	1190.4	0.4916	1.0962	1.5878	312.05	1107.6	120
130	347.32	0.01796	3.455	318.81	872.9	1191.7	0.4995	1.0817	1.5812	318.38	1108.6	130
140	353.02	0.01802	3.220	324.82	868.2	1193.0	0.5069	1.0682	1.5751	324.35	1109.6	140
150	358.42	0.01809	3.015	330.51	863.6	1194.1	0.5138	1.0556	1.5694	330.01	1110.5	150
160	363.53	0.01815	2.834	335.93	859.2	1195.1	0.5204	1.0436	1.5640	335.39	1111.2	160
170	368.41	0.01822	2.675	341.09	854.9	1196.0	0.5266	1.0324	1.5590	340.52	1111.9	170
180	373.06	0.01827	2.532	346.03	850.8	1196.9	0.5325	1.0217	1.5542	345.42	1112.5	180
190	377.51	0.01833	2.404	350.79	846.8	1197.6	0.5381	1.0116	1.5497	350.15	1113.1	190
200	381.79	0.01839	2.288	355.36	843.0	1198.4	0.5435	1.0018	1.5453	354.68	1113.7	200
250	400.95	0.01865	1.8438	376.00	825.1	1201.1	0.5675	0.9588	1.5263	375.14	1115.8	250
300	417.33	0.01890	1.5433	393.84	809.0	1202.8	0.5879	0.9225	1.5104	392.79	1117.1	300
350	431.72	0.01913	1.3260	409.69	794.2	1203.9	0.6056	0.8910	1.4966	408.45	1118.0	350
400	444.59	0.0193	1.1613	424.0	780.5	1204.5	0.6214	0.8630	1.4844	422.6	1118.5	400
450	456.28	0.0195	1.0320	437.2	767.4	1204.6	0.6356	0.8378	1.4734	435.5	1118.7	450
500	467.01	0.0197	0.9278	449.4	755.0	1204.4	0.6487	0.8147	1.4634	447.6	1118.6	500
550	476.94	0.0199	0.8424	460.8	743.1	1203.9	0.6608	0.7934	1.4542	458.8	1118.2	550
600	486.21	0.0201	0.7698	471.6	731.6	1203.2	0.6720	0.7734	1.4454	469.4	1117.7	600
650	494.90	0.0203	0.7083	481.8	720.5	1202.3	0.6826	0.7548	1.4374	479.4	1117.1	650
700	503.10	0.0205	0.6554	491.5	709.7	1201.2	0.6925	0.7371	1.4296	488.8	1116.3	700
750	510.86	0.0207	0.6092	500.8	699.2	1200.0	0.7019	0.7204	1.4223	498.0	1115.4	750
800	518.23	0.0209	0.5687	509.7	688.9	1198.6	0.7108	0.7045	1.4153	506.6	1114.4	800
850	525.26	0.0210	0.5327	518.3	678.8	1197.1	0.7194	0.6891	1.4085	515.0	1113.3	850
900	531.98	0.0212	0.5006	526.6	668.8	1195.4	0.7275	0.6744	1.4020	523.1	1112.1	900
950	538.43	0.0214	0.4717	534.6	659.1	1193.7	0.7355	0.6602	1.3957	530.9	1110.8	950
1000	544.61	0.0216	0.4456	542.4	649.4	1191.8	0.7430	0.6467	1.3897	538.4	1109.4	1000
1100	556.31	0.0220	0.4001	557.4	630.4	1187.8	0.7575	0.6205	1.3780	552.9	1106.4	1100
1200	567.22	0.0223	0.3619	571.7	611.7	1183.4	0.7711	0.5956	1.3667	566.7	1103.0	1200
1300	577.46	0.0227	0.3293	585.4	593.2	1178.6	0.7840	0.5719	1.3559	580.0	1099.4	1300
1400	587.10	0.0231	0.3012	598.7	574.7	1173.4	0.7963	0.5491	1.3454	592.7	1095.4	1400
1500	596.23	0.0235	0.2765	611.6	556.3	1167.9	0.8082	0.5269	1.3351	605.1	1091.2	1500
2000	635.82	0.0257	0.1878	671.7	463.4	1135.1	0.8619	0.4230	1.2849	662.2	1065.6	2000
2500	668.13	0.0287	0.1307	730.6	360.5	1091.1	0.9126	0.3197	1.2322	717.3	1030.6	2500
3000	695.36	0.0346	0.0858	802.5	217.8	1020.3	0.9731	0.1885	1.1615	783.4	972.7	3000
3206.2	705.40	0.0503	0.0503	902.7	0	902.7	1.0580	0	1.0580	872.9	872.9	3206.2

* Abridged from 'Thermodynamic Properties of Steam' by Joseph H Keenan and Frederick G Keyes John Wiley & Sons Inc New York

TABLE IX.—PROPERTIES OF SUPERHEATED STEAM*

Abs. press., lb. per sq. in. (sat. temp.)		Temperature-degrees Fahrenheit												
		200°	300°	400°	500°	600°	700°	800°	900°	1000°	1100°	1200°	1400°	1600°
	v	392.6	452.3	512.0	571.6	631.2	690.8	750.4	809.9	869.5	929.1	988.7	1107.8	1227.0
1	h	1150.4	1195.8	1241.7	1288.3	1335.7	1383.8	1432.8	1482.7	1533.5	1585.2	1637.7	1745.7	1857.5
(101.74)	s	2.0512	2.1153	2.1720	2.2233	2.2702	2.3137	2.3542	2.3923	2.4283	2.4625	2.4952	2.5566	2.6137
	v	78.16	90.25	102.26	114.22	126.16	138.10	150.03	161.95	173.87	185.79	197.71	221.6	245.4
5	h	1148.8	1195.0	1241.2	1288.0	1335.4	1383.6	1432.7	1482.6	1533.4	1585.1	1637.7	1745.7	1857.4
(162.24)	s	1.8718	1.9370	1.9942	2.0456	2.0927	2.1361	2.1767	2.2148	2.2509	2.2851	2.3178	2.3792	2.4363
	v	38.85	45.00	51.04	57.05	63.03	69.01	74.98	80.95	86.92	92.88	98.84	110.77	122.69
10	h	1146.6	1193.9	1240.6	1287.5	1335.1	1383.4	1432.5	1482.4	1533.2	1585.0	1637.6	1745.6	1857.3
(193.21)	s	1.7927	1.8595	1.9172	1.9689	2.0160	2.0596	2.1002	2.1383	2.1744	2.2086	2.2413	2.3028	2.3598
	v		30.53	34.68	38.78	42.86	46.94	51.00	55.07	59.13	63.19	67.25	75.37	83.48
14.696	h		1192.8	1239.9	1287.1	1334.8	1383.2	1432.3	1482.3	1533.1	1584.8	1637.5	1745.5	1857.3
(212.00)	s		1.8160	1.8743	1.9261	1.9734	2.0170	2.0576	2.0958	2.1310	2.1662	2.1989	2.2603	2.3174
	v		22.36	25.43	28.46	31.47	34.47	37.46	40.45	43.44	46.42	49.41	55.37	61.34
20	h		1191.6	1239.2	1286.6	1334.4	1382.9	1432.1	1482.1	1533.0	1584.7	1637.4	1745.4	1857.2
(227.96)	s		1.7808	1.8396	1.8918	1.9392	1.9829	2.0235	2.0618	2.0978	2.1321	2.1648	2.2263	2.2834
	v		11.040	12.628	14.168	15.688	17.198	18.702	20.20	21.70	23.20	24.69	27.68	30.66
40	h		1186.8	1236.5	1284.8	1333.1	1381.9	1431.3	1481.4	1532.4	1584.3	1637.0	1745.1	1857.0
(267.25)	s		1.6994	1.7608	1.8140	1.8619	1.9058	1.9467	1.9850	2.0212	2.0555	2.0883	2.1498	2.2069
	v		7.259	8.357	9.403	10.427	11.441	12.449	13.452	14.454	15.453	16.451	18.446	20.44
60	h		1181.6	1233.6	1283.0	1331.8	1380.9	1430.5	1480.8	1531.9	1583.8	1636.6	1744.8	1856.7
(292.71)	s		1.6492	1.7135	1.7678	1.8162	1.8605	1.9015	1.9400	1.9762	2.0106	2.0434	2.1049	2.1621
	v			6.220	7.020	7.797	8.562	9.322	10.077	10.830	11.582	12.332	13.830	15.325
80	h			1230.7	1281.1	1330.5	1379.9	1429.7	1480.1	1531.3	1583.4	1636.2	1744.5	1856.5
(312.03)	s			1.6791	1.7346	1.7836	1.8281	1.8694	1.9079	1.9442	1.9787	2.0115	2.0731	2.1303
	v			4.937	5.589	6.218	6.835	7.446	8.052	8.656	9.259	9.860	11.060	12.258
100	h			1227.6	1279.1	1329.1	1378.9	1428.9	1479.5	1530.8	1582.9	1635.7	1744.2	1856.2
(327.81)	s			1.6518	1.7085	1.7581	1.8029	1.8443	1.8829	1.9193	1.9538	1.9867	2.0484	2.1056
	v			4.081	4.636	5.165	5.683	6.195	6.702	7.207	7.710	8.212	9.214	10.213
120	h			1224.4	1277.2	1327.7	1377.8	1428.1	1478.8	1530.2	1582.4	1635.3	1743.9	1856.0
(341.25)	s			1.6287	1.6869	1.7370	1.7822	1.8237	1.8625	1.8990	1.9335	1.9664	2.0281	2.0854
	v			3.468	3.954	4.413	4.861	5.301	5.738	6.172	6.604	7.035	7.895	8.752
140	h			1221.1	1275.2	1326.4	1376.8	1427.3	1478.2	1529.7	1581.9	1634.9	1743.5	1855.7
(353.02)	s			1.6087	1.6683	1.7190	1.7645	1.8063	1.8451	1.8817	1.9163	1.9493	2.0110	2.0683
	v			3.008	3.443	3.849	4.244	4.631	5.015	5.396	5.775	6.152	6.906	7.656
160	h			1217.6	1273.1	1325.0	1375.7	1426.4	1477.5	1529.1	1581.4	1634.5	1743.2	1855.5
(363.53)	s			1.5908	1.6519	1.7033	1.7491	1.7911	1.8301	1.8667	1.9014	1.9344	1.9962	2.0535
	v			2.649	3.044	3.411	3.764	4.110	4.452	4.792	5.129	5.466	6.136	6.804
180	h			1214.0	1271.0	1323.5	1374.7	1425.6	1476.8	1528.6	1581.0	1634.1	1742.9	1855.2
(373.06)	s			1.5745	1.6373	1.6894	1.7355	1.7776	1.8167	1.8534	1.8882	1.9212	1.9831	2.0404
	v			2.361	2.726	3.060	3.380	3.693	4.002	4.309	4.613	4.917	5.521	6.123
200	h			1210.3	1268.9	1322.1	1373.6	1424.8	1476.2	1528.0	1580.5	1633.7	1742.6	1855.0
(381.79)	s			1.5594	1.6240	1.6767	1.7232	1.7655	1.8048	1.8415	1.8763	1.9094	1.9713	2.0287
	v			2.125	2.465	2.772	3.066	3.352	3.634	3.913	4.191	4.467	5.017	5.565
220	h			1206.5	1266.7	1320.7	1372.6	1424.0	1475.5	1527.5	1580.0	1633.3	1742.3	1854.7
(389.86)	s			1.5453	1.6117	1.6652	1.7120	1.7545	1.7939	1.8308	1.8656	1.8987	1.9607	2.0181
	v			1.9276	2.247	2.533	2.804	3.068	3.327	3.584	3.839	4.093	4.597	5.100
240	h			1202.5	1264.5	1319.2	1371.5	1423.2	1474.8	1526.9	1579.6	1632.9	1742.0	1854.5
(397.37)	s			1.5319	1.6003	1.6546	1.7017	1.7444	1.7839	1.8209	1.8558	1.8889	1.9510	2.0084
	v				2.063	2.330	2.582	2.827	3.067	3.305	3.541	3.776	4.242	4.707
260	h				1262.3	1317.7	1370.4	1422.3	1474.2	1526.3	1579.1	1632.5	1741.7	1854.2
(404.42)	s				1.5897	1.6447	1.6922	1.7352	1.7748	1.8118	1.8467	1.8799	1.9420	1.9995

TABLE IX.—PROPERTIES OF SUPERHEATED STEAM.*—(*Continued*)

Abs. press., lb. per sq. in. (sat. temp.)		Temperature-degrees Fahrenheit												
		200°	300°	400°	500°	600°	700°	800°	900°	1000°	1100°	1200°	1400°	1600°
	v				1.9047	2.156	2.392	2.621	2.845	3.066	3.286	3.504	3.938	4.370
280	h				1260.0	1316.2	1369.4	1421.5	1473.5	1525.8	1578.6	1632.1	1741.4	1854.0
(411 05)	s				1.5796	1.6354	1.6834	1.7265	1.7662	1.8033	1.8383	1.8716	1.9337	1.9912
	v				1.7675	2.005	2.227	2.442	2.652	2.850	3.065	3.269	3.674	4.078
300	h				1257.6	1314.7	1368.3	1420.6	1472.8	1525.2	1578.1	1631.7	1741.0	1853.7
(417 33)	s				1.5701	1.6268	1.6751	1.7184	1.7582	1.7954	1.8305	1.8638	1.9260	1.9835
	v				1.4923	1.7036	1.8980	2.084	2.266	2.445	2.622	2.798	3.147	3.493
350	h				1251.5	1310.9	1365.5	1418.5	1471.1	1523.8	1577.0	1630.7	1740.3	1853.1
(431.72)	s				1.5481	1.6070	1.6563	1.7002	1.7403	1.7777	1.8130	1.8463	1.9086	1.9663
	v				1.2851	1.4770	1.6508	1.8161	1.9767	2.134	2.290	2.445	2.754	3.055
400	h				1245 1	1306.9	1362.7	1416.4	1469.4	1522.4	1575.8	1629.6	1739.5	1852.5
(444 59)	s				1.5281	1.5894	1.6398	1.6842	1.7247	1.7623	1.7977	1.8311	1.8936	1.9513

		500°	550°	600°	620°	640°	660°	680°	700°	800°	900°	1000°	1200°	1400°	1600°
	v	1.1231	1.2155	1.3005	1.3332	1.3652	1.3967	1.4278	1.4584	1.6074	1.7516	1.8928	2.170	2.443	2.714
450	h	1238.4	1272.0	1302.8	1314.6	1326.2	1337.5	1348.8	1359.9	1414.3	1467.7	1521.0	1628.6	1738.7	1851.9
(456.28)	s	1.5095	1.5437	1.5735	1.5845	1.5951	1.6054	1.6153	1.6250	1.6699	1.7108	1.7486	1.8177	1.8803	1.9381
	v	0.9927	1.0800	1.1591	1.1893	1.2188	1.2478	1.2763	1.3044	1.4405	1.5715	1.6996	1.9504	2.197	2.442
500	h	1231.3	1266.8	1298.6	1310.7	1322.6	1334.2	1345.7	1357.0	1412.1	1466.0	1519.6	1627.6	1737.9	1851.3
(467.01)	s	1.4919	1.5280	1.5588	1.5701	1.5810	1.5915	1.6016	1.6115	1.6571	1.6982	1.7363	1.8056	1.8683	1.9262
	v	0.8852	0.9686	1.0431	1.0714	1.0989	1.1259	1.1523	1.1783	1.3038	1.4241	1.5414	1.7706	1.9957	2.219
550	h	1223.7	1261.2	1294.3	1306.8	1318.9	1330.8	1342.5	1354.0	1409.9	1464.3	1518.2	1626.6	1737.1	1850.6
(476.94)	s	1.4751	1.5131	1.5451	1.5568	1.5680	1.5787	1.5890	1.5991	1.6452	1.6868	1.7250	1.7946	1.8575	1.9155
	v	0.7947	0.8753	0.9463	0.9729	0.9988	1.0241	1.0489	1.0732	1.1899	1.3013	1.4096	1.6208	1.8279	2.033
600	h	1215.7	1255.5	1289.9	1302.7	1315.2	1327.4	1339.3	1351.1	1407.7	1462.5	1516.7	1625.5	1736.3	1850.0
(486.21)	s	1.4586	1.4990	1.5323	1.5443	1.5558	1.5667	1.5773	1.5875	1.6343	1.6762	1.7147	1.7846	1.8476	1.9056
	v		0.7277	0.7934	0.8177	0.8411	0.8639	0.8860	0.9077	1.0108	1.1082	1.2024	1.3853	1.5641	1.7405
700	h		1243.2	1280.6	1294.3	1307.5	1320.3	1332.8	1345.0	1403.2	1459.0	1513.9	1623.5	1734.8	1848.8
(503.10)	s		1.4722	1.5084	1.5212	1.5333	1.5449	1.5559	1.5665	1.6147	1.6573	1.6963	1.7666	1.8299	1.8881
	v		0.6154	0.6779	0.7006	0.7223	0.7433	0.7635	0.7833	0.8763	0.9633	1.0470	1.2088	1.3662	1.5214
800	h		1229.8	1270.7	1285.4	1299.4	1312.9	1325.9	1338.6	1398.6	1455.4	1511.0	1621.4	1733.2	1847.5
(518.23)	s		1.4467	1.4863	1.5000	1.5129	1.5250	1.5366	1.5476	1.5972	1.6407	1.6801	1.7510	1.8146	1.8729
	v		0.5264	0.5873	0.6089	0.6294	0.6491	0.6680	0.6863	0.7716	0.8506	0.9262	1.0714	1.2124	1.3509
900	h		1215.0	1260.1	1275.9	1290.9	1305.1	1318.8	1332.1	1393.9	1451.8	1508.1	1619.3	1731 6	1846.3
(531.98)	s		1.4216	1.4653	1.4800	1.4938	1.5066	1.5187	1.5303	1.5814	1.6257	1.6656	1.7371	1.8009	1.8595
	v		0.4533	0.5140	0.5350	0.5546	0.5733	0.5912	0.6084	0.6878	0.7604	0.8294	0.9615	1.0893	1.2146
1000	h		1198.3	1248.8	1265.9	1281.9	1297.0	1311.4	1325.3	1389.2	1448.2	1505.1	1617.3	1730.0	1845.0
(544.61)	s		1.3961	1.4450	1.4610	1.4757	1.4893	1.5021	1.5141	1.5670	1.6121	1.6525	1.7245	1.7886	1.8474
	v			0.4532	0.4738	0.4929	0.5110	0.5281	0.5445	0.6191	0.6866	0.7503	0.8716	0.9885	1.1031
1100	h			1236.7	1255.3	1272.4	1288.5	1303.7	1318.3	1384.3	1444.5	1502.2	1615.2	1728.4	1843.8
(556.31)	s			1.4251	1.4425	1.4583	1.4728	1.4862	1.4989	1.5535	1.5995	1.6405	1.7130	1.7775	1.8363
	v			0.4016	0.4222	0.4410	0.4586	0.4752	0.4909	0.5617	0.6250	0.6843	0.7967	0.9046	1.0101
1200	h			1223.5	1243.9	1262.4	1279.6	1295.7	1311.0	1379.3	1440.7	1499.2	1613.1	1726.9	1842.5
(567.22)	s			1.4052	1.4243	1.4413	1.4568	1.4710	1.4843	1.5409	1.5879	1.6293	1.7025	1.7672	1.8263
	v			0.3174	0.3390	0.3580	0.3753	0.3912	0.4062	0.4714	0.5281	0.5805	0.6789	0.7727	0.8640
1400	h			1193.0	1218.4	1240.4	1260.3	1278.5	1295.5	1369.1	1433.1	1493.2	1608.9	1723.7	1840.0
(587.10)	s			1.3639	1.3877	1.4079	1.4258	1.4419	1.4567	1.5177	1.5666	1.6093	1.6836	1.7489	1.8083

TABLE IX.—PROPERTIES OF SUPERHEATED STEAM.*—(*Continued*)

Abs. press., lb. per sq. in. (sat. temp.)		Temperature-degrees Fahrenheit													
		500°	550°	600°	620°	640°	660°	680°	700°	800°	900°	1000°	1200°	1400°	1600°
	v				0.2733	0.2936	0.3112	0.3271	0.3417	0.4034	0.4553	0.5027	0.5906	0.6738	0.7545
1600	h				1187.8	1215.2	1238.7	1259.6	1278.7	1358.4	1425.3	1487.0	1604.6	1720.5	1837.5
(604.90)	s				1.3489	1.3741	1.3952	1.4137	1.4303	1.4964	1.5476	1.5914	1.6669	1.7328	1.7926
	v					0.2407	0.2597	0.2760	0.2907	0.3502	0.3986	0.4421	0.5218	0.5968	0.6693
1800	h					1185.1	1214.0	1238.5	1260.3	1347.2	1417.4	1480.8	1600.4	1717 3	1835.0
(621 03)	s					1.3377	1.3638	1.3855	1.4044	1.4765	1.5301	1.5752	1.6520	1.7185	1.7786
	v					0.1936	0.2161	0.2337	0.2489	0.3074	0.3532	0.3935	0.4668	0.5352	0.6011
2000	h					1145.6	1184.9	1214.8	1240.0	1335.5	1409.2	1474.5	1596.1	1714.1	1832.5
(635.82)	s					1.2945	1.3300	1.3564	1.3783	1.4576	1.5139	1.5603	1.6384	1.7055	1.7660
	v							0.1484	0.1686	0.2294	0.2710	0.3061	0.3678	0.4244	0.4784
2500	h							1132.3	1176.8	1303.6	1387.8	1458.4	1585.3	1706.1	1826.2
(668.13)	s							1.2687	1.3073	1.4127	1.4772	1.5273	1.6088	1.6775	1.7389
	v								0.0984	0.1760	0.2159	0.2476	0.3018	0.3505	0.3966
3000	h								1060 7	1267.2	1365.0	1441.8	1574.3	1698.0	1819.9
(695.36)	s								1.1966	1.3690	1.4439	1.4984	1.5837	1.6540	1.7163
	v									0.1583	0.1981	0.2288	0.2806	0.3267	0.3703
3206.2	h									1250.5	1355.2	1434.7	1569.8	1694.6	1817.2
(705.40)	s									1.3508	1.4309	1.4874	1.5742	1.6452	1.7080
	v								0.0306	0.1364	0.1762	0.2058	0.2546	0.2977	0.3381
3500	h								780.5	1224.9	1349.7	1424.5	1563.3	1689.8	1813.6
	s								0.9515	1.3241	1.4127	1.4723	1.5615	1.6336	1.6968
	v								0.0287	0.1052	0.1462	0.1743	0.2192	0.2581	0.2943
4000	h								763.8	1174.8	1314.4	1406.8	1552.1	1681.7	1807.2
	s								0.9347	1.2757	1.3827	1.4482	1.5417	1.6154	1.6795
	v								0.0276	0.0798	0.1226	0.1500	0.1917	0.2273	0.2602
4500	h								753.5	1113.9	1286.5	1388.4	1540.8	1673.5	1800.9
	s								0.9235	1.2204	1.3529	1.4253	1.5235	1.5990	1.6640
	v								0.0268	0.0593	0.1036	0.1303	0.1696	0.2027	0.2329
5000	h								746.4	1047.1	1256.5	1369.5	1529.5	1665.3	1794.5
	s								0.9152	1.1622	1.3231	1.4034	1.5066	1.5839	1.6499
	v								0.0262	0.0463	0.0880	0.1143	0.1516	0.1825	0.2106
5500	h								741.3	985.0	1224.1	1349.3	1518.2	1657.0	1788.1
	s								0.9090	1.1093	1.2930	1.3821	1.4908	1.5699	1.6369

* Abridged from "Thermodynamic Properties of Steam," by Joseph H. Keenan and Frederick G. Keyes, John Wiley & Sons, Inc., New York.

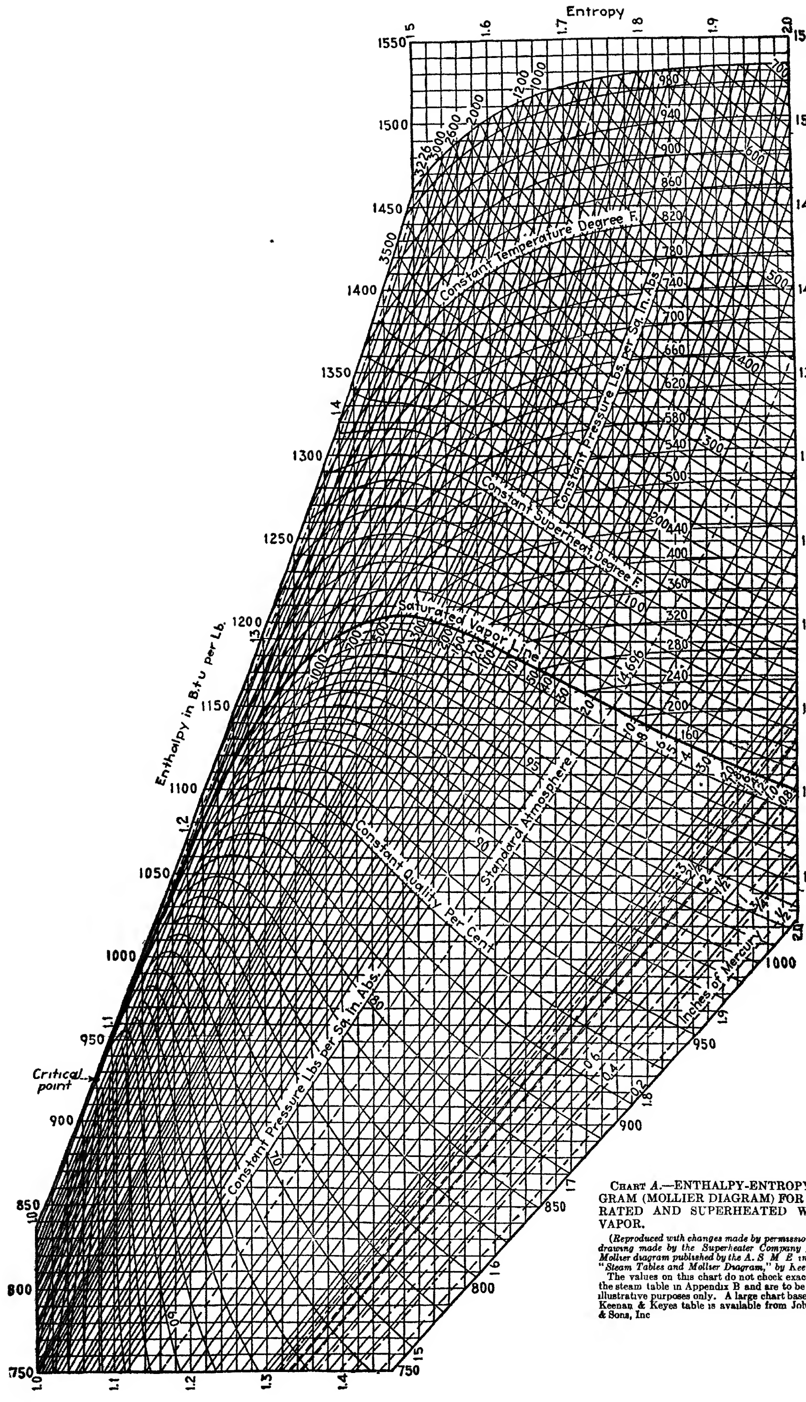


CHART *A*.—ENTHALPY-ENTROPY DIAGRAM (MOLLIER DIAGRAM) FOR SATURATED AND SUPERHEATED WATER VAPOR.

(Reproduced with changes made by permission from a drawing made by the Superheater Company from the Mollier diagram published by the A. S. M. E. in 1930 in "Steam Tables and Mollier Diagram," by Keenan.)

The values on this chart do not check exactly with the steam table in Appendix B and are to be used for illustrative purposes only. A large chart based on the Keenan & Keyes table is available from John Wiley & Sons, Inc

APPENDIX D

Properties of Refrigerants

	CO_2	NH_3	CCl_2F_2	CH_3Cl	SO_2	CCl_3F	CH_2Cl_2	H_2O
B. p. at atm. pressure, °F	−108 4	−28.0	−21.7	−10.6	14.0	74 7	104.9	212 0
Gage pressure, lb per sq in.:								
At 5°F.	319.7	19.6	11.8	6 2	5.9*	23 8*	27 39	Solid
At 86°F	1,024.3	154 5	93 2	80 8	51 8	3 6	9 62*	28.56*
At 40°F	553 1	58 6	37 0	28 1	12.4	15.5*	23 07*	29 67*
At 100°F.	Above critical	197.2	117.0	102.3	69.8	9.0	2 67*	27.99*
Refrigerating effect:								
86°F. condensing and 5°F. evaporator temp	55.5	474.4	51.1	148.7	141.4	65.4	134.6	
100°F. condensing and 40°F. evaporator temp	......	467.8	51.5	150 8	138.5	67.9	136 0	1,008.9
Refrigerant circulated, cu ft per min per ton:								
86-5°F. conditions	0.960	3 44	5.815	6.10	9 08	37 0	74 0	Solid
100-40°F. conditions	Above critical	1.696	3 075	3.60	4 17	16 1	27 6	484 0

* Subatmospheric pressures, in inches of mercury.

ANSWERS TO PROBLEMS

Solutions by GERARD PESMAN

Instructor in Mechanical Engineering, Montana State College

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1. 12 deg.
2. (*a*) 0.05 in.
   (*b*) 0.435 in. from bottom
3. 24 deg.
4. 73.5 deg from upper
   87 7 deg from lower
5. (*a*) Exhaust opens at 122.9 deg and closes at 237.1 deg from upper dead center.
   (*b*) Intake opens at 129.1 deg and closes at 230.9 deg from upper dead center.
6. 115.
7. (*a*) 394.5 cu in. per revolution.
   (*b*) 411 cu ft per min.
   (*c*) 2.055 cu ft per hp.
8. 3.7 cu ft per hp.
9. 3.735 cu ft per hp.
10. 3.46 cu ft per hp.
11. 150.2 ft per sec.

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1. (*a*) 11.2 cu ft per lb.
   (*b*) 7.8 cu ft per lb
2. 3.15 cu ft per lb.
3. (*a*) 1.69 per cent H_2.
   29.6 per cent CO.
   0.68 per cent O_2.
   9.32 per cent CO_2.
   58.70 per cent N_2.
   (*b*) 15.2 cu ft per lb.
4. (*a*) 84.3 per cent by weight CH_4
   6.9 per cent C_2H_4.
   8.8 per cent CO_2
   (*b*) 20.52 cu ft per lb
5. (*a*) 10.24 per cent H_2.
   16.32 per cent CO.
   54.10 per cent CH_4.
   5.41 per cent C_2H_4.
   8.50 per cent CO_2.
   5.41 per cent N_2.
   (*b*) 34.7 cu ft per lb.
6. (*a*) 6.5 lb air per pound fuel.
   (*b*) 9.09 lb air per pound fuel.
7. (*a*) 15.2 lb per pound fuel.
   (*b*) 22,040 Btu per lb.
8. (*a*) 13.46 lb per lb fuel.
   (*b*) 19,101 Btu per lb.
9. (*a*) 14.35 lb per lb fuel.
   (*b*) 20,503 Btu per lb.
10. (*a*) 10.51 lb per lb fuel.
    (*b*) 14,340 Btu per lb.
11. (*a*) 1.29 lb per lb.
    (*b*) 1.055 cu ft per cu ft gas.
    (*c*) 150.7 Btu per cu ft.
12. (*a*) 15.62 lb per lb gas.
    (*b*) 9.45 cu ft per cu ft gas.
    (*c*) 21,580 Btu per lb.
13. (*a*) 14.14 lb per lb fuel.
    (*b*) 5.06 cu ft per cu ft fuel.
    (*c*) 21,145 Btu per lb.
    (*d*) 609 Btu per cu ft.
14. 85 per cent N_2.
    15 per cent CO_2.
15. 85 per cent N_2.
    15 per cent CO_2.
16. 88.3 per cent N_2.
    11.67 per cent CO_2.
17. 11.38 per cent CO_2.
    4.45 per cent O_2.
    84.25 per cent N_2.

18. 11.43 per cent CO_2.
7.15 per cent O_2.
81.40 per cent N_2.
19. (*a*) 15.59 lb per lb.
(*b*) 28.82 lb per lb.
20. (*a*) 10.59 lb per lb.
(*b*) 14.98 lb per lb.
21. (*a*) 34.9 lb per lb.
(*b*) 10.85 per cent H in fuel.
22. (*a*) 18.4 lb per lb.
(*b*) 21.05 per cent excess.
23. 2.98 lb per lb fuel CO_2.
0.725 lb per lb fuel O_2.
0.049 lb per lb fuel CO.
13.85 lb per lb fuel N_2.
1.40 lb per lb fuel H_2O.
24. 3.12 lb per lb fuel CO_2.
4.91 lb per lb fuel O_2.
26.75 lb per lb fuel N_2.
0.978 lb per lb fuel H_2O.
25. 2.71 lb per lb fuel CO_2.
0.1435 lb per lb fuel CO.
1.15 lb per lb fuel O_2.
11.5 lb per lb fuel N_2.
0.36 lb per lb fuel H_2O.
26. (*a*) 20.0 per cent excess.
(*b*) 2.93 lb per lb fuel CO_2.
0.747 lb per lb fuel O_2.
14.82 lb per lb fuel N_2.
1.99 lb per lb fuel H_2O.

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1. 1,248,000 hp or 932,000 kw.
2. 2.04 hp.
3. (*a*) 237 hp
(*b*) 0.001525.
4. 38.2 hp.
5. (*a*) 0.000424.
(*b*) 37.0 hp.
6. 5.25 ft or 63 in.
7. 3,380 bhp.
8. 1.847 bhp.
9. 28,350 hp.
10. 22.4 hp.
11. 662 ihp.
12. 136.7 hp.
13. 502 hp.
14. 3,080 ihp.
15. 19.15 ihp.
16. 83.2 per cent.
17. 82.5 per cent.
18. 161.5 lb net brake load.
19. (*a*) 123.2 ihp.
(*b*) 87 bhp.
(*c*) 70.6 per cent.
20. 62.6 ihp.
21. 186.5 ihp.
22. 89.3 ihp.
23. (*a*) 237 bhp.
(*b*) 276.5 ihp.
(*c*) 85.7 per cent.
24. 24.27 friction hp.

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1. 237,000 Btu.
0.2015 hp.
2. 37.15 hp.
3. 163.9 lb per sq in.
4. (*a*) 0.932 lb per sq in.
(*b*) 28.1 in. Hg.
5. (*a*) 102.4 ft.
(*b*) 44.335 lb per sq in.
(*c*) 92.4 ft.
6. (*a*) 333.5 hp.
(*b*) 1,026,000 gal per hr.
7. 6,800,000 Btu per hr.
8. 2,260.
9. (*a*) 12.35 lb.
10. 15.58 lb.
11. 238,200,000 cu ft.
12. (*a*) 7,580 cu ft.
(*b*) 576 lb.
13. 1.048.
14. 1.013.
15. (*a*) 1.228 cu ft.
(*b*) 1.04.
(*c*) 139°F.
(*d*) 85,400 ft-lb.
(*e*) 98.7 Btu abstracted.
16. (*a*) $V_1 = 13.75$ cu ft.
$V_2 = 1.846$ cu ft.
(*b*) 1.036.

(c) 59,300 ft-lb.
(d) 69.6 Btu.

17. 860°R. and 77.88 lb per sq in.

18. (a) 451°R.
(b) 3.22 cu ft.
(c) 16.69 cu ft.

19. 9,920 ft-lb.

20. (a) 294°R.
(b) 4.17 cu ft.
(c) 2,320 ft-lb.
(d) 298 Btu.

21. (a) $P_1 = 12.3$, $T_1 = 60$°F., $V_1 = 15.67$ cu ft.
$P_2 = 100$, $T_2 = 60$°F., $V_2 = 1.927$ cu ft.
$P_3 = 100$, $T_3 = -325$°F., $V_3 = 0.5$ cu ft.
$P_4 = 12.3$, $T_4 = -363$°F., $V_4 = 2.925$ cu ft.
(b) $Wk_{1-2} = -57{,}400$ ft-lb.
$Wk_{2-3} = -20{,}550$ ft-lb.
$Wk_{3-4} = +18{,}000$ ft-lb.
$Wk_{4-1} = +15{,}680$ ft-lb.
(c) Net work $= -44{,}270$ ft-lb.

22. (a) $P_1 = 14.7$, $T_1 = 62$°F., $V_1 = 13.15$ cu ft.
$P_2 = 164.7$, $T_2 = 476$°F., $V_2 = 2.11$ cu ft
$P_3 = 26.15$, $T_3 = 476$°F., $V_3 = 13.15$ cu ft
(b) $Wk_{1-2} = 69{,}700$ ft-lb. Heat added = 00
$Wk_{2-3} = +91{,}200$ ft-lb. Heat added = +117.2 Btu
$Wk_{3-1} = 0000$. Heat added = −71.5 Btu.
(c) Net work $= +21{,}500$ ft-lb.

23. (a) $P_1 = 14.7$, $T_1 = 32$°F., $V_1 = 12.39$ cu ft.
$P_2 = 100$, $T_2 = 217$°F., $V_2 = 2.51$ cu ft.
$P_3 = 14.7$, $T_3 = -360.5$°F., $V_3 = 2.51$ cu ft
(b) Heat added 1 − 2 = 00
Heat added 2 − 3 = −99.6 Btu.
Heat added 3 − 1 = +94.7 Btu.
(c) $Wk_{1-2} = -49{,}650$ ft-lb.
$Wk_{2-3} = 00000$.
$Wk_{3-1} = +20{,}920$ ft-lb.
Net work = − 28,730 ft-lb.

24. (a) $P_1 = 14$, $T_1 = 60$°F., $V_1 = 13.75$ cu ft.
$P_2 = 133.7$, $T_2 = 532$°F., $V_2 = 2.75$ cu ft.
$P_3 = 914$, $T_3 = 6322$°F., $V_3 = 2.75$ cu ft.
$P_4 = 95.7$, $T_4 = 3090$°F., $V_4 = 13.75$ cu ft.
(b) Net work = 566,100 ft-lb.

25. (a) $P_1 = 13.5$, $T_1 = 100$°F., $V_1 = 15.37$ cu ft.
$P_2 = 602$, $T_2 = 1203$°F., $V_2 = 1.025$ cu ft.
$P_3 = 602$, $T_3 = 5370$°F., $V_3 = 3.44$ cu ft.
$P_4 = 74.2$, $T_4 = 2585$°F., $V_4 = 15.37$ cu ft.
(b) Net work = 447,000 ft-lb.

26. 0.373 unit.

27. (a) 0.071 unit.
(b) 0.0508 unit.

28. 0.0647, 0.0468, 0.0369, 0.0304, 0.0259, 0.0225.

29. Entropy at 1 = 2. 3 = 4. +0.3184 from 2 to 3. −0.3184 from 4 to 1.

30. 0.3135 unit total change.
0.000, 0.0427, 0.0385, 0.0359, 0.0349, 0.0346, 0.0238, 0.0329, 0.0311, 0.0335, units change in 20° steps.

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1. 99.3 hp.
2. 73.0 hp.
3. 2,970 cu in. volume. Bore and stroke vary.
4. 108.7 hp.
5. 562 hp.

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1. (*a*) 105.4 cu ft per hp-hr.
   (*b*) 11,080 Btu per hp-hr.
   (*c*) 22.95 per cent.
2. (*a*) 0.443 lb per hp-hr.
   (*b*) 8,425 Btu per hp-hr.
   (*c*) 30.2 per cent.
3. (*a*) 19.39 cu ft per hp-hr.
   (*b*) 9,600 Btu per hp-hr.
   (*c*) 26.5 per cent.
4. (*a*) 0.440 lb per hp-hr.
   (*b*) 8,800 Btu per hp-hr.
   (*c*) 28.9 per cent.
5. (*a*) 12,000 Btu per hp-hr.
   (*b*) 21.2 per cent.
6. (*a*) 71.1 per cent mechanical efficiency.
   (*b*) 0.461 lb per bhp-hr.
   0.3275 lb per ihp-hr.
   (*c*) 8,530 Btu per bhp-hr.
   6,060 Btu per ihp-hr.
   (*d*) 29.85 per cent brake thermal efficiency.
   42.00 per cent indicated thermal efficiency.
7. (*a*) 0.00167.
   (*b*) 8.76 ft.
   (*c*) 0.422 lb per bhp-hr.
   (*d*) 8,270 Btu per bhp-hr.
   (*e*) 30.75 per cent.

8. Run number	1	2	3	5
(*a*) Shaft hp	492.5	253	370	139
(*b*) Fuel economy, lb	0.417	0.458	0.417	0.598
(*c*) Heat economy, Btu	8,140	8,940	8,140	11,670
(*d*) Thermal efficiency, %	31.25	28.5	31.25	21.85

9. Run number	1	2	3	4	5
(*a*) Brake hp	124.5	100.0	75.3	50.2	25.1
(*b*) Ihp	137.5	111.9	86.5	60.2	34.9
(*c*) Mechanical efficiency, %	90.8	89.5	87.3	83.6	71.8
(*d*) Ihp. fuel economy, lb	0.374	0.341	0.334	0.3315	0.373
(*e*) Ihp. heat economy, Btu	7,340	6,698	6,560	6,505	7,325
(*f*) Thermal efficiency, %	34.7	38	38.8	39.1	34.75

10. Run number	1	2	3	4	5
(*a*) Fuel economy, lb	0.735	0.714	0.663	0.652	0.668
(*b*) Heat economy, Btu	14,330	13,940	12,930	12,710	13,035
(*c*) Thermal efficiency, %	17.75	18.25	19.67	20.02	19.52

11. Run number	1	2	3	4	5
(*a*) Brake hp	599	544	409	275.5	140.1
(*b*) Fuel economy, lb	0.4225	0.422	0.434	0.474	0.627
(*c*) Heat economy, Btu	8,280	8,270	8,520	9,300	12,300
(*d*) Thermal efficiency, %	30.75	30.77	29.85	27.37	20.70
(*e*) Jacket loss:					
Btu per hr	1,550,000	1,407,000	1,135,000	912,000	715,000
% of supplied	31.2	31.25	32.55	35.6	41.5

12. Run number	1	2	3	4
Exhaust loss, Btu per lb fuel	1,617	2,150	2,806	3,155
Moisture loss, Btu per lb fuel	1,830	1,915	2,025	2,080

13. Run number	1	2	3	4	5
(*a*) Fuel economy, lb	0.396	0.382	0.374	0.416	0.538
(*b*) Heat economy, Btu	7,780	7,500	7,350	8,180	10,580
(*c*) Brake thermal efficiency, %	32.7	33.9	34.6	31.1	24.05
(*d*) Heat to water:					
Btu	235,500	211,000	168,500	131,100	100,100
%	24.95	25.35	26.9	28.25	33.5

14.	Btu	Per cent
(*a*) Cylinder work	4,260,000	28
(*b*) Jacket loss	3,980,000	26.1
(*c*) Exhaust-gas loss	5,670,000	37.25
(*d*) Friction, etc	1,310,000	8.61

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1. (*b*) The water is in the vapor state.
2. (*a*) 1.372 Btu per degree rise.
   (*b*) Infinity.
3. (*a*) Mixture of water and vapor.
   (*b*) Superheated steam.
4. 1.484 units.
5. (*a*) 203.14°F.
   (*b*) 1,146.85 Btu per lb.
   (*c*) 1.7706.
6. Checks within 2 per cent.
7. (*a*) 97.7 per cent quality, wet steam.
   (*b*) Superheated 101.57°F.
8. (*a*) 0.658 lb
   (*b*) 827 Btu.
9. Steam, quality 53 per cent
10. 12 15 lb steam.
11. 19.27 lb water and 1.73 lb ice
12. 0.645 lb vapor remain.
13. 209.56°F.
14. 323.88°F.
15. 99.4 per cent.
16. (*a*) 303.5°F.
    (*b*) 0.3 per cent too much moisture.
17. 97.2 per cent.
18. 93.7 per cent.
19. (*a*) 91.9 per cent.
    (*b*) 0.0797 in.
20. (*a*) 93.6 per cent quality.
    (*b*) 97.3 per cent quality.
21. 30.7°F superheat.
22. (*a*) 99.5 per cent.
    (*b*) 95.4 per cent.
    (*c*) 94.9 per cent.
23. 7,030 lb per hour.
24. 9,590 lb per hour.
25. 3.855 lb per second.

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1. 239 hp.
2. 64, 87.5, 111.2, 134.7, 158.3 hp.
3. 131.9, 128.75, 125.5, 122.3 hp.
4. (*a*) 22.67 by 36.25 in.
   (*b*) 20.75 by 33.2 in.
5. 12.72 by 20.4 in. Stroke 1.6 times bore.
6. (*a*) 3.42.
   (*b*) 280.4 cu in.
7. (*a*) 548.2 cu in.
   (*b*) 3.54.
   (*c*) 0.0954 lb condensate.
8. 87.8 per cent quality.
9. 83.8 per cent quality.
10. 88.9 per cent quality.
11. 97.5 per cent quality.
12. 81.8 per cent quality.
13. 17.25 per cent.
14. 32.35 per cent.
15. 38.9 per cent.
16. (*a*) 10.22 per cent.
    (*b*) 25.3 per cent.
    (*c*) 40.5 per cent.
17. (*a*) 6.46 per cent.
    (*b*) 14.76 per cent.
    (*c*) 0.4375 or 43.75 per cent.
18. (*a*) 16.45 per cent.
    (*b*) 26.72 per cent.
    (*c*) 0.616 or 61.6 per cent.
19. (*a*) 51.85 hp.
    (*b*) 43.8 bhp.
    (*c*) 84.5 per cent.
    (*d*) 97 per cent.
    (*e*) 3.01 per cent.
    (*f*) 15.22 per cent.
    (*g*) 19.75 per cent.
20. (*a*) 495.5 ihp.
    (*b*) 457 bhp.
    (*c*) 92.1 per cent.
    (*d*) 99.2 per cent quality.
    (*e*) 8.2 per cent.
    (*f*) 26.3 per cent.
    (*g*) 31.3 per cent.
21. $1,974 loss due to failure to meet guarantee.
22. $3.69 per hp per year.

23. 75.2 per cent.
24. (*a*) 154,000,000.
(*b*) 11.6 lb.
(*c*) 19.77 per cent
(*d*) 26 per cent.
25. (*a*) 129,600,000.
(*b*) 16.67 per cent.
(*c*) 15.22 per cent.
(*d*) 26.75 per cent.
(*e*) 62.3 per cent or 0.623.
27. (*a*) 9.62, 9.84, 9.12 per cent.
(*b*) 19.2 per cent.

28. Load	30	60	90	120
(*a*) At 140 lb	11.55%	12.87	12.93	12.6
At 150 lb	11.65	13.0	13.07	12.8

(*b*) At 140 lb, 17.37%. At 150 lb, 18.01%.

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1. 89.9 ihp per cylinder end.
121.2 ihp per cylinder end.
144.3 ihp per cylinder end.
162.2 ihp per cylinder end.
2. 55.8 ihp at 80.
76.2 ihp at 100.
96.6 ihp at 120.
117.0 ihp at 140.
3. 125.5 ihp at 4.
115.2 ihp at 10.
101.6 ihp at 16.
89.7 ihp at 25.
4. 10.92 per cent increase.
5. (*a*) 19.35 lb per ihp-hr.
(*b*) 19,320 Btu ihp-hr.
(*c*) 16,320 Btu per ihp-hr.
6. Compound engine uses least heat per indicated horsepower-hour.
7. 16.7 per hp-hr.
8. (*a*) 4 Btu.
(*b*) 23.3 per cent.
(*c*) 23.0 per cent.
9. (*a*) 0.0352 lb.
(*b*) 12° superheat.
10. 290 per cylinder end.
11. (*a*) 106.5 per cylinder end, or 213 total.
(*b*) 229.7 total indicated horsepower.
12. (*a*) 0.517.
(*b*) 11.35.
13. (*a*) 760 ihp total.
(*b*) 790 ihp total.
(*c*) 0.868.
(*d*) 0.881.
(*e*) 1,550 ihp total.
(*f*) 12.82 lb per ihp-hr.
(*g*) 14,200 Btu per hr.
(*h*) 17.95 per cent.
(*i*) 21.86 per cent.
(*j*) 0.698.
14. (*a*) 1.82.
(*b*) 2.41.
(*c*) 8.55.

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1. Graphical solution.
2. Graphical solution.
3. (*a*) 6,000 lb. (*b*) 45.4 hp.
4. 100,500 ft per min, or 1,675 ft per sec.
5. (*a*) 0.162 of stroke.
(*b*) 0.222 of stroke.
6. 5.86 sq in.
7. 5,170 lb.
8. (*a*) Counterclockwise.
(*b*) Clockwise.
9. (*a*) 6.8 hp.
(*b*) 0.0159.

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1. 122 ft per sec.
2. 2,734 ft per sec.
3. 1,760 ft per sec.
4. (*a*) 2,122 ft per sec.
   (*b*) 123° superheat.
5. (*a*) 1,700 ft per sec.
   (*b*) 93.5 per cent quality.
6. (*a*) 2,835 ft per sec.
   (*b*) 435°F., or 148° superheat.
   (*c*) 7.84 per cent.
7. (*a*) 5.5 per cent moisture.
   (*b*) 2,880 ft per sec.
   (*c*) 2,783 ft per sec.
8. (*a*) 33.8°.
   (*b*) 492.7 ft per sec.
   (*c*) 1,315 ft per sec. entering. 115.5 ft per sec exhaust.
   (*d*) 48.53 hp.
   (*e*) 87.7 per cent.
9. (*a*) 25.8 deg first stage, 37.6 deg second stage, assuming blade velocity one-fourth of jet velocity.
   (*b*) 1,630 ft per sec, first; 992 ft per sec, second.
   (*c*) 2,615 entering, 1,322 leaving, first; 1,292 entering, zero leaving, second.
   (*d*) 191.2 hp.
   (*e*) 87.25 per cent.
10. (*a*) 140, 95, 62.5, 40, 25, 15 lb per sq in. abstract.
    (*b*) 5.22 ft diameter of blade circle.
11. (*a*) 96.5, 93.4, 90.8 per cent.
    (*b*) 1,656 ft per sec.

12. Run number	1	2	3	4	5
(*a*) Brake hp	1,037	1,385	1,683	2,205	2,578
(*b*) Steam rate	25.9	24.82	24.5	25.3	23.47
(*c*) Thermal efficiency, %	8.9	9.37	9.58	10.16	10.17

13. Load, kw	625	1,250	1,875	2,500	3,125
(*a*) Thermal efficiency, %	16.63	19.85	21.35	22.05	22.05
(*b*) Rankine ratio	0.528	0.610	0.656	0.678	0.678
(*c*) Total steam per hr, lb	10,570	17,680	24,650	31,850	39,850

14. Load, kw	2,000	1,500	1,000
(*a*) Thermal efficiency, %	18.97	18.36	16.75
(*b*) Rankine ratio	0.624	0.603	0.551
(*c*) Total steam per hr	30,200	23,400	17,100

15. (*a*) 27.4 per cent thermal eff.
    (*b*) 36.2 per cent Rankine eff.

16. Load, kw	43,750	35,000	31,000	25,000	15,000
(*a*) Thermal efficiency, %	25.35	26.1	27.06	26.5	24.9
(*b*) Rankine ratio	0.708	0.729	0.755	0.74	0.695
(*c*) Total steam lb per hr	459,000	357,000	305,500	251,300	160,500

17. Load, kw		10,000	15,000	20,000	25,000
(*a*) Guaranteed thermal eff., %		22.9	24.1	24.95	24.7
Load, kw	5,467	10,495	15,565	20,620	25,393
(*b*) Btu per lb steam	1,239.7	1,244.6	1,254.3	1,276.8	1,246.6
(*c*) Steam lb per kw-hr	14.43	11.65	10.9	10.63	10.72
(*d*) Actual thermal efficiency, %	19.05	23.7	24.9	25.1	25.5
(*e*) Shaft hp	8,120	15,150	22,000	28,800	35,250
(*f*) Steam lb per hp-hr	9.72	8.06	7.68	7.63	7.72

18. (*a*) 34.45 per cent.

(*b*) Load, kw	40,000	50,000	60,000
Over-all thermal efficiency, %	25.05	25.52	25.05
Shaft thermal efficiency, %	25.8	27.05	26.5

19. (*a*) Load, kw	2,000	1,500	1,000
Guaranteed thermal efficiency, %	24.5	25.45	26.85
(*b*) Efficiency of ideal cycle, %	30.5	30.5	30.5

(This is giving the cycle credit for the steam that is bled off.)

20. (*a*) 5.4 per cent thermal efficiency.
(*b*) 21.35 Rankine efficiency.
(*c*) 0.253.

21. (*a*) 25.5 per cent.
(*b*) 0.726.

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1. (*a*) 20.1 lb per lb steam.
(*b*) 11,200 gal per hr.

2 (*a*) 99 per cent.
(*b*) 5,090 gal per hr.

3. (*a*) 1,055.1 Btu per lb.
(*b*) 17.16 lb per lb steam.

4. (*a*) 91.5 per cent.
(*b*) 33.1 lb per lb steam.

5. (*a*) 95.5 per cent.
(*b*) 24.5 lb per lb steam.

6. (*a*) 23.9° superheat.
(*b*) 32.9 lb per lb steam.

7. (*a*) 1,025 Btu per lb.
   (*b*) 25,200 lb per hr.
8. (*a*) 981 Btu per lb.
   (*b*) 1,955,000 gal per hr.
9. (*a*) 30.1°.
   (*b*) 30.8°.
10. 1,298 sq ft.
11. 34,500 sq ft. with 50° cooling water.
12. (*a*) 14.55°.
   16.0°.
   (*b*) 5,425 tubes.
   (*c*) 4.45 ft per sec.
   (*d*) 407.5 Btu per hr per sq ft per deg logarithmic temperature difference.
13. Graphical solution.
14. (*a*) 24.85°.
   (*b*) 230 Btu per hr per sq ft per deg logarithmic mean.
   (*c*) 999 Btu per pound.

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1. 0.0301 lb.
2. (*a*) 14,180 Btu per lb dry.
   (*b*) 0.00091 lb.
3. (*a*) C, 67.40 per cent.
   H, 6.54.
   O, 23.05.
   N, 1.21.
   S, 4.37.
   (*b*) 12,470 Btu.
   (*c*) 0.157 lb.
   (*d*) 10,571 Btu.
4. (*a*) 12,110 Btu.
   (*b*) 14,260 Btu.
5. (*a*) C, 81.70 per cent.
   H, 2.61.
   O, 6.08.
   N, 0.80.
   S, 0.48.
   (*b*) 13,240 Btu.
   (*c*) 13,046 Btu.
6. (*a*) 11,040 Btu.
   (*b*) 12,560 Btu.
   (*c*) 0.5625 lb.
7. 0.609 in.
8. 3.69 in.
9. (*a*) 1.849 in. at 0°.
   1.674 in. at 22.5°.
   1.516 in. at 45.0°.
   1.369 in. at 67.5°.
   1.238 in. at 90.0°.
   (*b*) 1.504 in. at 29.75-in. barometer.
   1.516 in. at 30.00-in. barometer.
   1.530 in. at 30.25-in. barometer.
10. 0.73 in.
   0.75 in.
   0.788 in.
   0.827 in.

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1. 6,300 lb.
2. 4.5 in.
3. 27,200.
4. 8.5×10^{-6} O_2 cu. ft.
   15.3×10^{-6} N_2 cu. ft.
5. 30.6×10^{-6} to 4.6×10^{-6} lb per lb.
6. 11.07×10^{-6} cu. ft. O_2.
   20.15×10^{-6} cu. ft. N_2.
7. Assuming 30 lb per hp-hr 1.36 lb.

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1. 1.017 Btu per lb.
2. 1120 Btu.
3. 314.
4. 274.
5. 348.
6. 10,440.
7. 130.7 per cent rating.
8. (*a*) 8.09 lb per lb coal wet.
   (*b*) 9.01 lb per lb.
   (*c*) 9.53 lb per lb.

9. (*a*) 937.5 lb per hr
(*b*) 253.2.
10. 145 per cent.
11. (*a*) 22.5 cts. per 1,000 lb steam.
(*b*) 19.2 cts. per 1,000,000 Btu
12. (*a*) 8 lb per lb coal.
(*b*) 9.1 lb equivalent
(*c*) 73.5 per cent.
(*d*) 330.
13. (*a*) No.
(*b*) No.
14. (*a*) 8.65 lb per lb coal
(*b*) 4,960
15. (*a*) 626.
(*b*) 18,000 lb.
16. 59.9 per cent
17. (*a*) 11.97.
(*b*) 9.02.
(*c*) 71 1 per cent.
(*d*) 152,101 per cent of rating.
18. (*a*) 11.01 lb per lb coal equivalent.
(*b*) 89.0 per cent.
(*c*) 779.
(*d*) 153.3 per cent.
(*e*) $0.2214 per 1,000 lb steam.
(*f*) $0.1756 per 1,000,000 Btu.
19. (*a*) 4,630 Btu per lb fuel.
(*b*) 568 Btu per lb fuel.
20. (*a*) 1,403 Btu per lb coal, or 10.55 per cent.
(*b*) 543 Btu per lb coal, or 4.075 per cent.
(*c*) 69.5 Btu per lb coal, or 0.52 per cent.
(*d*) 85.8 Btu per lb coal, or 0.645 per cent.
21. (*a*) 72 3 Btu, or 0.55 per cent
(*b*) 280 Btu, or 2.13 per cent
22. 343 Btu
23. (*a*) 10,220 Btu, or 76.8 per cent to boiler
1,730 Btu to flue gas or 13.00 per cent.
72 Btu to moisture, or 0.54 per cent.
565 Btu, or 4.25 per cent to moisture.
94 Btu, or 0.71 per cent loss to carbon in refuse
59 Btu, or 0.44 per cent loss to CO in flue gas
560 Btu, or 4.22 per cent unaccounted for
(*b*) 836 Bhp.
24. (*a*) 12,365 Btu, or 83.2 per cent to the boiler.
1,527 Btu, or 10.27 per cent to the dry flue gas.
30 Btu, or 0.20 per cent loss to moisture in the fuel.
504 Btu, or 3.39 per cent loss to burning of hydrogen.
95 Btu, or 0.64 per cent loss to carbon in the refuse.
128 Btu, or 0.86 per cent loss to CO in the flue gas.
222 Btu, or 1.49 per cent loss to unaccounted-for places.
(*b*) 721.
25. (*a*) 1,131 Btu per lb.
(*b*) 8,910 equivalent pounds per hour.
(*c*) 258.
(*d*) 72.6 per cent.
(*e*) 35.9 per cent.
(*f*) 102 Btu per lb.
26. (*a*) 22,000; 32,950; 43,950; 54,900; 65,900 lb.
(*b*) 740; 1,110; 1,480; 1,850; 2,220 boiler hp.
27. (*a*) 66.5 per cent.
(*b*) 2,930.
(*c*) $115,800.

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1. 42.4 tons.
2. 2.875
4. (*a*) 112,900 Btu per hr.
(*b*) 132 Btu per lb.

(*c*) 98 Btu per lb.
(*d*) 18.5 per cent, 12.5 per cent.

5. (*a*) 17.38 per cent.
(*b*) 125 Btu per hr.
(*c*) 3.915.

6. (*a*) $-0.57°F$.
(*b*) 5.18.
(*c*) 7.98 tons.
(*d*) 7.16 hp.
(*e*) 6.67.
(*f*) 77.8 per cent.

7. E_{mr} = 1152 Btu per min.

8. (*a*) 245 lb per hr.
(*b*) 11.75 hp.

9. (*a*) 5.89°F.
(*b*) 4.71.
(*c*) 19.65 tons.
19.7 hp.
(*d*) 0.444 cu ft.

10. (*a*) 5.89°F.
(*b*) 5.0.
(*c*) 17.01 tons.
16.08 hp.
(*d*) 3.94 cu ft.

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