DIESEL ENGINE DESIGN

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PREFACE

The methods of calculation adopted for these chapters are those which the author has found to be understood and retained most readily by the men working with him and remembered most easily for extemporaneous work when no library is available.

No derivations for formulas in ordinary use are given except where it is known that existing texts are written more for mathematicians than for men whose natural bent is mechanical rather than scientific.

A serious attempt was made to incorporate some of the more academic instruction emanating from the National Advisory Committee for Aeronautics at Langley Field, Washington, D. C., and from Pennsylvania State College, State College, Pa. It was abandoned after much fruitless labor because the original papers require not only consideration, but also extensive and able discussion.

Naturally these papers should be acquired from the sources and diligently studied as well as like papers from the Verein Deutscher Ingenieure.

The long-threatened but not yet imminent fuel shortage is the basis of much recent research beginning with the fuel-testing methods devised by Boerlage and Broeze at Delft, Holland, for the Royal Dutch Shell and those of Pope and Murdock for the Waukesha Engine Company of Waukesha, Wis. These will be discussed briefly under "Combustion."

A narrowly defined diesel fuel specification can result only in higher prices for fuel. It should be remembered that engines as well as fuels are being graded when such tests as those at the Delft laboratories are undertaken.

The diesel engineer's objective is not so much to seek out the best fuels but rather so to develop his product that it will burn a rather broad range of those refinery products which are in least demand for other purposes.

To this end the studies on combustion-chamber gas and wall temperatures in this book are most valuable in connection with the inevitable developments to result from more general practise of fuel testing.

HAROLD F. SHEPHERD.

Sept. 20, 1934.

CONTENTS

DIESEL ENGINE DESIGN

CHAPTER I

HISTORICAL

Rudolf Diesel's narrative of four years of faithful endeavor resulting in the definition of the principles of design governing the air injection type of engine is contained in his last thin volume, "Die Entstehung des Dieselmotors." It is a brief history compared to that of the diesel with direct injection of fuel.

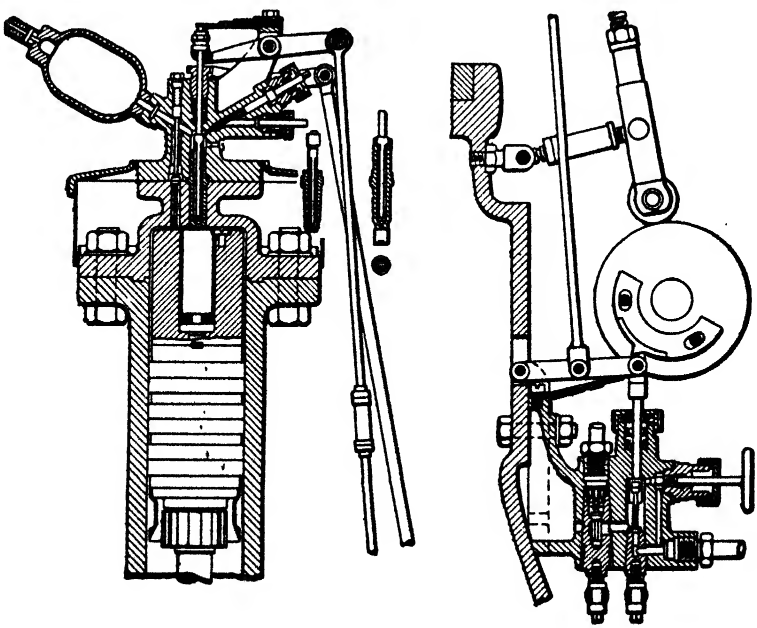


FIG. 1

In 1894 he wrote in his log concerning the engine shown in Fig. 1:

> The cylinder was removed and replaced by columns in order to make experiments in atomization of fuel in the open air, the engine being driven from a line shaft.
>
> Next, injection by means of the fuel pump direct, only an automatic valve in the nozzle, over the seat of which a conical distribution and atomization of the fuel shall take place. The operation is entirely unreliable. The degree of atomization observed under various pressures: at high pressure, good; at low pressure, uninterrupted jets without the least atomization.
>
> Further experiments, injecting directly with the pump and simultaneous opening of the mechanically operated spray valve needle. The pump plunger and spray valve needle are linked together and operated by the same cam. Injection very precise.
>
> Next the pump was set aside and it was sought to accomplish injection by mechanical operation of the needle, the fuel pipe being kept under constant pressure. After these trials in the open air followed tests in operation with direct injection out of the fuel pressure pipe; ignition excellent. Exhaust gases burning still as they leave the cylinder, the spray valve is most unreliable, injection uncontrollable, the system must be abandoned.

For many years after this declaration the diesel engine used air injection of fuel exclusively.

Herbert Akroyd Stuart had built an Otto-cycle engine using direct injection of fuel in 1890. His British patent 7146 of that year describes

> Means for preventing the premature or pre-ignition of an explosive charge of combustible vapor or gas and air when a permanent igniter (such as a continuous spark or a highly heated igniting chamber) is in communication with the interior of the cylinder, by first of all compressing the necessary quantity of air for the charge and then introducing into this quantity of compressed air the necessary supply of combustible liquid, vapor or gas to produce the explosive mixture.

Engines of this general type became known as "semi-diesels."

Fifteen years after Diesel, James McKechnie was well on the road to success with the Vickers engine, Fig. 2, using what has become known as the common-rail system in which a pump *D* forces fuel into the main or "rail" common to all nozzles. A relief valve discharges the excess above that required to maintain the desired pressure, and cams operate the fuel nozzles *S* at the timed intervals of injection.

This system, essentially that described by Diesel as "unreliable" and "uncontrollable," is especially adapted to variable-speed marine work, and being so very obvious in its way of

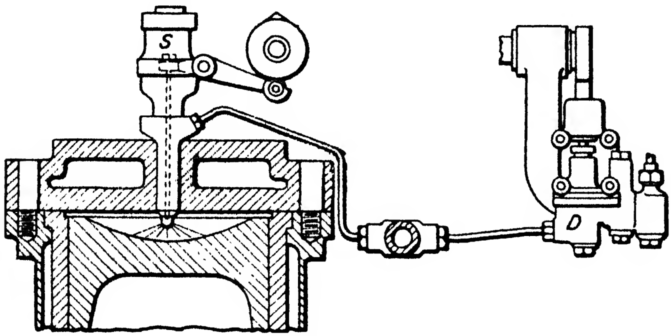


FIG. 2

functioning it has been much used on fishing boats, work boats and yachts although its popularity is waning in favor of commercial pumps and nozzles capable of operating reliably at much higher speeds.

Europe did not adopt the common-rail system, and there remained a desire world wide to use in cold starting diesels the simpler fuel pressure operated nozzle valves together with individual cam-operated fuel pumps which had been used so successfully on semi-diesel engines.

This was first accomplished by Knut Jonas Elias Hesselman in Sweden and soon after with much-simplified mechanism in this country. Subsequently several simple and very economical models were produced in Germany.

One of Hesselman's patent drawings is shown in Fig. 3.

What Hesselman did essentially was to study the hydraulics and kinetics of fuel systems with the result that he learned how to measure and deliver accurately small charges by means of individual pumps at pressures high enough to serve high-compression engines. The compressibility of the fuel contained in needlessly large clearances in pumps and nozzles was so great as to make many of the earlier solid injection devices inoperative at more than a few hundred pounds' pressure.

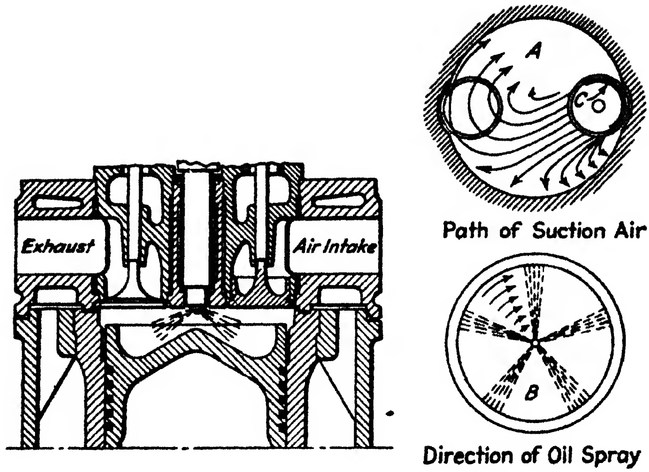

Exhaust
Air Intake
A
Path of Suction Air
B
Direction of Oil Spray

FIG. 3

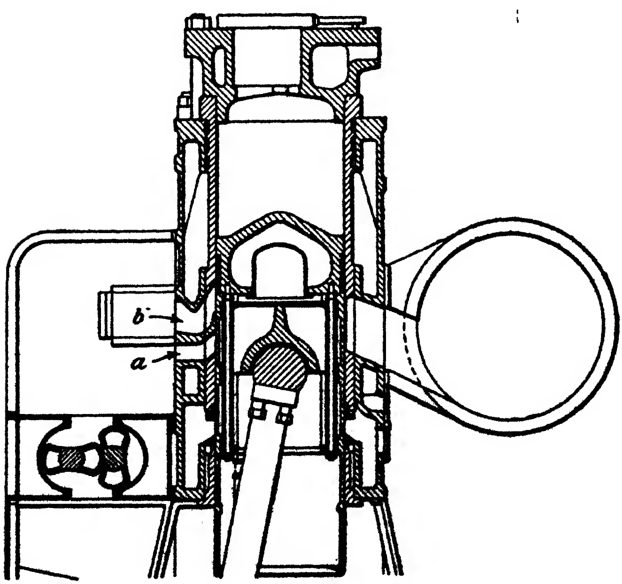

b
a

FIG. 4

Unfortunately the published accounts of Hesselman's work are somewhat confused by transcendental matter. His patent cases likewise fail to reveal his views.

In his original application filed in the U. S. Patent Office, July 21, 1922, Serial No. 578,251 Div. 28, he states on page 3, Line 12,

> The piston is formed with a flange 4 in the extension of the piston wall to prevent the fuel injected from touching the comparatively cold cylinder wall which would otherwise result in cooling of the fuel.

In the allowed patent 1,555,204 Sept. 29, 1925, he withdraws this description and says:

> the speed of injection of the fuel is adjusted to correspond to the speed of evaporation of the fuel in order to cause the fuel to evaporate before reaching the cylinder wall.

The rim remains a part of all Hesselman piston designs although in recent large engines it appears as part of the cylinder head (Fig. 4). It had been used previously to his patent, on small-bore diesel engines using the Vickers system of injection to prevent over-penetration of fuel. That the fuel spray actually finds a target in most engines has been shown by a succession of photographs taken from various models and published in the *Zeitschrift* des Vereins Deutscher Ingenieure, one of which is reproduced in Fig. 5.

Thus in chronological order Herbert Akroyd Stuart invented the method and a means of injecting liquid fuel at firing time to avoid pre-ignition. His invention made possible the use of the less volatile hydrocarbons. Rudolf Diesel discovered that fuel injected by the Akroyd method could be fired by the heat of compression. He devised a way of injecting fuel into this dense atmosphere by means of compressed air, thus realizing the great thermodynamic advantage of high compression. James McKechnie succeeded in burning fuel economically by one of Diesel's abandoned methods. K. J. E. Hesselman perfected another method abandoned by Dr. Diesel.

These two latter accomplishments are the basis of all the larger modern diesel engine developments. They have greatly simplified design, manufacture and operation.

The McKechnie and Hesselman engines employ axial nozzles with five or more orifices to distribute the fuel. These drilled orifices become so fine for small engines that they are impractical in manufacture and use, and other forms of combustion

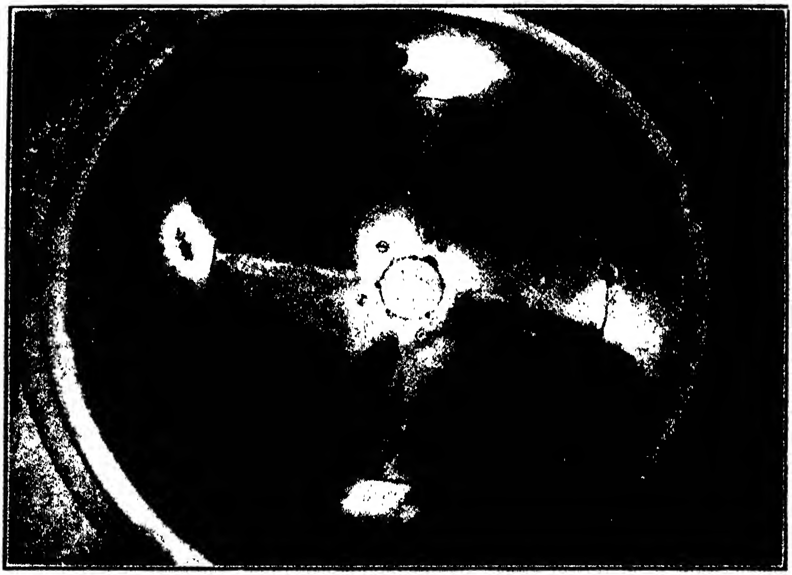


FIG. 5

chambers had to be developed for engines of less than 7-in. bore in order that single-orifice nozzles might be used.

As now built there are three distinct types of combustion chambers for small engines, i.e.:

1. The ante-chamber or pre-combustion chamber in which all the fuel is injected into a small cell in constant communication with the cylinder proper. This cell contains only a fraction of the compressed air charge at firing time, consequently only part of the fuel injected into it is burned there, but the resulting explosion drives fuel and gases into the main chamber where combustion is completed with the aid of a high degree of turbulence.

Diesel in his German patent D.R.P. 82,168 (1893) shows a pre-combustion chamber, and there are numerous earlier suggestions of the idea in the patent records. The patent situation in this field is reviewed in "Die Patente über Vorkammer, Diesel

Maschinen" by F. Ernst Bielefeld, privately published by the author at 175 Borstelmannsweg, Hamburg 26.

2. The after-chamber or air-cell engine in which a part of the air charge is compressed into a cell in constant communication with the cylinder proper. Fuel is injected into the main chamber, and inventors have claimed that the air stored in the cell is fed out during the expansion stroke and is used to complete combustion. This is poor thermodynamics and it is of course quite impossible to keep fuel out of the air cell. Fuel blown into the cell by displacement or explosion of the main charge explodes in the cell, and the returning products of combustion set up renewed turbulence with beneficial results. The patent situation in this art is described in "Die Luftspeicher-Diesel Maschine" by B. Klaften, published by Carl Heymanns Verlag, Berlin.

3. The turbulence chamber in which all the air charge is forced through a neck entering a chamber of nearly spherical form at a tangent. The air charge rotates before the fuel nozzle during injection. There are numerous suggestions of this idea in the patent records, some of which are described in "Die kompressorlose Diesel Maschine" by Ludwig Hausfelder, published by M. Krayn, Berlin.

The design of these various combustion chambers is discussed in Chapter III where they are illustrated.

CHAPTER II

COMBUSTION

The use of fuel in process and in power generation usually has preceded exact knowledge of combustion.

By way of illustration, Simon Sturtevant in 1611 patented the use of mineral coal for iron smelting.

More than 150 years after Sturtevant, Joseph Priestley wrote in scientific vein "On Different Kinds of Air." He prepared a gas from red oxide of mercury which he stated was not "common air" but "a substance in much greater perfection," "a supporter of combustion." At almost the same time Cavendish was writing of "Factitious Airs," "Inflammable Air" (hydrogen) and "Fixed Air"(carbon dioxide). Contemporaneously Watt was developing the steam engine (1760–70).

In 1735 Abram Darby constructed and used the first coke-charged blast furnace. Blast-furnace gas was used as early as 1811 for heating steel, but nearly 100 years had elapsed before Bunsen performed his great service to the steel industry that was to be so valuable to the internal-combustion engine craft.

In 1627 Francis Bacon investigated the "original concretion of bitumen" and observed that the"naphtha of Babylon attracts fire from afar." Yet the first chemical investigation of petroleum by Winterl followed 150 years later, and not until 1855 did Professor Silliman open the career of the petroleum technologist.

The chemistry of the blast furnace was applied to the gas producer. The chemistry of the petroleum technologist was applied to the liquid-fuel engine and the natural-gas engine. Today, in the oil engine industry we are pioneering ahead of true science — which is not to be wondered at since there is yet no absolute agreement among scientists as to the exact nature of combustion reactions in open fires.

THE SEMI-DIESEL OR HOT SURFACE ENGINE

This type of oil engine was conceived before the diesel and was a commercial success long before the diesel engine with solid injection arrived to supplant it.

It was built in a profusion of designs utilizing in different combinations every type of spray nozzle and fuel pump used today. Many schemes for creating turbulence and directing it were employed, and the hot surface area was varied from a totally uncooled combustion chamber to a mere hot spot in the wall of that vessel.

During this period of research and invention two facts stood out which apply with equal effect to solid injection diesel engine design:

1. When oil is sprayed against a red-hot target within the combustion chamber it burns smolderingly as though smothered, soot appears in the exhaust and coke deposits build up on the hot surface, power is reduced and the unit fuel consumption greatly increased.

2. When the degree of compression results in too great a temperature at the time of injection the same phenomena are observed.

These actions, inevitable under certain conditions, were attributed to isolation of the injected fuel from the air by the products of combustion and to cracking.

THE DELAY ANGLE

It was observed too that a slight delay in ignition after the introduction of the first fuel particles is essential to good combustion. This follows naturally as a result of correct hot surface and compression temperatures. This delay still is considered necessary to allow partial vaporization and diffusion of the fuel before spontaneous ignition.

Later researches have shown that too great a delay in effecting ignition is one factor contributing to detonation, and measures are taken to control the lag between introduction of fuel and spontaneous ignition.

PRACTICAL HOT SURFACE TEMPERATURES

The engine shown in Fig. 6 was devised to study the effect of hot surface temperatures on combustion.

The fuel target is the bottom of a pot which is filled with mercury and sealed under atmospheric pressure. The boiling point of mercury at sea level is 672°F, so the hot surface is held at that temperature constantly, regulation being effected by the condensation of mercury vapors rising to the jacketed cover and the return of the condensed vapors to the bath below. The condenser jacket over the mercury pot is separate from the

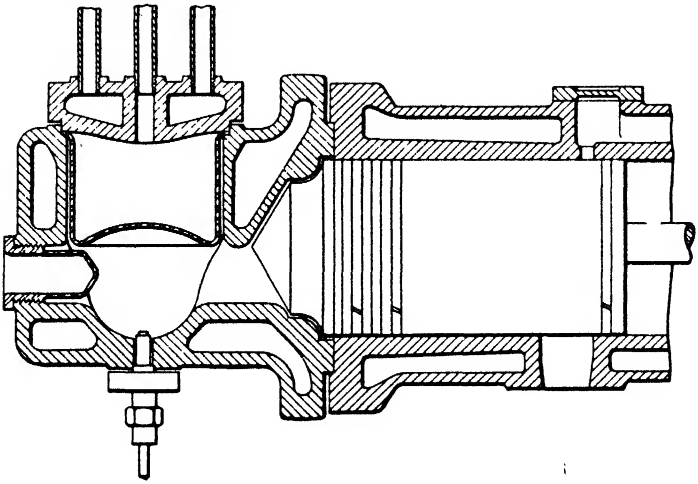


FIG. 6

head jacket. By bringing the two overflows to the same temperature no exchange of heat takes place between the flanges. The weight of the condenser jacket discharge water and its temperature rise give, directly, the amount of heat to be disposed of when different fuels are employed.

It was shown that very light fuels play some part in cooling the target as they take up their latent heat of vaporization.

The total difference for an 8-in. target in a range of fuels having average boiling points from 595°F to 740°F was about 1000 Btu per hr, which is not great. Other tests during which the vaporizer temperature was raised by applying air pressure over the mercury to raise its boiling point proved that for all of the many fuels tested the best target temperature is near 850°F. Combustion is good down to 680°F. Smoke and general bad performance appear above 900°F.

ESTIMATED COMPRESSION PRESSURE

If the fuel is prevented from striking the walls of the combustion chamber the heat stored in the compressed air charge may function in exactly the same way as that stored in the hot surface of the more primitive engine if the thermal capacity of the air charge suffices.

Assuming compression end temperature limits of 680° to 850°F with a mean compression exponent of 1.35 and an initial charge temperature of 60°F, we might expect good working within pressure ratio limits of 21 to 35. At 13.7 lb absolute initial pressure these values result in a minimum compression of 274 lb gage and a maximum compression pressure of 466 lb gage.

Since the mean of these two values 370 lb is not far removed from the compression found most suitable for many commercial engines the above assumptions seem to be well founded.

Very volatile petroleums may burn with an abnormal delay angle, and alcohols may refuse to ignite at all in the fully jacketed combustion chamber although they burn well enough in low-compression hot surface engines. This is due to the lesser thermal capacity of the air charge, which, however, suffices for initiating good combustion of all domestic crude oils, straight-run fuel oils, cracked burning oils and some residues.

EXCEPTIONAL COMPRESSION PRESSURES

Reasons for departing from this apparently rational compression stage multiply as the range of sizes, types and applications is broadened.

Very small engines (4-in. bore or less) seldom realize the mean exponent 1.35 during compression. It may fall as low as 1.20 as compression progresses. In such cases the compression pressure may be raised so that the temperatures desired are attained if possible.

It has been shown that, when the surface enveloping the combustion chamber is relatively very great, compression may be increased from 400 lb to 600 lb or more with little or no increase in temperature at all, the latter stages approaching isothermal on account of excessive cooling. In such cases compression may be increased for the sole purpose of limiting penetration of fuel.

In some services any degree of detonation is offensive. With some risk of smoke particularly with slight overloads or subnormal condition of engine the compression may be increased to the upper temperature limit or a trifle beyond to decrease ignition lag.

DEDUCTIONS REGARDING COMBUSTION

Unfortunately, complete vaporization of all fuels at the known practicable compression and hot surface temperatures is quite impossible. This is shown by the vaporization and critical data for some paraffin series petroleum fractions, Table I. Many fuel oils contain fractions boiling above 850°F and therefore having critical temperatures above 850°F, which is the observed practical limit of wall or compression temperature.

It is totally impossible to avoid cracking although it proceeds very slowly at the temperature prevailing at the beginning of injection.

The constant-volume phase or explosion, resulting in a rise of 150 to 200 per cent of the compression pressure, is always a necessary part of the cycle of the solid injection engine which will not burn fuel successfully in the constant-pressure cycle.

TABLE I

PARAFFINS OR METHANE SERIES FROM PENNSYLVANIA CRUDE

	Boiling point, °F, at 30 in. Hg	Critical temperature, °F	Critical pressure, atmospheres	Specific gravity	Heat of vaporization, Btu per lb
Pentane	86–100	390	24	0.626	
Hexane	142–156	450	22	0.680	(154.4)
Heptane	196–208	515	20	0.699	(142.9)
Octane	244–257	565	18	0.719	(140.0)
Nonane	277	640	16	0.741	(128.5)
Decane	316	680	15	0.757	(109.4)
Undecane	....	720	14		
Duodecane	360	760	13	0.765	
Tridecane	421	850	10.5	0.792	
Tetradecane	460	900	9		

It might be put forward then that the cycle consists of:

1. Compression to approximately 850°F. Fuel injected before dead center is partially vaporized without great cracking

and spontaneously ignited after momentary delay due to absorption of latent heat.

2. Ignition of the vaporized, cracked and partly diffused lighter fractions results in explosion, raising the temperature to 1600° or 2000°F, the blast creating intense turbulence.

This brings the remainder of the fuel to a temperature sufficient to accelerate greatly all reactions of combustion completing the consumption of fuel in the short time available.

EXCEPTIONS

It may be urged that solid injection engines could be built to avoid this dual combustion process by greatly increasing wall temperatures, initial charge temperature or compression.

This has been done, but the result has always been much the same. The hot cycle contributes to mechanical failure of pistons, valves and cylinders due to the abnormal heat flux.

The temperature peak of the normal engine endures but a short time and is productive of no real difficulties.

AIR INJECTION ENGINES

It is true that engines using air blast to atomize fuel do not exhibit the qualities described in this chapter. This fact goes far toward proving that direct or solid injection does not immediately and of itself prepare fuel for combustion.

ENGINES WITH PRE-COMBUSTION

These behave somewhat like air injection engines, but by no means ideally.

TURBULENCE

Early investigators sought to study combustion phenomena in pressure vessels and learned that the reactions were much retarded when initiated in the still atmosphere of the observation chamber. It was recognized that the induction of the charge into the engine cylinder sets up a state of turbulence persisting to the time of ignition. This turbulence, assisting in flame propagation, is partly responsible for the rapid union of the elements necessary to complete combustion in the time allowed by even moderate speed of rotation. Induction turbulence suffices for many large and slow-speed engines using fuel nozzle arrangements effecting good fuel distribution.

Forced and directed turbulence is now in use both for mixing and for controlling the rate of flame propagation. When nozzles are placed radially to the combustion space or at a tangent to some inner diameter, rotary turbulence of the air charge may be effected by shrouding one side of the inlet valve as in Fig. 3. Air enters through one side of the valve, only, causing it to follow a circular or helical path under the guidance of the cylinder wall. This rotation persists throughout compression and injection as has been demonstrated by clever and adequate experiments. Its intention is to distribute the fuel mist or vapor which has entered in rays.

Rotary turbulence is quite unnecessary when the axial nozzle is used; however, its improvement of combustion in engines using tangent nozzles is measurable.

In practise the shroud on the valve is rather a nuisance as carbon builds up on it and adjacent parts of the valve seat neither of which are washed by air during induction. The shroud also reduces the inlet area to a very undesirable extent.

The first use of the shrouded inlet valve seems to have been by Reichenbach, Austrian patent 38,003 Kl. 46 a, issued July 26, 1909.

Rotary scavenging of two-cycle engines has been effected by causing the scavenging ports to enter the cylinder at a tangent to a circle relatively small as compared to the bore. This is of value in scavenging and consequently in combustion, and is discussed in Chapter VIII.

High-speed engines operating at 1000 or more revolutions per minute require the use of forced turbulence to shorten the delay angle. If only induction turbulence is available the interval between the beginning of injection and ignition is so long that most of the fuel or all of it has been injected when ignition takes place. The consequence is simultaneous inflammation of a large part of the fuel charge resulting in a terrific rate of pressure rise and objectionable detonation. The three forms of combustion chambers described in Chapter I as suitable for small engines and discussed from the design angle in Chapter III (Figs. 18, 19 and 21) are the surviving forms of forced turbulence arrangements for eliminating detonation.

COMMERCIAL FUELS

Well-developed engines of all sizes burn any clean domestic crude oil. When using the heavier asphaltic oils such as the 14°Bé California crude the fuel consumption may be increased 10 per cent and the rating may fall correspondingly, but there should be no mechanical disabilities except those resulting from water or earths in the oil. Both these impurities may be eliminated by centrifuging, a proper precaution for all oils bought in bulk.

Natural distillates between 24° and 34°Bé are, almost without exception, suitable fuels. Above 36° or 40°Bé, viscosity is likely to be so low that the leakage of unpacked valves and plungers may be excessive or wear resulting from working with fuel having no lubricating properties may soon make it so.

Those products of the modern cracking plant known usually as furnace oils may burn with more or less objectionable detonation. They are recognized in the field by their surprisingly low gravity relative to their fluidity. They are free boiling in the comparatively low temperature ranges. Whereas a good 28°–32° natural fuel oil (topped crude) may yield only 60 per cent of distillate at 662°F (350°C) and leave 3 or 4 per cent of coke when distilled to dryness, cracked fuels leave little or no residue and usually distil off completely below 662°F, often at so low a temperature as 572°F (300°C). Their low ignitability is partly the result of free vaporization with extraction of unusual quantities of latent heat from the air compressed in the combustion space.

Unfortunately, the most desirable fuel oils are being requisitioned as the best stocks for cracking into gasoline of high antiknock quality. The refinery wastes are no longer topped fuels from the poorer crudes or cuts intermediate between gasoline, kerosene and the lubricants from the richer crudes but the dry straw-colored furnace oils and tarry residues. Attempts on the part of refiners to dispose of these by-products together by blending have yielded no satisfactory or economical results to the engine builder or user. Such blended fuels often are not solutions but rather unstable mechanical mixtures carrying excessive free carbon, asphalt of the least soluble or combustible sort and ash-forming impurities. They are troublesome to filter, harmful to pumps and nozzles and erratic in burning, causing rough running and excessive upkeep costs.

FUEL TESTING

Although it is well known that the ignition quality of a diesel fuel is dependent upon its chemical structure, as yet no standardized method of measuring it has been developed and generally accepted. At present it must be measured in actual engines. At least two methods have been proposed, however — one in this country by the Waukesha Motor Company in which use is made of a modified C.F.R. engine to determine the critical compression ratio at which the fuel will fire within three seconds after being injected into the cylinder when the engine is externally driven at 600 rpm, the intake air temperature being maintained at 100°F and the jacket temperature at 212°F. Another method has been proposed by Boerlage and Broeze, who measure the actual ignition delay in a large slow-speed solid injection engine at a compression pressure of 450 lb. This delay is compared with the delay obtained on mixtures of cetene and alphamethyl-napthalene, the percentage of cetene which gives the same delay as the fuel under test being called the "cetene number." This cetene number, therefore, is analogous to the "octane number" of gasoline, which is a measure of the ignition quality of the latter fuel.

Some of the test engine performance attributed to fuel properties by Boerlage and Broeze are well-known engine disabilities which have been eliminated by invention and design in other engines many years ago. Naturally, the engine builder's objective should be to design for use of the broadest range of fuels, perhaps from 25 to 75 cetene number.

The use of varying compression ratios in the test engine should not be taken as an indication that compression ratios for diesel engines should be adjusted by the operator to suit the fuel. This idea is quite impracticable and with proper control of surface temperature quite as unnecessary.

CHAPTER III

THE INFLUENCE OF THE COMBUSTION-CHAMBER WALLS

The open type of combustion chamber like Figs. 3 and 4 is used in most single-acting engines of more than 7- or 8-in. bore. As the fuel spray usually is directed toward the piston its face forms the only wall of the open combustion chamber having a serious influence on combustion. The surface of the cylinder head seems to exert no great influence. Experiments with heads shielded by centrally fixed metal plates in thermal contact with the jacketed surface but naturally hotter than it resulted in no measurable gain.

The section of the top of the piston is not entirely a matter of choice or fancy. The patented (Austrian patent 101,521, Aug. 3, 1923) hemispherical depression, Fig. 7, is possible only when the stroke is long compared to the bore. A hemispheri-

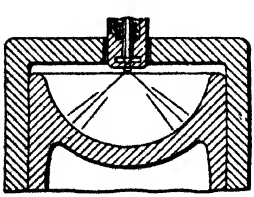


FIG. 7

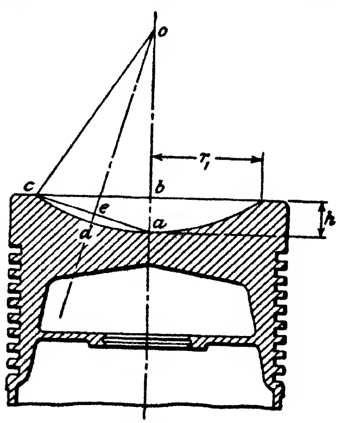


FIG. 8

cal cup contains a volume which in engines of ordinary ratio of stroke to bore exceeds the required compression space.

When the depression is the segment of a sphere, as it commonly is, Fig. 8, good results may be obtained with most stroke-bore ratios, but below 9-in. bore difficulties creep in on account of over-penetration of fuel. If the angle of fuel jet incidence is raised to prevent the fuel from skidding off the piston, over-richness of the central part of the charge causes loss of economy. For those smaller sizes, it is best to add a rim to the piston as shown in Fig. 3. In very large slow-speed designs, penetration is usually insufficient and the combustion-chamber radius may

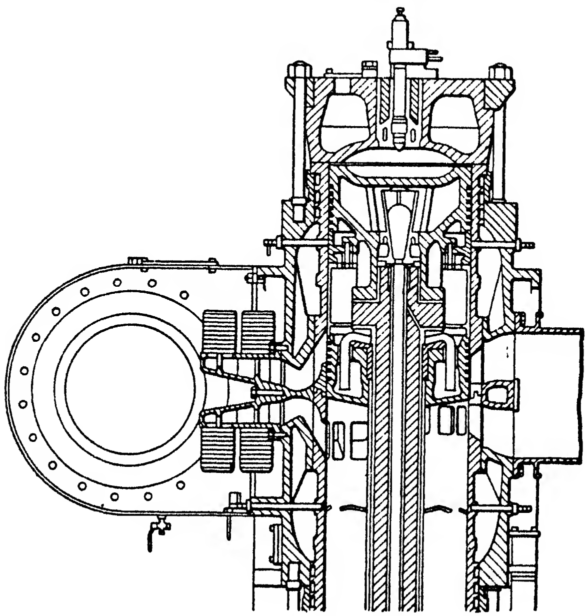


FIG. 9

be confined as in Fig. 9. For small engines of relatively short stroke good operation has been realized through deflection of the fuel back into the combustion chamber by a reflecting piston rim, as in Fig. 10. If this is done, operation will be less dependent on the impact angle of the jets, which ordinarily is a serious matter with very flat combustion chambers.

Flat piston crowns have been used with the central or axial fuel nozzle. To use this combination successfully fuel penetration must be limited by finer drilling with many holes in the tip. Seemingly, this should be the most perfect design. In practise it proves otherwise. Combustion is not improved and orifice stoppage with drilling smaller than 0.010- to 0.012-in. diameter is quite troublesome.

The particular part of the piston face or rim which serves as a target for the fuel must assume a suitable operating temperature. A "cold" piston allows white clouds of unburned oil vapor to pass out of the exhaust when the load is thrown on. A "hot" piston so accelerates ignition that the fuel burns smokily while yet in drop or spheroid form. Temperature control is accomplished by proportioning the metal thickness in the piston head and by regulating the number of piston rings. A thick piston with many rings conducts more heat to the jackets than a thin one with few rings and therefore runs cooler.

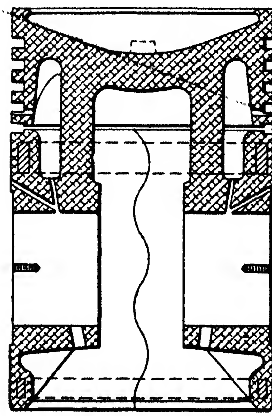


FIG. 10

According to experiments, the optimum piston face temperature is about 850°F. This is just below red heat in the daylight and causes metal to glow faintly in the dark. A more practicable value is 700°F as it results in very little reduction in economy and provides a margin of safety against overload, bad fuel and bad condition of the engine, all of which tend to overheat the piston. Fair full-load operation may be had with the piston face at 450°F, but light-load operation and idling will then be unstable.

Various methods have been used to determine the piston head temperature, even to continuous records made by thermocouples set into the face with leads carried out to indicating instruments through swinging links.

Often, more simple if less accurate means must be employed. Polished samples of the piston crown material may be subjected to electric furnace temperatures varying from 500°F to 1100°F, each sample being "soaked" four hours under very careful furnace control. Temper colors will result. "Soaking" is required because long subjection to heat results in deeper colors. These samples are compared with the colors assumed by the polished piston face after four hours of operation.

Unfortunately, cast iron takes its purest colors at low heat. At working heat they are rather indefinite. It is better to use as indicators tightly screwed plugs of super-ascoloy (Cr 17 to 30 per cent; Ni 7 to 10 per cent), an iron-nickel-chromium alloy. This metal colors well to 1000°F. The surface exposed to the heat should be polished as a roughly turned surface always absorbs more heat. Temper colors are described in Table II.

TABLE II
TEMPER COLORS

°F	Cast iron	Super-ascoloy
450		
500	Golden	
600	Bright blue	Silver faintly gilt
675	Purple and blue	Pale golden
700	Silvering on fading blue	
800	Bright slate overlaying silver	Deep gold
850	Bluish slate	Bronze
900	Dull slate — blackening	Bronze, faint purple tinge
1000	Dull slate with faint indications of red oxide	Rich purple
1100	Red oxide — easily scaled	Dark slate, green cast

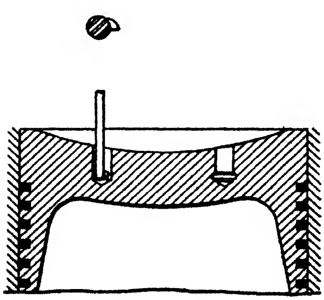


FIG. 11

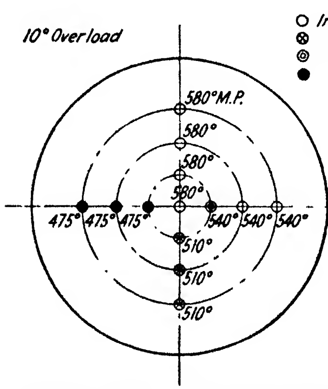
Another means of estimating temperature is by the use of soft metal or alloy fillings. The crown is drilled as in Fig. 11 with a $\frac{1}{4}$-in. drill. The twist drill is followed by a wabble drill which undercuts. This is made by upsetting a lip on a piece of $\frac{3}{16}$-in. drill rod just wide enough to enter a $\frac{1}{4}$-in. hole. Being crowded by the drill press feed this side cutting drill seeks bottom and under-reams the hole. Into this cavity, metals or alloys of known melting points are forced. Those that melt out in service indicate local temperatures.

Some "alloys" are very unstable, depending largely on mechanical mixing for their homogeneity. It is well to adopt the precaution of thorough stirring for all of them. A useful list of melting points is given in Table III.

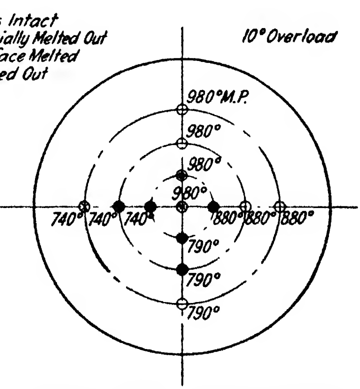
TABLE III
MELTING POINTS

Metal or alloy	Melting point, °F
33.3 Tin — 66.7 Lead	442
Tin	450
28.6 Tin — 71.4 Lead	469
Bismuth	513
16.7 Tin — 83.3 Lead	516
6 Tin — 94 Lead	557
Lead	621
Zinc	884
90 Tin — 10 Aluminum	914

The accuracy of this latter test is equal to the requirements. Deep and secure anchorage reduces error to a minimum. Very few firms are disposed to build the special engine and equipment required for accurate pyrometry of the piston face.



Aluminum Piston Temperature Chart
FIG. 12



Cast Iron Piston Temperature Chart
FIG. 13

The alloy method is the only simple one yet suggested for aluminum pistons which operate in a range below the temper color range of super-ascoloy. A charted temperature determination from an aluminum piston is shown in Fig. 12, one for a cast-iron piston in Fig. 13.

The general aim in design is to prevent *any part* of the piston face from exceeding 850°F at guaranteed overload when burning the most refractory fuel permitted by guarantee and to accept some variation of temperature downward toward the circumference. A degree of uniformity is secured in several ways. The metal thickness is varied increasingly toward the edge. The top ring is kept considerably below the end of the piston, partially equalizing the length of metal traversed by the heat taken up by local areas. The distance from piston end to top ring may be $0.1\,D$. D = piston diameter.

Heavy bosses in the piston head are to be avoided. They sponge up heat on account of their thermal capacity and flood the paths of conduction during the low-temperature intervals of the cycle when the heat flow should ebb.

Ribs, also, are undesirable and unnecessary. Their restraint of expansion causes pistons with four ribs to protrude at four points from true circularity; those with six ribs assume a bulged hexagonal section, etc. When seizure takes place these influences are quite marked.

For engines operating at the piston speeds of 750 to 900 ft per min it is usually satisfactory to make piston faces of a uniform thickness equal to $0.01\,D^2$. This gives proper temperature with excess strength when $D = > 8$ in. Below that diameter it is sometimes necessary to throttle the heat flow by thinning the ring belt and using a face thickness of $1/10\,D$. This is a rational expression for strength as may be learned by substituting finite values in the Grashof formulas.*

At higher speeds up to 1100 or 1200 ft per min it is well to make the face a trifle thinner at the center. Again, $1/10\,D$ is a good value, and it is good practise to make the edges at the junction with the ring belt thicker and equal to $0.012\,D^2$. Such a design is shown in Fig. 8.

When the speed makes aluminum-alloy pistons necessary the most desirable piston temperature is impossible. Aluminum softens above 600°F. The situation may be met by special designs of composite pistons, or somewhat inferior combustion at fractional loads may be accepted.

The question now arises: how much of the combustion-chamber volume should lie within the piston? With pistons

* Merriman, "Mechanics of Materials."

like Fig. 3 and Fig. 7 practically all of it may be contained. The exact amount depends on the valve gear and reversing mechanism. If inlet and exhaust valves open and close, respectively, at remote angles about the dead center, the inlet valve may strike the piston or the exhaust valve be struck by it, owing to overtaking. It is preferable, then, to keep the rim a little lower rather than to mill it locally to clear valves.

Some reversing mechanisms depress the valves more or less while they are out of action. This, too, must be provided for to avoid interference with the pistons.

When the cavity in the face of the piston is in the form of a spherical segment its radius is such as to make a rather shallow depression except in cases where the stroke-bore ratio L/D approaches the values 1.8 to 2.0, permitting the cup to be hemispherical.

For engines of $L/D = 1.25$, a very favorable ratio, the cup may be struck out with a radius equal to $0.7\,D$.

THE PISTON RING BELT

The number of rings and their width and thickness control the conduction of heat from piston to water jacket. Ordinarily their number equals $2\sqrt{D}$. If the ring width equals $\sqrt{D}/8$ and the number of rings is as above, then the total width of ring surface presented to the liner equals $D/4$ and the total surface is $\pi D^2/4$, which is quite rational since the piston surface, receiving heat by radiation, also varies as D^2 and the ring contact area equals the projected area of piston face.

Rings of the above width are required for two reasons. Many specifications, particularly those of our government departments, call for double seal rings like Fig. 14 which become too fragile in narrow widths. The slotted oil-control ring, Fig. 15, also requires some width to avoid fragility. As the total number of rings is not required for sealing, one or two of the lower ones may be slotted and still serve quite well for heat conductors. If special rings are not required all rings may be made as narrow as $D/32$ when the cylinder is without ports, but a general relation of total width of liner contact equal to $D/4$ should be preserved. This would give all pistons 8 rings.

The conductivity of aluminum being 4.4 times that of cast iron, half to three-fourths of the usual number of rings suffices even for higher piston speeds up to 1600 ft per min piston velocity. It is safe to recommend from experience with engines of $7\frac{1}{2}$- to 18-in. bore a number of rings such that the liner con-

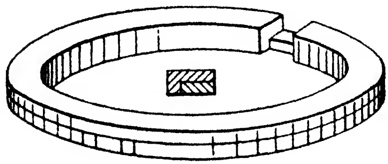


FIG. 14

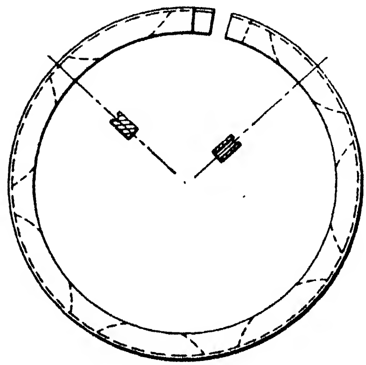


FIG. 15

tact width equals $D/8$ to a velocity of 800 ft, $5\,D/32$ up to 1200 ft, $3\,D/16$ up to 1600 ft per min.

When called upon to use cast-iron pistons up to 1200-ft velocity, some engineers having a strong prejudice against the aluminum piston, it is well to add one or two rings for the larger uncooled pistons (14 to 18 in.). Theoretically the mean cyclical temperature is the same and the total time of exposure the same regardless of revolutions or piston velocity, but the cycle thus far has shown best efficiency at about 800 ft mean piston velocity. Above that value more after-burning takes place and slightly more heat is reflected to the pistons.

The metal section in the ring belt tapers from approximately the piston face thickness down to just sufficient metal to resist the compressive stress. This provides a suitable path for heat conduction. For aluminum pistons this section may be much reduced.

COOLED PISTONS

Iron pistons of more than 18-in. diameter in four-cycle engines and pistons of far smaller diameter in two-cycle engines, depending on the thoroughness of scavenging, require cooling. This introduces a very special problem if fractional load operation is to be satisfactory. Two special designs are those of Krupp and Doxford. Krupp uses the mushroom piston, Fig. 16, the central portion being unjacketed and attached in such a way as to provide a long resistant path for heat travel to the piston jacket. This keeps the face at a suitable temperature.

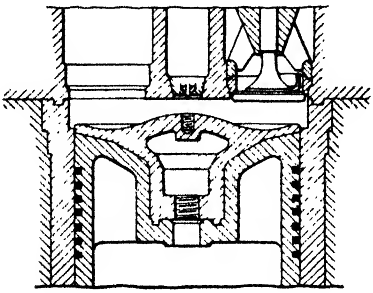


FIG. 16

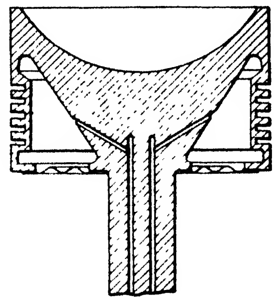


FIG. 17

The Keller patents which are the basis of Doxford pistons, Fig. 17, depend on mere thickness of face. To better the resistance to thermal stresses the piston head is made of forged steel. Both designs are heavy and suitable only for comparatively low speeds.

COMBUSTION CHAMBERS FOR SMALL ENGINES

Fig. 18 shows the pre-combustion chamber in which a portion of the fuel charge is burned or partially burned, furnishing a blast of hot gases and vapors to atomize and vaporize the major portion of the fuel and inject it turbulently into the combustion chamber proper.

Usually the volume of the pre-combustion chamber is 15 to 25 per cent of the total clearance volume, but there are exceptional designs in which it is relatively much larger. The total area of the orifices connecting the pre-combustion chamber with the main chamber often is 0.005 to 0.006 that of the piston area, although this too varies widely.

Fig. 19 shows a modern air-cell or after-chamber design in which fuel is injected through a pintle nozzle across the main combustion chamber, contained in a cavity in the piston or in the head, part of the fuel being carried into the air cell by the flow of air in that direction and by residual energy in the fuel jet. Whether it ignites in the main chamber first or in the after-chamber depends upon which portion of the air charge first reaches the temperature of spontaneous ignition. In the

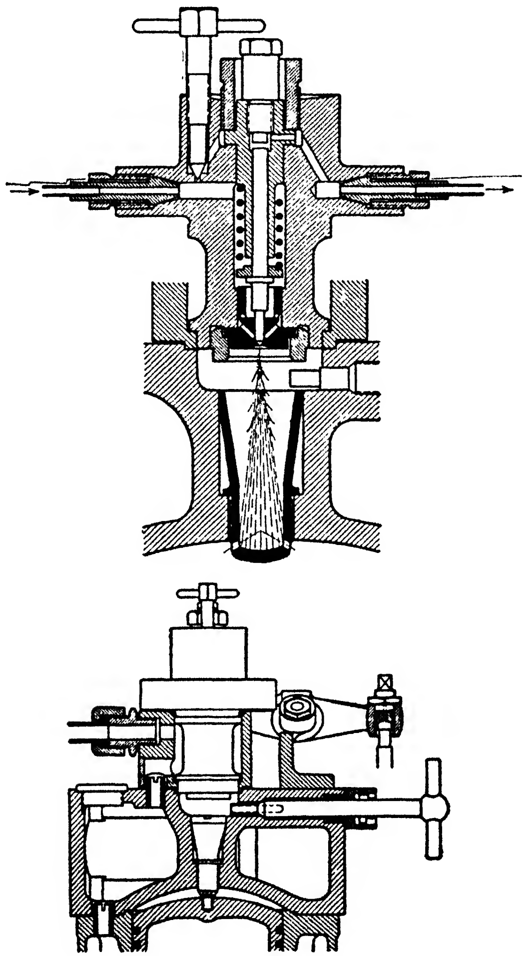


FIG. 18

latest designs by Lang, the foremost experimenter along these lines, the air cell is the igniter and reflux sets in producing effective turbulence in the main chamber as shown by diagrams, Fig. 20.

The volumes of the two chambers are related to each other in various designs as 70/30, 60/40 and even 50/50, the smaller being the air cell. The area of the orifice connecting the two chambers may be about 0.006 that of the piston area or larger.

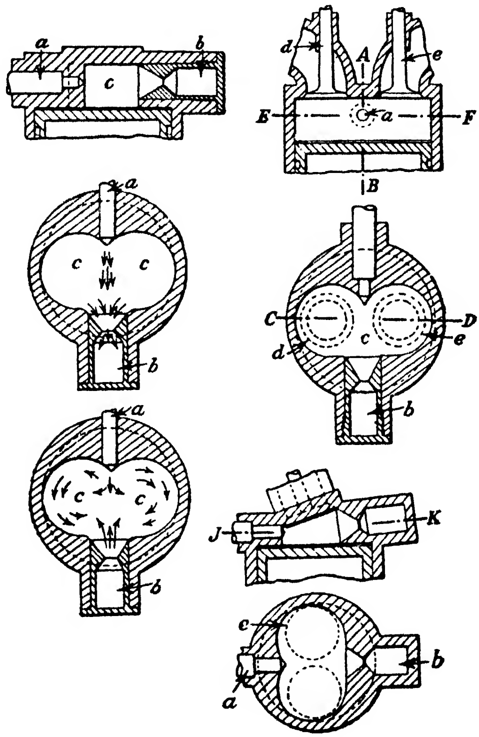
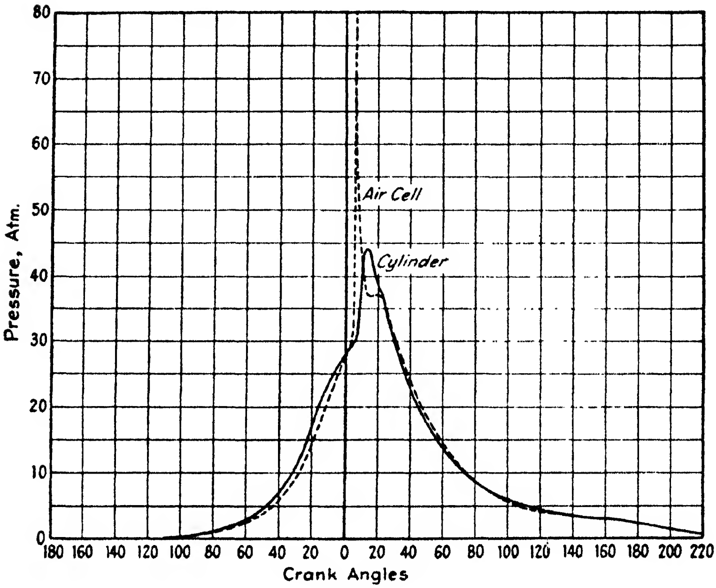


FIG. 19

Fig. 21 shows a spherical whirl or turbulence chamber design. The diameter of the sphere may be approximately 0.4 D where D equals the cylinder diameter. All the air is massed into the sphere and the connecting port except that small unavailable portion trapped in necessary clearances between the piston and head, in valve pockets, etc. The transverse area of the port may be 7 to 8 per cent of the piston area.

It is a property of each of these designs that when geometrically similar their linear dimensions are straight-line functions of the bore diameter. Thus, if a 6-in. bore engine has a whirl



Pressure, Atm.
80
70
60
50
40
30
20
10
0
180 160 140 120 100 80 60 40 20 0 20 40 60 80 100 120 140 160 180 200 220
Crank Angles
Air Cell
Cylinder

FIG. 20

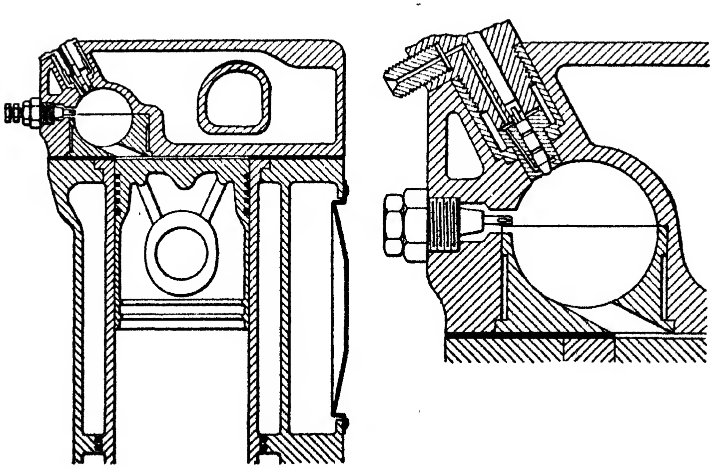


FIG. 21

chamber 2.4 in. in diameter, a contemplated 3-in. bore engine will have a whirl chamber of only 1.2-in. diameter. It is rather difficult to limit fuel penetration sufficiently for such small chambers; consequently it is well to use large bores and fewer cylinders where space and the degree of balance required permit.

Since all these designs have a very large amount of combustion-chamber cooling surface relative to their volume, large diameters and fewer cylinders also are favorable to cold starting and light-load operation.

The outstanding feature of all these designs is the bushing lining the special chamber and serving as a target for the fuel. These bushings are relieved to reduce their contact with the jacketed walls, and sometimes a degree of thermostatic temperature control is sought by fitting them so that the thermal contact is light or lacking at light loads, being increased in pressure by expansion of the bushing as the load and temperature increase.

The same laws apply to development of these bushings or liners for correct operating temperature as have been stated in Chapter II and in this chapter in discussing the piston face as a fuel target. If a reasonably effective compromise is not found the engine may fail to idle or if hot enough to idle well it may overheat and fail to develop a commercial rating.

For good starting and regular operation at fractional loads most of these smaller engines require not less than 450 lb compression. It is not advisable, however, to go to extremes in this direction. With very high compression a disproportionate amount of air is forced into mechanical clearances where it is unavailable and the combustion chamber walls are brought too close to the spray.

CHAPTER IV

THE FUEL NOZZLE

Four general types of devices for fuel admission have been used in solid injection diesel engines. They may be specified as follows:

1. (Fig. 22.) Simple nozzles with one or more orifices for discharge, directing fuel into the combustion chamber as delivered by timed fuel pressure impulses. They may be provided with lightly loaded check valves.

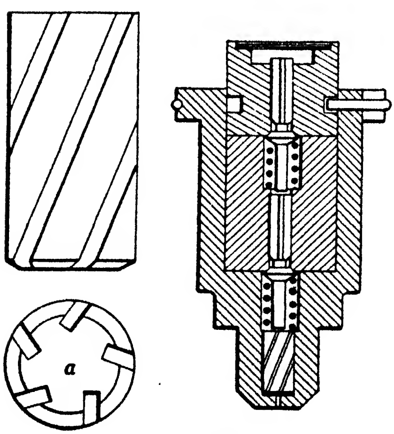


FIG. 22

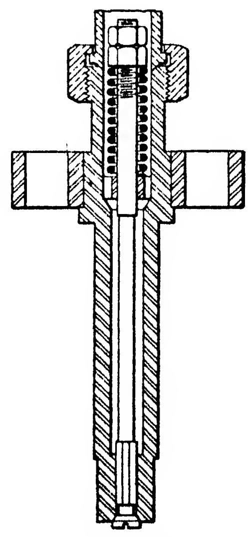


FIG. 23

2. (Figs. 23 and 24.) Poppet valve nozzles, loaded to "blow off" like safety valves under timed impulses of fuel pressure and discharging directly into the combustion chamber.

3. (Fig. 25.) Nozzles with mechanically operated needle valves served by a constant-pressure pump.

4. (Fig. 26.) Nozzles with "hydraulically operated needle valves, the fuel itself acting on the 'differential' area in timed

pressure impulses, tending to eject the needle from its guide, thus lifting it from its seat against spring pressure during the intervals of high fuel pressure."

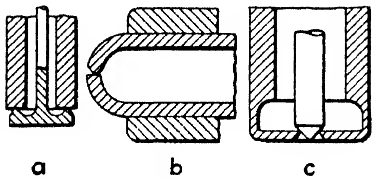


FIG. 24

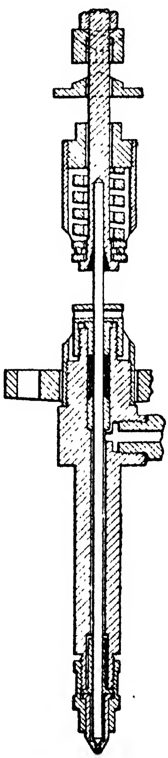


FIG. 25

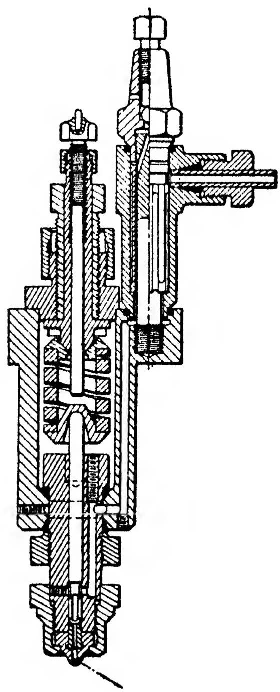


FIG. 26

Multiple-orifice nozzles are capable of serving combustion chambers which require a wide distribution angle. These chambers are typified by their approximation to cylindrical form of large diameter and moderate height. The nozzle usually is located on the cylinder axis, the fuel being injected radially and downward through five or more holes drilled at an included angle of 120 to 140°.

Fig. 22 shows a nozzle with one central orifice and helical whorl. By reason of the helical path which the fuel follows in reaching the orifice the jet is whirling as it issues. As soon as the guiding influence of the orifice is passed the fuel spreads in a hollow cone, being more or less atomized owing to disruption of the widening orbit of travel. A helical path is not strictly necessary; rather it is a primitive form. Modern nozzles of this type deliver the fuel to the orifice through three or four tangent saw curfs, only as in *a*, Fig. 22, these slots forming a closed path by the seating of the whorl bushing or plug into the spray tip. All attempts to produce wide angle distribution with this nozzle have failed.* It is used largely in conical or opposed conical combustion chambers, in which fuel is to be distributed in a hollow cone of 15° to 30° angle. Since much of the spray energy is directed into centrifugal force the atomization is perhaps better than that of other types although the cone of distribution shows concentration into a number of rays equal to the number of passages in the whorl. The penetration is much limited considering the size of the orifice. This nozzle is therefore especially suited to engines in which no hot target is present to limit spray travel.

The velocity of the oil through the tangent passages may be varied by varying their cross-section. In some degree the size of the circle to which the passages are tangent may be chosen. As the energy of rotation is increased (experimentally) the cone of distribution is widened until a point is reached at which penetration is so reduced that it actually narrows again.

The nozzles shown in Figs. 23 and 24 are rarely used now although the poppet type has recently re-appeared in the Dorner engine. Assume for a 40-hp engine cylinder a charge of 0.05 cu in. passed through a poppet with a $\frac{1}{4}$-in throat. The fuel velocity is 9600 in. per sec. The duration of injection is about 1/100 sec. Therefore, the projected length of charge is 96 in. The cross-section of the jet at the throat is then $0.05 \div 96 = 0.00052$ sq in.

* **Hawkes, "Some Experiments in Connection with the Injection and Combustion of Fuel Oil in Diesel Engines," pages 28–32.**

The valve throat circumference is $0.25 \times \pi = 0.7854$ in., so the required valve opening is $0.00052 \div 0.7854 \doteq 0.0007$ in. Naturally this is quite impracticable. This nozzle came down from the day of low-pressure injection against the hot surface. Even then it was difficult of design and manufacture in spite of its apparent simplicity.

To make the valve frequency high and the over-travel under the fuel impact and valve inertia as little as possible the valve spring scale should be great. This is opposed to other practical requirements which are that delicate adjustment of a calibrated spring should not be required and that a small amount of wear should not greatly alter the adjustment.

Figs. 24, *a*, *b* and *c*, illustrate attempts to avoid the spring and to use the elasticity of metal parts in tension or flexure instead. They have all failed in practise for the same obvious reasons as those given in respect to Fig. 23.

Perhaps the difficulty of designing a fixed orifice nozzle to be used with cam-operated pumps for variable-speed engines contributes most to the persistent attempts to use the Fig. 25 nozzle types in high-speed engines. The poppet-controlled orifice is self-adjusting and should accommodate itself to any engine speed, but it often fails to function properly because of its kinetic properties.

The fixed drilled orifice is less objectionable for a wide speed range. It must be admitted that we are still searching for a workable automatic nozzle for variable-speed engines although the nozzle with the cam-operated valve of the common-rail system is quite satisfactory for such work since the duration of injection is adapted to the speed of rotation.

Marine propulsion calls for variable speed in ratio 1 to 3 or more, and the requirements are fulfilled by the solid injection engine using type 3 or type 4 nozzles, Figs. 25 and 26, but in this service the power required varies as the cube of the speed. At half engine speed the resistance is only one-eighth the full power and the brake mep one-fourth the rated output. Under these favorable conditions exact functioning of injection mechanism at low speed comparable with that required at full load is not essential.

Fig. 27 represents a development in class I which is rather startling in its simplicity. The tip is designed like all other multiple-orifice tips. The connecting tube is called a "capillary" and is described as holding fuel in check between injections by capillary attraction, but it is really a drawn steel tube of quite measureable dimensions having a bore of $\frac{1}{16}$-in to $\frac{3}{16}$-in. diameter. Engines operate with remarkably good combustion when equipped with this simple nozzle.

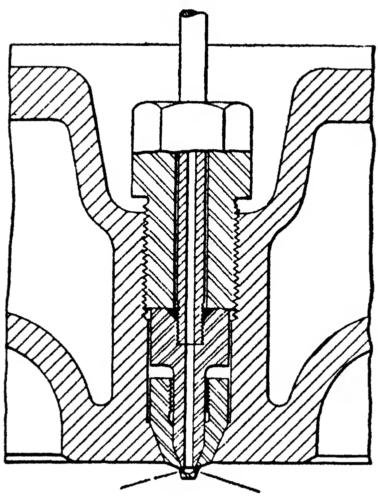


FIG. 27

The effect of capillarity is perhaps overestimated. The check valve is merely removed from the nozzle and placed near or at the pump. The pump by-pass reduces pressure suddenly to prevent or at least minimize dribble.

This system works best on good light fuel. On common run of market fuel oil it is, in the author's experience, inferior to nozzles having cam-operated spring-loaded valves, Fig. 25, or nozzles of the automatic differential type, Fig. 26.

Certain attempts are also being made to use the open fuel nozzle, Fig. 27, with perhaps a non-return valve in the nozzle body and a remote cam-operated fuel-admission valve. The fuel pump is of the constant-pressure type. These systems free the cylinder head of some operating mechanism. They work well on good fuel.

The remote admission valve is also used with the differential type spring-loaded valve, Fig. 26. With this combination results may be quite unsurpassed.

THE NOZZLE TIP FOR COMMON-RAIL AND IMPULSE SYSTEMS

That part of the nozzle which projects through the cylinder head wall into the combustion chamber is usually called "the tip." It is detachable for easy repair or removal and is provided with a seat for the needle valve if such a valve is used. The tip is perforated with one or more orifices for the passage of fuel into the combustion space.

Fig. 28 shows a tip with single jet orifice and pintle.

The pintle is a little projection of the needle valve reaching through the injection orifice. It permits use of an orifice of larger diameter. With or without the whorl the pintle improves atomization and thus limits penetration. The annular

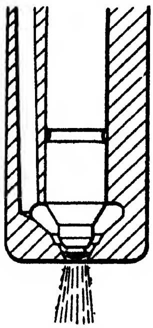


FIG. 28

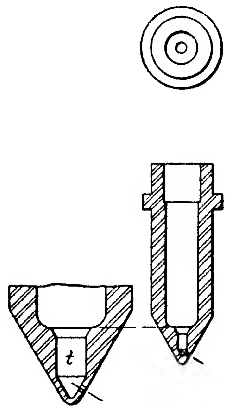
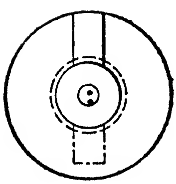


FIG. 29



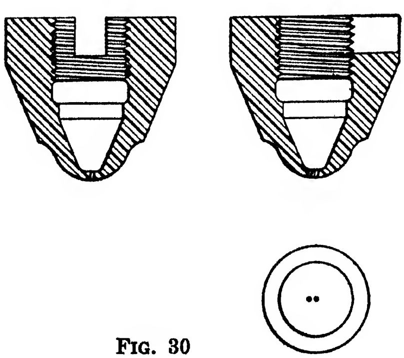


FIG. 30

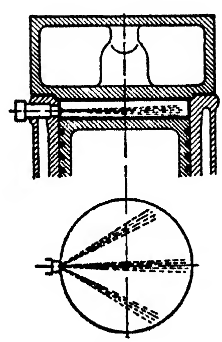


FIG. 31

jet produced by this nozzle has far less penetrating power than a solid one of the same area. This non-plugging nozzle has practically superseded all others for small high-speed engines.

Fig. 29 shows a conventional multiple-orifice tip with drilling radially symmetrical to the axis for conical distribution.

Fig. 30 is a multiple-orifice tip with drilling on a transverse diameter for fan spray. This tip is used when the nozzle enters the combustion chamber radially or nearly so as in Fig. 31.

ATOMIZATION AND PENETRATION

The secondary importance of atomization is made evident when one considers the wide difference in the quality of spray produced by the various types of nozzles. It is further evident when one considers that the orifice is often designed as such with no reference to atomization. If engine types exhibited broad variations in comparative economy we might point to the degree of fuel atomization or lack of it as cause, but modern diesel engines are almost monotonously alike and regular in their thermodynamic performance. The nozzle tip chosen is one which will properly distribute rather than atomize the fuel.

After leaving the spray tip the fuel jets penetrate the combustion chamber by virtue of their own kinetic energy. This implies the necessity of mass as well as velocity and is opposed to fine atomization.

Atomization effects from drilled multiple-orifice tips are largely on the surface. The apparent effect is enhanced if the orifice is very short, and there is a notable decrease in atomization and increase in penetration if the orifice is quite long. The usual wall thickness through which the orifice is drilled is two drill diameters, except when great penetration is desired. Then it may exceed six or eight diameters.

Certain European engines use multiple-orifice tips so drilled that the degree of penetration is just sufficient, no target being required. This results in very fine drilling. Domestic fuels allow no favorable experience with nozzle drilling smaller than 0.010 to 0.012 in.

Frequent stoppage of smaller holes due to impurities in fuel overweighs any other advantage. Drilling strictly on the orifice basis results in over-penetration for all sizes up to at least 18-in. bore with central nozzle. This is discussed and the manner of interposing a hot target for the spray treated in detail in Chapter III.

The angle at which the jets of the axial multiple-orifice nozzle impinge on the piston is a factor in controlling penetration. This angle is determined experimentally. Usual included jet angles are

120°	for a	stroke-bore	ratio of	1.50
130°	"	"	"	1.25
140°	"	"	"	1.00

FUEL NOZZLE ORIFICE DESIGN FOR COMMON-RAIL AND IMPULSE SYSTEMS (FOUR CYCLE)

At full load most engines with the central or axial multiple-orifice nozzle work best at a fuel pressure of 4000 to 5000 lb. Less pressure results in loss of rating or economy except with the pintle nozzle used in small engines where 1750 to 2000 lb pressure suffices. Greater pressure effects little or no improvement and is therefore avoided to ease pumps and piping.

The number of holes in the nozzle tip is chosen from experience about as indicated below

Piston velocity, ft per min	Number of holes in spray
800	5
1200	6–8
1500	10

A greater number of nozzle holes is used for higher speeds to avoid the too great fuel jet impact that would result by the use of a common number of orifices for all speeds and enlarged diameter of drilling for fast-running engines.

In designing nozzles for four-cycle engines a satisfactory basis for calculation is a fuel economy of 0.40 lb per bhp-hr, a fuel density of 7.5 lb per gal, an orifice coefficient of 0.65, a fuel pressure of 4000 lb per sq in. above the combustion chamber pressure and a discharge interval of 20° of crank travel. Under the prevailing law of flow $V = \sqrt{2gh}$, the fuel is delivered at a velocity of very nearly 800 ft per sec or 9600 in. per sec. The volume of a single discharge is obviously

$$\left(\frac{0.4 \times \text{bhp} \times 231}{7.5 \times 60 \times 0.5\ \text{rpm}}\right) \text{cubic inches}$$

The projected length of a single discharge is

$$\left(\frac{20}{360} \times 9600 \times \frac{60}{\text{rpm}}\right) \text{inches}$$

Dividing volume by length the aggregate effective area of the orifices is found to equal (0.00001283 × bhp) square inches. This is increased by application of the orifice coefficient to

$$\frac{0.00001283}{0.65}\text{bhp} = 0.0000197 \text{ bhp}$$

For the five-hole tip which is in most common use each orifice area becomes

$$\frac{0.0000197}{5}\text{bhp} = 0.00000394 \text{ bhp}$$

The diameter of each is then, for five holes, equal to

$$\sqrt{\frac{0.00000394}{0.7854}\text{bhp}} = (0.0022\sqrt{\text{bhp}}) \text{ in.}$$

Assuming the case of a 20-hp cylinder at 400 rpm the diameter of drilling works out to

$$0.0022\sqrt{20} = 0.010 \text{ in.}$$

which is a dimension in regular commercial use for cylinders of this size.

This is true for any speed, but if great over-penetration is feared the formula may be applied to the cylinder as rated at 750 to 900 ft piston velocity and the number of holes increased in direct proportion as the speed and rating are increased.

A manufacturer's schedule for nozzle drilling is given in Table IV. These engines are rated at 90 to 100 lb bmep. The table also includes calculated sizes.

TABLE IV

Hp	Bore, in.	Stroke, in.	Rpm	Bmep per sq in.	No. holes	Actual drilling, in.	Calculated drilling, in.
20.00	7½	10½	400	90	5	0.010	0.010
28.33	8½	12	350	....	5	0.012	0.0119
62.50	8½	12	800	90	10	0.012	0.0124
37.82	9½	14	300	....	5	0.014	0.0138
50.00	10½	14	350	....	5	0.016	0.0158
58.33	11¼	16	277	....	5	0.016	0.0171
70.00	12½	18	257	....	5	0.018	0.0187
100.00	15	22	225	....	5	0.020	0.0224

Engines having no positive spring-loaded valve in the nozzle may require somewhat smaller drilling as the duration of discharge may be indeterminate.

Having designed the jet drilling it is next advisable to make certain that the passage t, Fig. 29, joining it with the seat opening or throat is adequate. This passage should be as short as possible so it too may be treated as an orifice with sufficient accuracy.

For slow-speed engines the pressure drop due to entry to this passage may be 100 lb per sq in. The jet drilling is designed for a delivery pressure of 4000 lb per sq in. above the prevailing pressure in the combustion chamber. The relative volocities of the fuel in jet and throat are inversely proportional to their respective areas, also to the square roots of their pressure drops. Thus the area of the throat should exceed that of the total jet drilling area in ratio

$$\sqrt{\frac{4000}{100}} = \sqrt{40} = 6.66$$

It is common practise to make tip and throat drilling standard for a series of engines having a rather wide range of cylinder displacements, let us say a range in which the largest exceeds the least eight times.

Taking a practical example, the five-hole 0.020-in. drill spray tip in the table would require a throat diameter equal to

$$\sqrt{5 \times 0.02^2 \times 6.66} = 0.115 \text{ in.}$$

It is actually $\frac{1}{8}$ in., and this size is used for the whole series.

For high-speed engines it is sometimes advisable to accept a throat pressure drop of 200 lb per sq in. This merely increases the fuel pressure by that amount and is not a serious matter when cam-operated sprays are used. Two hundred pounds' throat drop should not be exceeded with automatic or pressure-actuated fuel valves.

THE FUEL NOZZLE VALVE SEAT FOR COMMON-RAIL AND IMPULSE SYSTEMS

Before the needle valve seat can be dimensioned its type must be chosen. Both cone seats, Fig. 25, and flat seats, Fig. 26, are used.

Two diametrically opposite considerations influence the choice. The flat seat may be reproduced cheaply by methods that are inherently accurate. It must be said that great accuracy is essential to the use of the flat seat. The cone seat admits of a far lower order of workmanship which is fortunate as a high degree of accuracy in its production is not possible with ordinary machine-shop methods. The perfect centering of needle cone and seat cone, if ever such originally, is soon destroyed in operation since the seat cone drifts laterally as it wears owing to lack of homogeneity in the material.

The methods of finishing flat seats are quite as rapid for hardened as for soft material. Those available for the female seat cone are suitable only for soft material.

Flat end needles and seats are first ground on the guiding surfaces. They are next ground and lapped on the seat end. This latter operation produces a square flat face of mirror-like finish.

Cone pointed needles may be finished accurately by grinding. The tip is drilled or reamed leaving a very narrow seat perhaps $\frac{1}{32}$ in. wide. After assembly the hard cone pointed needle is seated in the soft metal of the tip by a hammer blow on the end of the needle spindle.

Two inaccuracies are prevalent in assembled nozzles. The valve needle may approach the tip at a slight angle rather than normally. The needle axis may not be concentric with the tip axis. The latter error is the more prevalent as it is cumulative from a number of machine operations on nozzle body, tip and tip nut. Its effect on the flat seat is of course nil. For this reason alone, many, in fact most, cam-operated valves with cone seats are quite frankly unguided at the seat end as in Fig. 25.

If it were not for the almost inevitable failure of the two cones to center, causing the needle to bind in its guide, there would perhaps be no valid reason for using flat seats at all. Experience teaches the use of cone seats with cam-operated fuel valves having the seat end unguided. When needles run in lapped bushings to avoid packing and are guided close to the seat and especially when they are automatically operated by fuel pressure the flat seat is essential to avoid sticking of the valve as it closes owing to lack of concentricity. This condition is common and distressing.

UNIT COMPRESSIVE STRESS ON FUEL VALVE SEATS

It is common practise to load cam-operated fuel valves with a spring pressure such that the valve is held open by a test pressure of 6000 lb per sq in. Automatic sprays are lifted at the working discharge pressure. The $\frac{1}{2}$-in. needle of the nozzle,

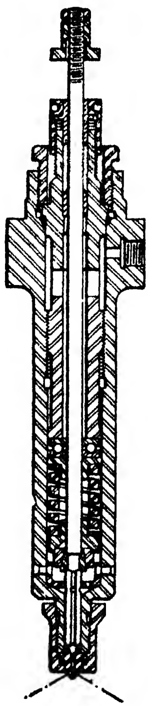


FIG. 32

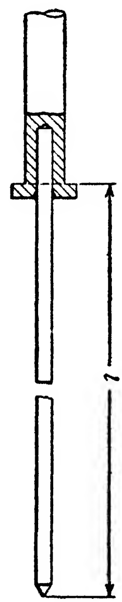


FIG. 33

Fig. 32, with an area of very nearly $\frac{3}{16}$ sq in. requires a spring load of 1200 lb for 6000 lb fuel pressure. In order to insure tightness when working it is necessary to stress flat seats to 20,000 and sometimes to 40,000 lb per sq in. when unrelieved by fuel pressure, cylinder pressure, etc. In round numbers the seat throat may be $\frac{1}{8}$ in, the outside diameter of seat $\frac{1}{4}$ in., the annular area 0.0491 sq in. − 0.0123 sq in = 0.0368 sq in., which under the above conditions results in a stress of 32,600 lb per sq in. In service at 4000 lb per sq in. and up, the seat loading is relieved by at least the force of the fuel pressure against the annular area outside the seat reducing the unit compressive stress to a little less than half the initial loading or 10,000 to 20,000 lb per sq in.

If the cone seat of 60° included angle is used the unit pressure is increased in the ratio 4/3 when the seat is a truncated cone half the height of a perfect cone.

At 4000 lb fuel pressure one-third of the spring force or 400 lb is available when the needle is off its seat to accelerate any operating gear as well as to overcome friction. At 5000 lb fuel pressure one-sixth of the spring pressure or 200 lb is useful for these purposes. The friction of needles packed with well-lubricated packing is about $12\frac{1}{2}$ lb for $\frac{1}{4}$-in. needles and 25 lb for $\frac{1}{2}$-in needles. The remaining spring force is ample in either case to accelerate any well-designed operating gear.

THE FUEL NEEDLE OR SPINDLE FOR COMMON-RAIL SYSTEM

The nozzle shown in Fig. 25 is a primitive form and is still popular. Its needle is a length of common drill rod, usually $\frac{1}{4}$ in. in diameter. The needle valve spring furnishes the accelerating and retarding force for the spray valve operating mechanism, and to overcome the inertia of unduly heavy gear it has been common to use 500 to 600 lb spring pressure. As the size of the engine increases the needle grows longer and becomes a very unstable column, particularly so as the bore through which the needle passes offers no support. This bore is a clearance hole to permit the passage of fuel. An example from practise is shown in Fig. 33. Although loaded with 600 lb spring pressure the critical load (P) of this column under the Euler formula is but

$$\frac{\pi EI}{l^2} = 630 \text{ lb}$$

This unstable column is destructive to the valve seat. It affects the accuracy of the needle movement adversely on account of the indefinite axial shortening of the needle under compression which must be relieved in lifting. It is particularly annoying in multicylinder engines in which the several sprays may vary in action owing to their individual peculiarities of deflection.

Fig. 32 shows a more advanced design in which the needle is in tension when lifting. The size of this needle is dictated by practical considerations as the lifting stress is insignificant.

FUEL CAMS FOR COMMON-RAIL SYSTEM

On account of roller interference it is quite difficult to design four-cycle engine cams with the lift graduated to produce a power diagram of any given profile. It may be possible to do so if an abnormally large cam is chosen, particularly if such a cam operates against a very small roller. The large cam is objectionable on account of the noise of impact due to its high peripheral velocity. The small roller often results badly on account of the large relative pin diameter, for if the roller pin friction moment is too nearly the adhesion moment the roller may fail to turn regularly and promptly flatten.

The following represents good practise in the design of four-cycle injection cams. Two-cycle engine cams present broader possibilities as the arc of operation is twice as long and therefore not so much influenced by the interference of the roller with the desired profile. However, it is doubtful if anything is gained by any profile other than the straight line.

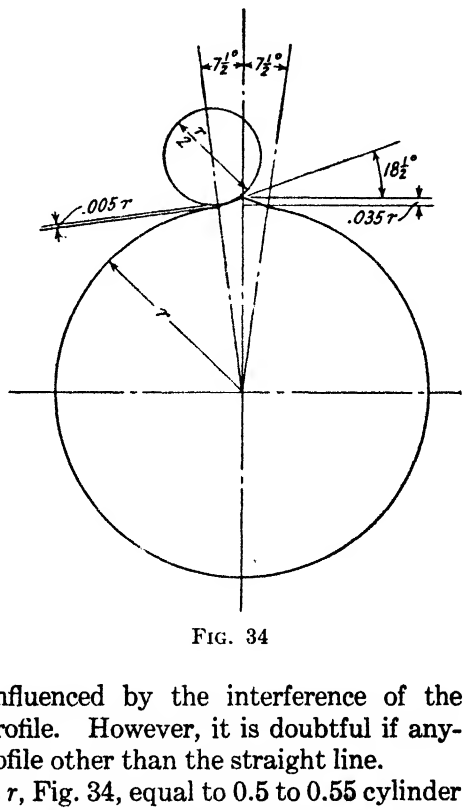


Fig. 34

Make cam drum radius r, Fig. 34, equal to 0.5 to 0.55 cylinder radius. Fix total rise at nose, including clearance, as 0.035 r, where r equals chosen cam radius. Make roller diameter $1/2\ r$. Let origin of nose surface contact with drum at $7\frac{1}{2}°$ each side of center line, and make the $7\frac{1}{2}°$ line likewise the position of roller center at the instant of roller contact. The slope of cam nose with its base will then be 18° 27′ (18½°), and the clearance between roller and drum during dwell will be (0.005 r).

Roller travel up the slope of the cam follows the laws of tangent cam motion. About the sharp peak which comes into contact with the roller 3° and 34′ before full lift the travel is equivalent to crank motion. The descending profile is the reverse of the rise.

By triangulation, Table V is developed in which Kr = cam lift at given angles when r = cam radius.

TABLE V

Four-cycle camshaft angle from cam	K = constant = lift for r = 1 in.
0	0.0304
1	0.0296
2	0.0272
3	0.0232
3° 34′	0.0204
4° 30′	0.0146
5° 30′	0.0093
6° 30′	0.0045
7° 30′	0.0000

For fuel cams of a two-cycle engine the camshaft angles are double those tabulated.

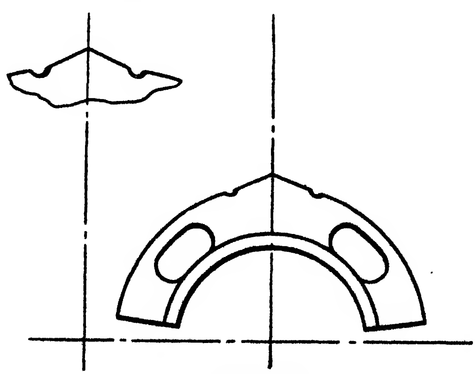


FIG. 35

Fig. 35 shows a shop drawing of a cam segment. The concentric part of the profile is ground on a cam grinder. The grinding wheel is lifted just as the cut reaches the clearance groove and returned after the nose has passed under the wheel. The nose is ground on a surface grinder. The very slight radius desired at the junction of the two slopes is not shown. Operation of the engine produces this promptly and, if the material and heat treatment are good, uniformly.

An excellent material for fuel cams is a crucible steel of the following alloy

Carbon	2.20– 2.40
Chromium	14.00–15.00

Its heat treatment is simple. The work is brought up very slowly to 1650°F and then promptly to 1675°F, when it is quenched in oil.

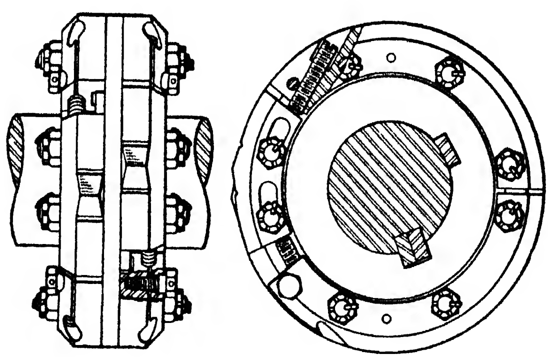


FIG. 36

An assembled fuel cam complete with its adjustments is shown in Fig. 36.

The preferred direct fuel valve operating motion is shown in Fig. 37. The beam operating motion used when the camshaft is in the engine crankcase is illustrated by Fig. 38. It halves the fuel valve lift. When the beam is made of steel its mass is excessive so it is usually made of aluminum. Very deep and stiff sections are used to avoid excessive deflection which on account of the low modulus of elasticity of aluminum may be a serious influence at light load when the fuel valve lift is small.

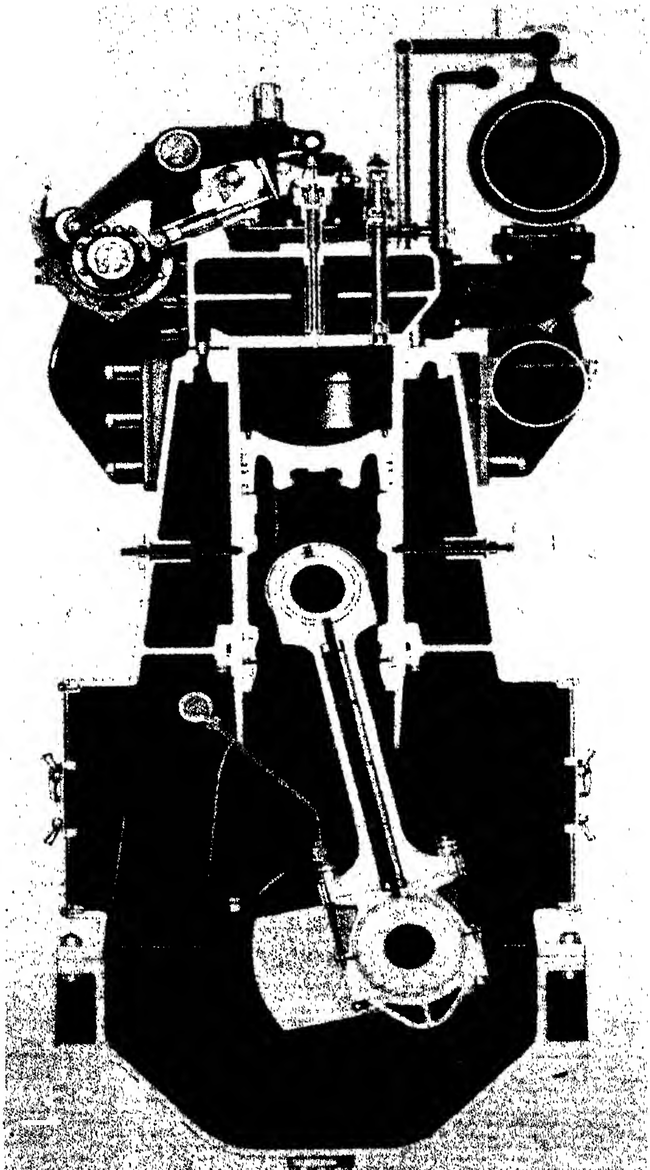


FIG. 37

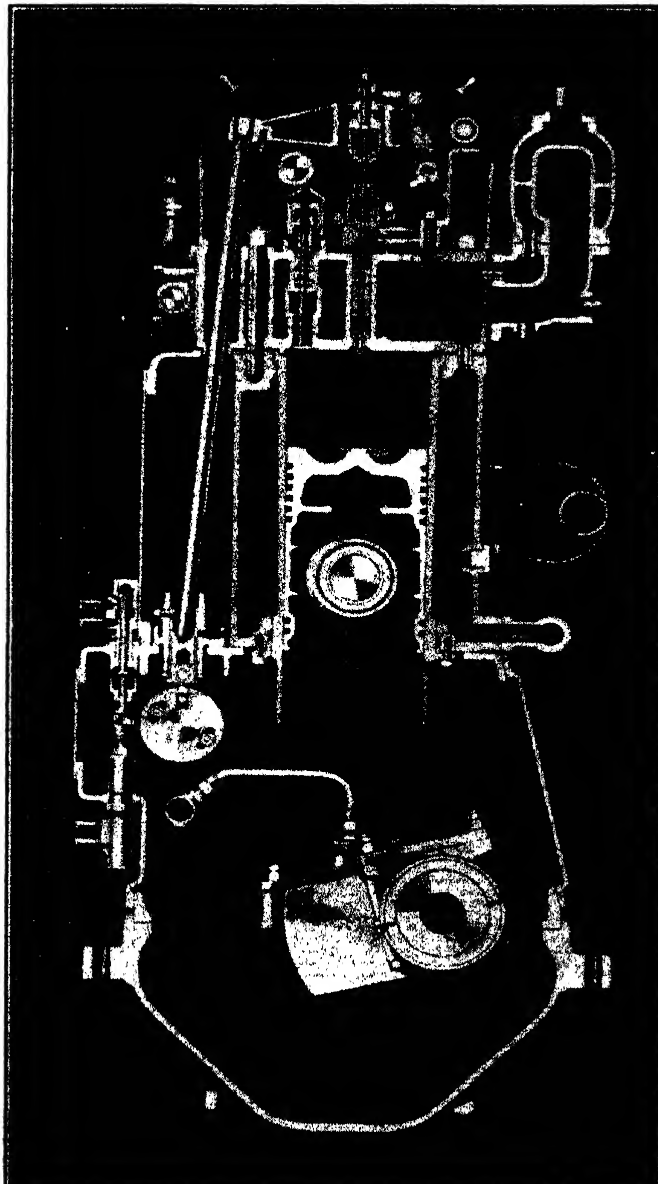


FIG. 38

THE PRESSURE-OPERATED FUEL VALVE

The nozzle type 4 is provided usually with the same tip as type 3, and the same considerations apply to design of the needle seat. As its movement is uncontrolled the needle spring and accessory parts should be kept light to avoid undue impact on the seat. A $\frac{5}{16}$-in. needle has been used quite extensively. The stuffing-box is rarely employed as the action of the packing in absorbing and giving out fuel at the high pressure now in use modifies the fuel delivery disastrously. The needle operates in a bushing which is ground and lapped to so perfect a fit that the leakage of fuel is insignificant.

The design of the valve seat determines the properties of the nozzle. The valve opens at one pressure and closes at another like any boiler safety valve. In fact, it may be said to have a "blow-down characteristic" similar to that valve.

Referring to Fig. 39, let the large diameter of the needle be $\frac{7}{16}$ in., the diameter of the seat hole in the spray tip $\frac{1}{8}$ in., the

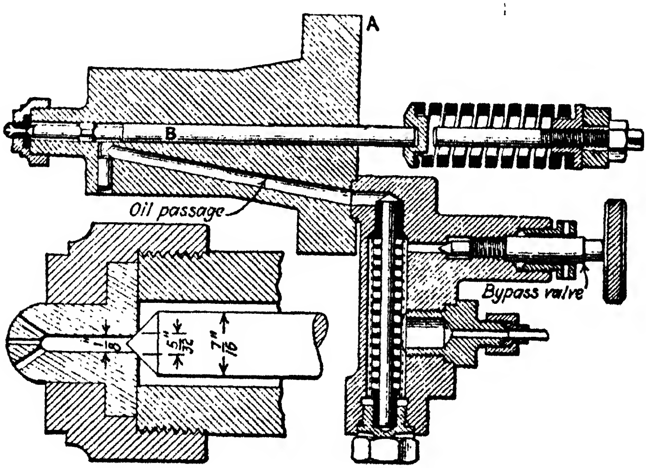


FIG. 39

diameter of the seat itself $\frac{5}{32}$ in. Let us assume that compression is 400 lb, the engine being a cold starter, and the fuel pressure at the time of the lifting of the needle 4000 lb. The forces tending to lift the needle are as follows. When it is on its seat, 4000 lb pressure acts on an annular area $\frac{7}{16}$ in. outside diameter and $\frac{5}{32}$ in. inside diameter. This area equals 0.13116 sq in., and the lifting force due to the applied pressure equals 524.64 lb. We are told by Reuleaux that a valve seat of this sort is acted upon by a pressure between the seating surfaces equal to the mean of the pressure on the inside and outside of the seat. This may not apply strictly to very narrow, highly stressed seats. Reuleaux of course assumes a liquid film between the seating surfaces which is acted upon transversely by the pressure outside the seat. Thus we may have another annular area $\frac{5}{32}$ in. in diameter by $\frac{1}{8}$ in. diameter with a surface of 0.00690 sq in. This is acted upon by the mean pressure between 4000 lb applied by the oil and 400 lb compression pressure or 2200 lb, resulting in a lifting force of 15.18 lb. And further we have the compression pressure of 400 lb acting on the area of a $\frac{1}{8}$-in circle or 0.01227 sq in., resulting in a lifting effect of 4.91 lb. These latter two forces are small and not especially significant. The three forces sum up to 544.73 lb, which is the proper spring load for this needle valve for lifting at 4000-lb oil pressure. The instant the valve is off its seat, the oil pressure works against the full area of the $\frac{7}{16}$-in. circle of 0.15033 sq in. and the needle is lifted to the stop. Before the needle valve can return to the seat the pressure must fall to 544.73 lb $\div$ 0.15033 or 3624 lb. Thus theoretically the valve seats when the pressure has fallen to 376 lb under the lift pressure. Greatly increasing the seat diameter will increase this drop of the pressure, very possibly causing the nozzle to deliver oil at velocities too low for suitable operation. By further and exactly similar calculation using the pressure of 3624 lb acting on the needle after it is seated and 400 lb prevailing at the spray tip, we may show that a force of 50.62 lb unbalanced pressure holds the needle to its seat under this condition. This force of 50.62 lb distributed over the seat area of 0.00690 sq in. stresses the seat above the film pressure between its surfaces 7336 lb per sq in. There is, therefore, very small chance of leakage if the seat is reasonably well made. The fuel pump indicator card, Fig. 40, shows at *a* the point of spray opening. The rise above *a* is due to a rate of delivery greater than the

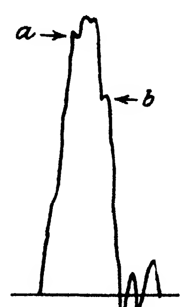


FIG. 40

orifice flow at the lift pressure. Point *b* indicates closure of the spray needle. The scale of these cards is 3600 lb per in. The indicator is fitted with a piston of $\frac{1}{16}$ sq in. area so that its displacement is small compared to the pump discharge.

There is good reason for limiting the "blow-down" to a maximum of 500 lb but preferably to as little as 250 lb.

THE FRICTION LOAD CHARGE

The volume of fuel under pressure in the system beyond the pump discharge valve at the time of cut-off is of great moment. The fuel pressure on spray and tubing will be relieved by the amount of the blow-down if the spray valve is spring loaded and completely relieved if it is not. When the amount of fuel discharged in reducing the fuel pressure to that required for the valve to seat exceeds the friction load charge the engine will tend to run at a higher minimum speed than desired under hand throttle control or will run irregularly under governor control.

It is good practise to limit the elastic discharge considered as apart from the forced discharge to not more than one-half the friction load charge volume. If the drop of pressure permitting the valve to seat is 500 lb or 33 atmospheres the elastic discharge will be

$$\frac{33 \times 70}{1{,}000{,}000} = 0.002310$$

cu in. per cu in. of clearance. The friction load charge usually is $\frac{1}{3}$ of the full-load charge, $\frac{1}{2}$ of which is then $\frac{1}{6}$ of the full charge. If the full-load charge is 0.060 cu in., for example, the elastic discharge should not be more than $0.060 \div 6 = 0.010$ cu in., and the maximum permissible volume of fuel between pump plunger and spray needle seat would be $0.010 \div 0.0023 = 4.3$ cu in. It is always well to make the volume even smaller if possible.

If the governor-controlled by-pass valve drains the fuel discharge system the above does not apply but governing will be less sensitive on account of leakage of cylinder gases back into the fuel system.

IMPULSE PUMP DELIVERY AT LOW SPEED

When the cam-operated pump delivers at reduced engine speed it is unable to maintain full pressure against the orifices. If the fuel valve is of the open type with no great resistance the pressure head maintained will vary as the square of the engine speed. Thus at one-third speed, or about as slow as the average marine engine with a light flywheel will turn, the fuel pressure head above compression and combustion pressure will fall to one-ninth the full pressure. Fortunately this is not an altogether impossible condition as comparatively little power is required for propulsion at this speed and engines always idle best under reduced fuel pressure.

When nozzles with automatic spring-loaded fuel valves are used, one of two phenomena appears. At reduced speeds the

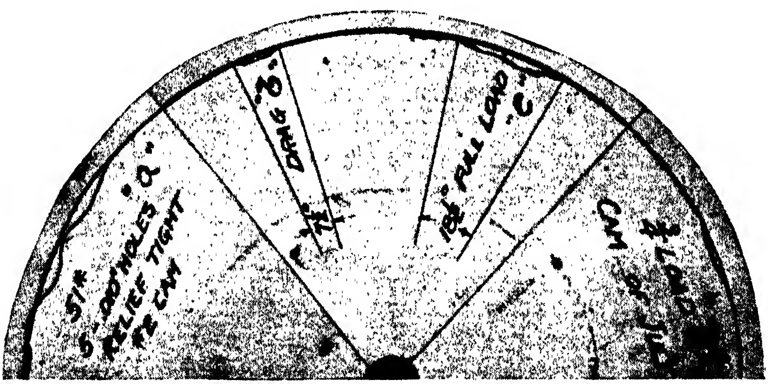


FIG. 41

pump may put up pressure and open the fuel valve. Upon the escape of this pressure the valve closes until a second wave is built up during the same pump movement and a second discharge takes place. This is shown at *a* in the chart of fuel valve movements, Fig. 41. This chart was made by attaching an indicator pencil motion directly to the valve spindle and allowing the pencil to record on a revolving disk placed in proximity to the spray and driven from the crankshaft. For very small discharges under much-reduced speed the valve may open only enough to allow the fuel to ooze out as in *b*, Fig. 41; *c* shows a normal discharge and *d* a discharge from a pump operated by a cam having no excess velocity of plunger movement. The favorable conditions of ship propulsion, alone, make these operating conditions serviceable. However, engines using this injection mechanism will operate at full mep at various speeds down to one-third design or rated speed with fair economy and performance. Table VI shows an economy test under such conditions

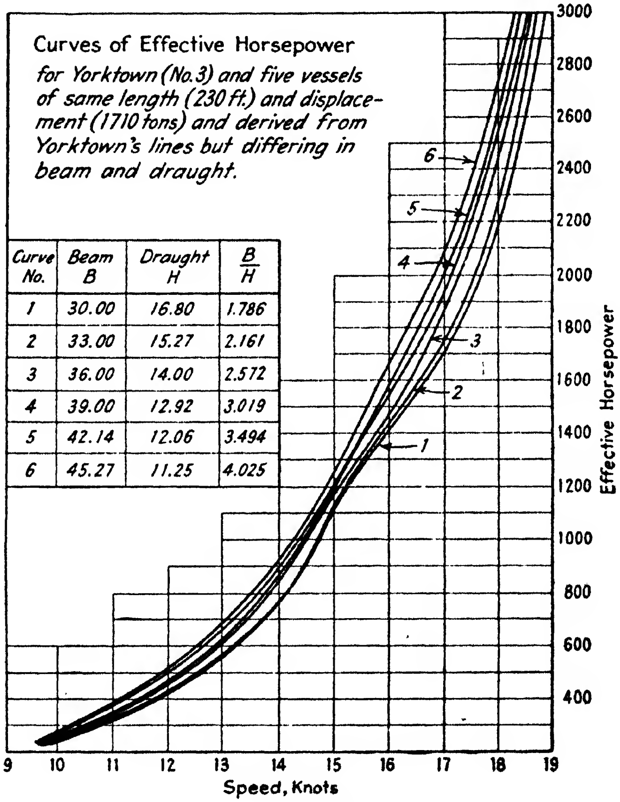


Fig. 42

and Fig. 42 a characteristic curve of the horsepower required to propel a ship.

TABLE VI

VARIABLE SPEED TEST — CONSTANT TORQUE

Rpm	Bhp	Fuel lb per bhp-hr
277	151	0.413
250	138	0.412
225	124	0.406
200	110.5	0.412
150	82 6	0.448
100	55.3	0.582
138	18.05	0.94*

* Reduced torque — $\frac{1}{2}$ speed $\frac{1}{4}$ torque.

The conditions described above occur also at full speed when the fuel pump leaks, when the nozzle orifice is too large or when the fuel valve spring tension is too great.

FUEL NOZZLE KINKS

Most spray nozzle devices are already of such clouded origin that the original inventors are forgotten. Design has been notable in the refinement of the various devices rather than in any recent discovery. As some of these improvements are vital they are well worth a few paragraphs.

The ball bearing used as a spring base in Fig. 32 serves two purposes. It permits slight rotation of the spring in operation which eases spring stress and seats the valve advantageously. It also permits adjustment of the spring without turning the needle on its seat. It will be understood readily that the seat is too heavily loaded to act as a step bearing without abrasion of the seat surfaces.

The vent valve at the top of nozzle assembly in Fig. 26 is used to discharge air when the system is primed. It uses a hardened ball and set screw in preference to the usual European needle valve. Its production and repair are cheaper and more certain of success.

The collet-type locknuts used on the stop and the spring adjustment in Fig. 26 are the invention of Mr. Hesselman. They permit of far more accurate adjustment than common locknuts which invariably pull back and destroy the desired setting of the screws.

The loose bottom in the stuffing-box of Fig. 25 takes up the packing automatically, and its sharp inner edge prevents the packing fiber from working into the spray body.

The tip, Fig. 27, provides far better contact and better tip cooling than the older types shown. Hot tips invariably cause the building up of carbon about the jets. Contrary to the usual belief which favors heating the oil in the tip before spraying, the tip cannot be kept too cool. Jacketed tips are used in Europe but they are too intricate to meet favor under American manufacturing and operating conditions.

As the jet leaves the orifice some mist is formed at the tip. This tends to deposit and carbonize on any near surface, developing in some hours veritable wasps' nests through which the fuel jets issue as best they can.

To avoid carbonizing, the tip should project through the combustion-chamber wall so far that the jets clear any near wall in passing by at least $\frac{1}{4}$ in. and for large engines $\frac{3}{8}$ to $\frac{1}{2}$ in.

DETONATION

To graduate fuel delivery it is sometimes necessary to allow the hydraulically operated needle valve to lift at a pressure as low as one-fourth that attained at full flow through the jets. This may be effective in cutting down detonation but only at the expense of reduced fuel economy and inability to burn the heavier grades.

CHAPTER V

THE FUEL PUMP

Fuel pumps may be divided into two classes and several sub-classes, thus:

1*a*. Constant-pressure pumps, charging a common rail or fuel manifold provided with an accumulator and by-passing the excess fuel through an automatic spring-loaded relief valve.

1*b*. The same except that the governing of the fuel pressure is by automatic or hand regulation of the fuel pump suction valves. Pumps 1*a* and 1*b* are used only with cam-operated fuel valves.

2*a*. Impulse pumps, cam or eccentric operated and timed to deliver fuel at the injection interval. Governing is effected by hand or automatic control of the suction valve or of a special by-pass valve. A pump is provided for each engine cylinder.

2*b*. Like 2*a* but governed by variable plunger stroke.

2*c*. Like 2*a* except that the governing is effected by partial filling of each forcing pump by means of a common measuring pump.

2*d*. May be like any of the class 2 pumps but the fuel pump is operated by a multiple cam and placed in communication with each fuel nozzle in its turn by a distributor. Often one pump suffices for all cylinders. For four-cycle engines the single pump makes a number of strokes per engine revolution equal to the number of engine cylinders divided by 2.

THE COMMON-RAIL OR VICKERS SYSTEM

The fuel pumps for this system are merely good hydraulic pumps. They may be motor driven, chain or gear driven from the engine crankshaft or operated by walking-beams from any convenient part of the crankshaft or camshaft.

The Vickers system of fuel injection perfected by Sir James McKechnie is distinguished from other systems by the use of cam-operated fuel valves, all of which are in constant communication with the fuel manifold, or "rail," Fig. 43. Fuel pressure is maintained practically constant within the manifold by the use of one or more accumulators, which often are round steel flasks $2\frac{1}{2}$ or 3 in. in diameter and 12 to 18 in. long. The elasticity of the walls of the flasks, and that of the fuel, act to prevent violent

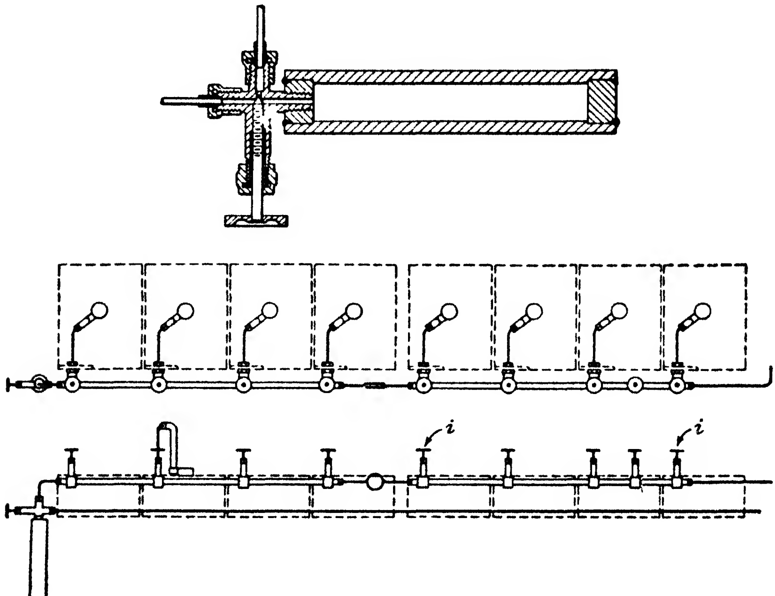


FIG. 43

fluctuations of the pressure due to the fuel pump deliveries and the nozzle discharges.

The pump may have one to four cylinders. Multiple cylinders are used to reduce the size of the operating mechanism. The pump capacity is necessarily greater than the engine consumption. A good design rule is to make it twice as much. The excess fuel is discharged back to the day tank through an adjustable pressure-relief valve. The plungers are socketed into the crossheads, Fig. 44, so as to permit self-alignment with the stuffing-boxes, which are packed with properly lubricated fibrous packing. They may be lapped and fitted to lapped bushings. In this case the plungers should be spring returned and not solidly connected with the operating mechanism to avoid wrecking in case a plunger should seize in its bushing. If the plunger is cam operated the cam roller should run on roller or needle bearings on account of the long duration of the pressure interval which makes lubrication of a plain pin difficult. The outer races of commercial anti-friction bearings are not stiff enough to be used as cam rollers. Many failures have resulted from using them in this way. Special roller bearing units should be designed for the purpose.

Pumps are made either en bloc with valves in cages, or as single units connected with brazed steel tubing suction and discharge

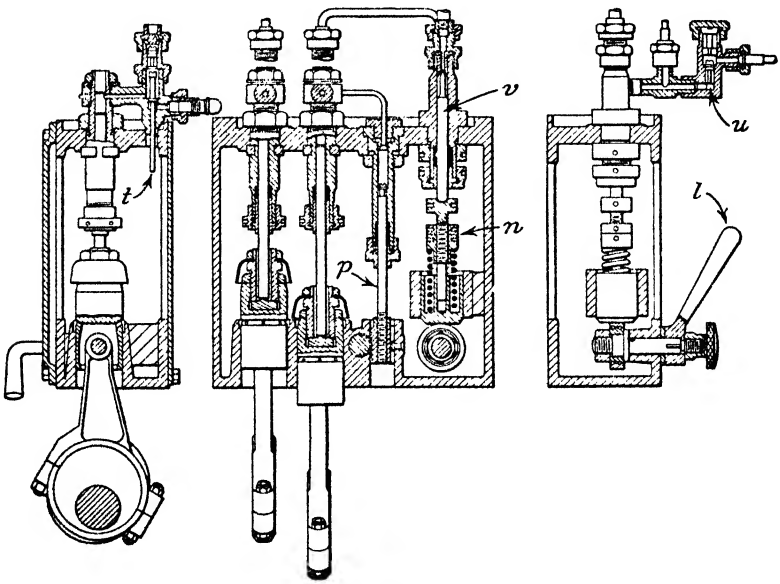


FIG. 44

manifolds having union connections, Fig. 44. These manifolds are concealed behind the gage board. The only unusual feature of the pumps is the projection of the suction valve stem or a tickler *t*, Fig. 44, contacting with it, to the outside. This is useful in two ways. If any cylinder of the pump is airbound, its admission valve may be lifted to release the air or to assist the entry of oil into the pump. A leaking discharge valve or the combination of a leaking discharge valve and air is detected by the great resistance of the suction valve against lifting by hand. A little auxiliary suction valve may be centered in the main valve to make lifting easier with the pressure on.

The pump priming plunger *p*, Fig. 44, has a seat on its inner end in the conventional way and requires no packing. It is hand operated. Large installations are provided with a separate motor-driven priming pump. This is particularly useful on ships operating in dense marine traffic or in locks. In such quarters there is little time to prime a pump that by accident has received a volume of air.

The pressure relief valve *v* is connected directly to a branch of the fuel pump discharge and not to the fuel manifold, which is cut off from the rest of the system by a good non-return check valve, *u*, Fig. 44. This enables the accumulator and manifold pressure to stand during stops even with a slightly leaking pressure-relief valve. It is quite common to find pressure standing on the accumulators after an overnight shut-down.

The tubing connecting the pump to the rail and the rail to the fuel valve is of the seamless drawn-steel variety with brazed union ends. After brazing the whole assembly should be heat-treated by quenching in water at 1550° F and drawing to 1000 to 1200° F. This, with tubing properly secured, will avoid many breakages in service due to vibration. For small high-speed engines of 9-in. bore or less, tubing 0.125 in. i.d. and 0.250 in. o.d. is satisfactory. For larger engines up to 19-in. bore at piston speeds up to 1200 ft, tubing of $\frac{13}{64}$-in. bore and $\frac{13}{32}$-in. o.d. is in common use. Tubing of too small capacity is not so troublesome with impulse pumps. It merely raises the pump pressure unduly above the spray pressure. With the Vickers system the

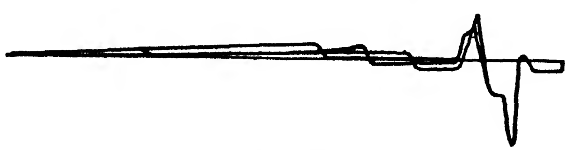


FIG. 45

reverse is true. The pressure drop upon opening the spray valve may be serious although a moderate drop is useful in preventing detonation. An indicator diagram, Fig. 45, shows a normal condition. This was taken at the fuel valve and shows the sudden drop upon opening of the valve, the establishment of flow pressure and the shut-off.

An isolating valve *i*, Fig. 43, is provided at each spray valve tap to shut off fuel from any cylinder at will.

The fuel pressure relief valve is a spring-loaded differential plunger as in Fig. 46. The pressure acts upon the annulus *a* formed by the plunger body and the seat, tending to eject the plunger through the stuffing-box against the set spring pressure. If it succeeds in lifting the plunger *p* owing to the excess supply

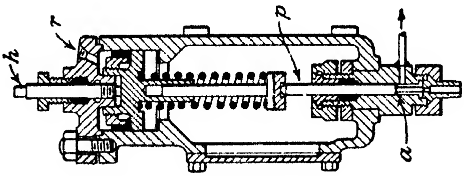


FIG. 46

of fuel, as it normally does, the fuel escapes through the small hole in the seat. The stuffing-box is packed with fibrous packing and is rather firmly set up, since it must act as a damper on the valve chatter that would otherwise be produced.

The discharge from the relief valve should not go back to the fuel pump suction, as is too often the case, but should be delivered to the day tank by its own pressure. If returned to the fuel pump, entrained air is not automatically eliminated, and heating takes place by reason of the wiredrawing of oil from a pressure of 4000 lb or more down to nearly atmospheric. The day tank dissipates this heat readily, but the fuel pump does not; and if the fuel is light and the heat considerable, the pump may be vaporbound.

The relief valve is provided with two adjustments, a permanent one on the spring spindle *n*, Fig. 44, and an operative one which is conveniently manipulated by lever, cam and quadrant or by a screw and handwheel as at *l*, Fig. 44, or by compressed air as at *r*, Fig. 46. Sometimes both air and hand control *h*, Fig. 46, are fitted.

The quick adjustment is useful mainly for starting and idling. If the starter turns the engine over at 60 rpm, the normal speed of operation being 300 rpm, the fuel valves are open five times as long as with normal speed and the charges for the first few revolutions will be five times as great as the full-load charge if delivered at full pressure. This method of starting results in detonation and, possibly, in the blowing of cylinder head or valve gaskets. Consequently, engines are started and maneuvered with a fuel pressure of 1000 to 2000 lb, more nearly the latter. If the normal pressure of full-load operation is 5000 lb per sq in., the fuel flow at 1000 lb is $100 \times \sqrt{\frac{1000 - 600}{5000 - 600}}$ or about 30 per cent of the normal rate. The item 600 represents the mean explosion pressure, higher at starting than in normal operation where 500 lb gage has been assumed to be the mean. The first charges will then exceed the full-load charge by about 50 per cent, perhaps too little for easy starting of a cold engine. At 2000 lb pressure and under the same conditions the ratio will be

$$5 \times \sqrt{\frac{2000 - 600}{5000 - 600}} = 2.82$$

which is ample.

Operators often adjust the fuel pressure to the load. The only advantage gained by this practise, and that a doubtful one, is relief of the pump mechanism. Careful tests show that best economy at all loads down to one-quarter rating is obtained with the full-load pressure prevailing.

Idling under low fuel pressure permits higher fuel valve lift which makes the engine less sensitive to small variations in fuel valve cams due to wear.

The fuel relief is provided with a tapered or pointed valve seat or with the flat seat as an alternative. The lead hammer is a popular tool for seating the former; the lap must be used for the latter. Failure is caused by grooving out the seat due to dirty fuel and wiredrawing. The use of a fuel centrifuge avoids much of this trouble. High-chromium steel for both valve and seat prolong the life greatly and justify the additional cost.

The air ram, Fig. 46, is packed with a leather cup and seats at the upper end of its travel against a leather washer. No attempt is made to graduate the fuel pressure by manipulation of the three-way cock that admits and releases the air. The ram is operated against stops giving starting or idling and running fuel pressures.

The relief valve housing is provided with a drain connected to the fuel sump or to the main underground tank.

IMPULSE PUMPS — THE PUMP POSITION

All attempts to equalize the deliveries of impulse pumps placed in a battery at one end of the engine have been rather unsatisfactory for large engines. Even when delivery pipes are made of equal length and the excess coiled, results do not seem to be uniform. It is quite impossible to operate well at all loads if each pump and nozzle system has an individual tubing length. For all

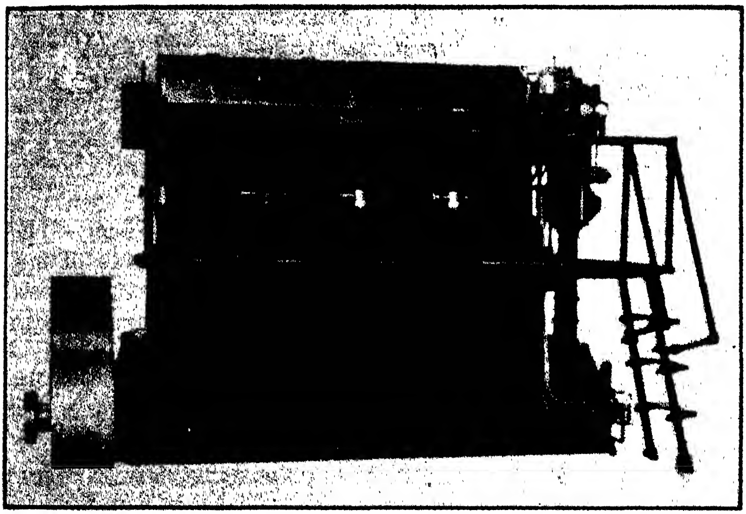


FIG. 47

except automotive types and sizes of engines the pumps should be built as single units, each placed in the same position relative to the cylinder it serves as in Fig. 47. For the high-speed engine it is quite desirable to build the pump into the nozzle assembly. A study of the known facts regarding velocity of stress through fluids and of water hammer in pipes* convinces one of the necessity of preserving exact similarity of conditions in each injection unit.

Fig. 48 describes a by-pass type pump belonging to class 2*a*.

Impulse pumps governing by suction valve control, popular for air injection engines, are used for solid injection. As usually built they operate by varying the angular duration of suction valve

* Merriman, "Treatise on Hydraulics," pages 248–412.

closure. This is done by a sliding or rotary suction valve travelling over the inlet port. The cut-off and by-pass edges are shaped to vary the timing with the load or speed as required by service conditions.

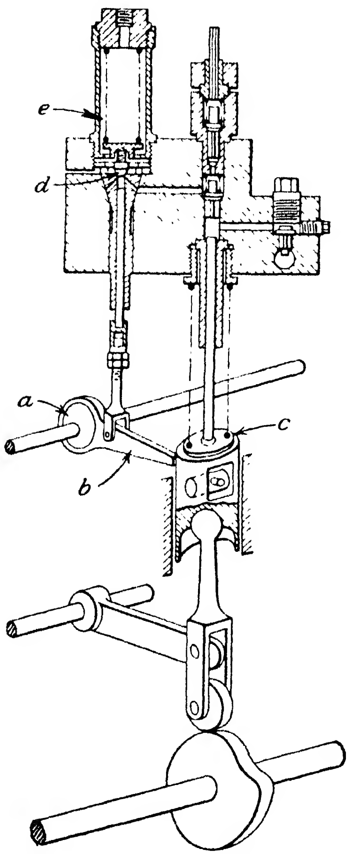


FIG. 48

Type 2*b* is no longer used. When regulation is effected by varying the clearance between cam and roller, the pump is noisy and the impact is destructive to mechanism.

Eccentric-operated "bumper" pumps also are noisy and have passed out of modern practise.

The inertia governor, operating pumps through an eccentric, once so much used, also has been abandoned. It is capable of marvelous regulation but tends to become noisy with wear. Neither is it well adapted to multicylinder engines although it is still used on some air injection types.

The system 2*c* requires delicate adjustment and care. It goes wrong easily if air is admitted with the fuel. The impact of the plunger against the fuel in the partially filled pump is responsible for much pump noise and detonation in the cylinders. Fuel inlet valves require delicate calibration for lifting pressure and natural frequency of beat to avoid great variation in the quantities of fuel admitted. The low-pressure piping system between the measuring pump and forcing pumps is very difficult to design on account of friction and kinetic influences.

Type 2*d* is used with the idea that a single pump as a single measuring device for all cylinders will deliver equal charges of fuel to all. Unfortunately the condition of check valves, variations in the action of individual fuel valves and necessary differences in the arrangement of the tubes leading to the various

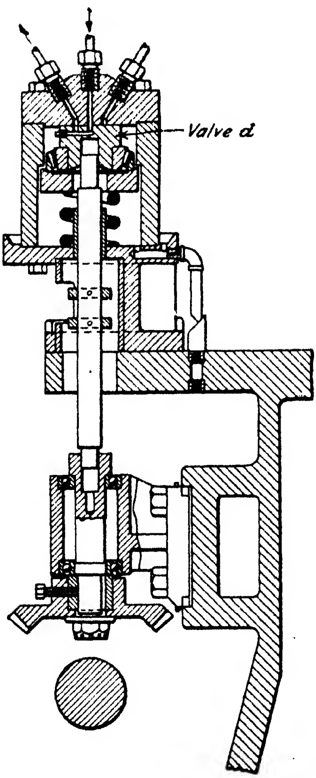


FIG. 49

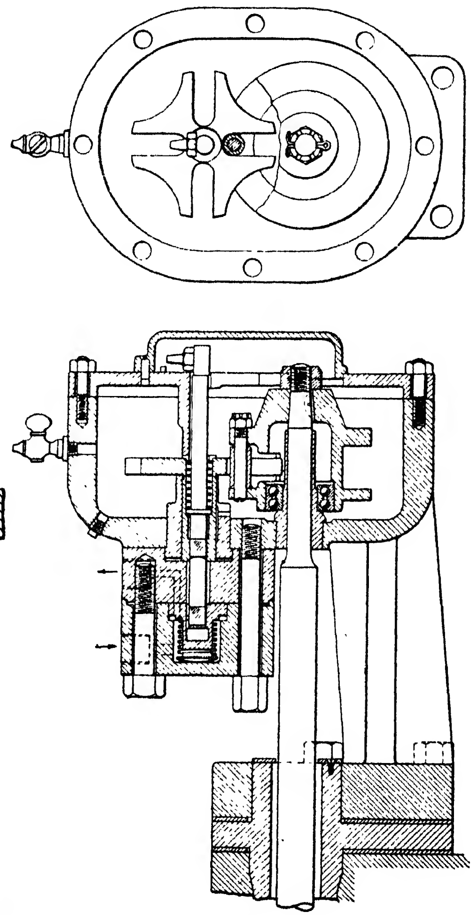


FIG. 50

nozzles introduce other errors of measurement for which there is ordinarily no means of compensation. When these errors are minimized by good design, however, a very practical mechanism results.

Three commercial forms of distributing valves are used when one or more impulse pumps is used for a multicylinder engine. The ported rotary valve, Fig. 49, is used with the Price engine. It comprises a disk *d* with a single through passage connecting at the proper intervals with each of a number of ports admitting fuel to the nozzles in sequence. The disk is held down by fuel pressure. To rotate the disk under an unbalanced pressure of 4000 lb per sq in. or more with such an indifferent lubricant as fuel oil is not very practicable. In an improved form, Fig. 50, the disk is operated through a Geneva stop motion. It is brought up to register with a port, and pressure is applied during a dwell. When the pressure is relieved through the by-pass at the termination of injection the motion carries the disk to the next port and another cylinder is served.

Hesselman uses a series of poppet valves operated to admit fuel to each nozzle in turn. Chorlton uses piston valves registering with drilled holes in their sleeves to establish communication between pump and nozzle.

The impulse-type injection pump differs from ordinary pumps in several essentials. The valves and in fact all working parts are of unusually refined workmanship and materials to insure reasonable life and ready restoration to the original condition. The liquid measured out by the pump movements is considered as a compressible medium* in calculating the volume displaced. Deflections of pump members too small to be notable in common pumping mechanisms are carefully regarded. The dilation of tubing and vessels in the system must be minimized. The compression of packing by the rising fuel pressure may modify the pump delivery, so the lapped plunger and bushing are used instead.

FUEL PUMP CAMS

As possibilities in cam design are limited it is well to deal with cam design first and derive pump proportions from it.

A constant-acceleration cam, Fig. 51, may be used giving acceleration to half lift and deceleration to the end of the stroke. Fuel is compressed for the first quarter stroke. Discharge utilizes all or part of the second quarter. The by-pass cuts out the remainder of the stroke as in Fig. 48.

* "Treatise on Hydraulics," by Merriman, 9th Ed., pages 9, 10 and 11.

For this cam the time of the first half lift will be related to that of the first quarter lift as

$$\sqrt{\frac{1}{0.5}} \quad \text{or as} \quad \frac{1}{0.7071}$$

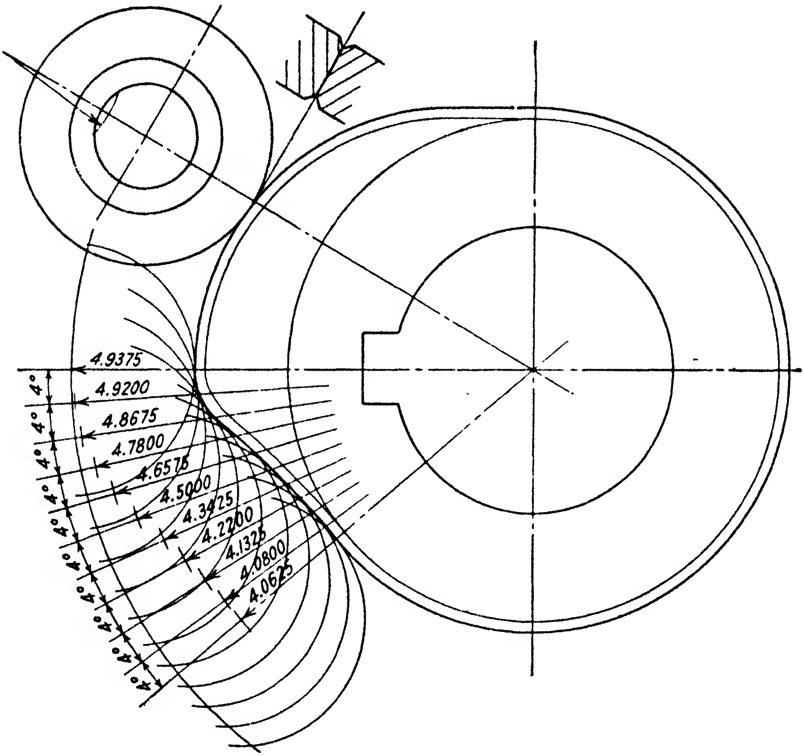


FIG. 51

The relative time duration of the first quarter to the second quarter is

$$\frac{0.7071}{1 - 0.7071} = \frac{2.41}{1}$$

Thus if the angle of this four-cycle cam profile devoted to injection equals 10° the entire upward cam movement of four times the full-load charge displacement requires 2(24.10° + 10°) = 68.20° of camshaft travel.

Such a cam is satisfactory with spring-loaded fuel valves which do not begin to discharge until full pressure is reached, particularly as a by-pass is used with the pump so that injection cannot be prolonged by elastic or kinetic discharge. With the open nozzle the cam is made quicker. Charge volume displacement may take place in about 6° of cam travel or 12° of crankshaft travel, the whole profile using 40° of cam travel.

It is rarely possible to design a cam for quicker injection than this although by reason of mechanical limitations injection is seldom too rapid. Prolonged injection is fatal to good economy and cool running as it tends to after-burning and high exhaust temperatures.

The parabolic profile of the constant-acceleration cam is developed by very simple means.* Thus if the lift is $\frac{3}{4}$ in. with acceleration to mid-stroke the profile is divided into ten equal parts (or any other convenient number divisible by 2) which are numbered consecutively. The division numbers represent t in the formula $s = \frac{1}{2}at^2$. The lift increments between each angular increment vary as the square of t. The height to mid-lift s_5 is divided by $5^2 = 25$. Then follows $\frac{s_5}{t^2} = \frac{s_5}{5^2} =$ constant k.

In this case $s_5 = 0.375$; $t^2 = 25$; $k = 0.375 \div 25 = 0.015$.
For all other positions $s_x = kt^2$.
Thus

TABLE VII

t	t^2	$s_x = kt^2$	ds
0	0	0	
1	1	0.015	0.015
2	4	0.060	0.045
3	9	0.135	0.075
4	16	0.240	0.105
5	25	0.375	0.135
6	....	0.510	
7	....	0.615	
8	....	0.690	
9	....	0.735	
10	....	0.750	

* See also "Valves and Valve Gears," Vol. 1, by Furman, pages 64, 65 and 66.

The second half of the profile is developed by adding the increments ds in the reverse order.

A finite value of t in seconds may be substituted from which both the acceleration and velocity of cam-driven members may be calculated.

If for instance the camshaft operates at 150 rpm, the four-cycle engine which it serves operating at 300 rpm, one camshaft revolution is completed in $60 \div 150 = 0.4$ sec.

Let the angular travel to mid-lift equal $5 \times 4 = 20°$. Then the time required to reach half-lift of 0.375 in. equals

$$\frac{20}{360} \times 0.4 = 0.0222 \text{ sec}$$

$$s = \tfrac{1}{2}\, at^2$$

$$0.375 = \tfrac{1}{2}\, a \times .0222^2$$

$$a = 1522 \text{ in. per sec}^2 = 127 \text{ ft per sec}^2$$

$$V = at$$

Thus

TABLE VIII

Position	Angle	t, sec	Velocity, ft per sec = $127\,t$
0	0	0	0.0
1	4	0.00444	0.565
2	8	0.00889	1.130
3	12	0.01333	1.695
3.5	14	0.01555	1.978
4	16	0.01777	2.260
5	20	0.02222	2.824

Compression of fuel takes up the first 14° of cam movement. Full-load charge delivery occurs between the 14° and 20° positions.

The mean plunger velocity between the 14° and 20° position is $\dfrac{2.824 + 1.978}{2} = 2.40$ ft per sec.

The fuel being sprayed under a 4000-lb head, its jet velocity is about 800 ft per sec as shown in the chapter on the spray valve. To maintain this head the plunger must move inward at a certain rate, relative to its diameter, or in other words the displacement of fuel at the pump must equal the discharge at the nozzle.

If the plunger is advanced at 1.5 times the theoretical rate required to maintain pressure on the jets the pump plunger diameter relative to the total spray orifice areas will be determined on the basis of $2.4 \div 1.5 = 1.6$ ft mean plunger velocity.

This excessive rate of plunger movement is provided to allow for pump wear and leakage, etc.

Thus the plunger area is related to the total area of tip drilling at the vena contracta as

$$\frac{800}{1.6} = 500 \text{ to } 1$$

Taking into account the orifice coefficient of 0.65, the relation between area of plunger and total area of tip drilling becomes

$$0.65 \times 500 = 325 \text{ to } 1$$

If the five jets of 0.018-in. diameter are used the plunger diameter will then be

$$\sqrt{325 \times 5 \times 0.018^2} = 0.73 \text{ in.}$$

The plunger would ordinarily be made $\frac{3}{4}$-in. diameter as the governor compensates for any little excess of capacity.

This pump will deliver a charge volume during the second quarter of its travel equal to

$$0.75^2 \times 0.7854 \times 0.1875 = 0.083 \text{ cu in.}$$

Its hourly delivery will be, at 300 engine rpm or 150 pump strokes per minute,

$$0.083 \times \frac{150}{231} \times 60 = 3.2 \text{ gal}$$

which will serve a 60-hp cylinder at that speed.

For other horsepowers the pump bore and stroke vary in direct proportion to the cylinder bore and stroke.

The greatest compressibility of petroleum fractions equals $70 \div 1{,}000{,}000$ per atmosphere. To overcome the mean pressure of 500 lb per sq in. during combustion the pump must deliver at 4500 lb or about 300 atmospheres. This results in a liquid compression of $300 \times 70 \div 1{,}000{,}000$ or 0.021 cu in. per cu in. of fuel contained in pump piping and spray at the time full pressure is reached. With a $\frac{3}{4}$-in. plunger having an area of

0.442 and a stroke of $\frac{3}{4}$ in., 0.1875 in. of which is reserved for compressibility, the volume of fuel in pump and nozzle at the beginning of the pump stroke may be

$$\frac{0.442 \times 0.1875}{0.021} = 3.9 \text{ cu in.}$$

This will be somewhat reduced by deflections in pump and members and by dilation of tubing and parts under fuel pressure.

Generally some ingenuity is required to bring the volume of fuel contained between pump plunger and nozzle orifice down to the theoretical limit so there is little danger of its being much too small.

Although the exact duration of the compression interval is not vital it is useful in easing the load on the pump and in confining delivery to the high-velocity stage of the plunger travel.

To derive from practise a final value for the ratio of excess rate of plunger movement to the theoretical rate and a correct clearance volume requires meticulous records of all volumes, pressures, deflections, etc.

One high-speed engine pump tested with great care gave the following values.

TABLE IX

Inches of stroke	Employment
0.102	Compression of fluid
0.022	Dilation of strainer and tubing
0.010	Deflection in pump
0.019	Displacement of indicator for pressure measurements
0.050	Delivery of fuel
0.203	Total
0.200	Actual stroke

Existing pump cams may be analyzed by graphical integration or by numerical differentiation.*

An example is given in Table X. This cam has a tangent acceleration profile and a small radius at the end of the tangent path for deceleration. It is used with a pump (type 2c) having no by-pass, the pressure of the fuel serving to force the plunger

* "Graphical and Mechanical Computation" by Lipka, Chapter IX.

to follow the cam contour during the rather harsh deceleration period. The plunger velocity, it will be seen, varies much more during the injection period than is desirable. Pumps (2c) are designed on the basis of mean velocity for full discharge, and

TABLE X

Camshaft angle, °		Roller displacement, in. = s	ds	$V = \frac{ds}{dt}$, in. per sec	V, ft per sec
Fuel Compression	0	0.0000			
			0.0027	1.12	0.09
	2	0.0027			
			0.0083	3.46	0 29
	4	0.0110			
			0.0138	5.75	0.48
	6	0.0248			
			0.0194	8.08	0.67
	8	0.0442			
			0.0252	10.50	0 87
	10	0.0694			
			0.0311	12.96	1.08
	12	0.1005			
			0 0373	15.54	1.30
	14	0.1378			
			0.0435	18.13	1.51
	16	0.1813			
			0.0503	20.96	1.75
	18	0.2316			
			0.0572	23.84	1.99
Maximum Fuel Discharge	20	0.2888			
			0.0497	20.71	1.73
	22	0.3385			
			0.0437	18.21	1.52
	24	0.3822			
			0.0306	12.75	1.06
	26	0.4128			
			0.0185	7.71	0.64
	28	0.4313			
			0.0062	2.58	0.22
	30	0.4375			

$$2^\circ = \frac{1}{180}\text{ rev.}$$

$$\frac{1}{\text{rev}} = \frac{60}{138}\text{ sec.}$$

$$\frac{1}{180} \times \frac{60}{138} = \frac{1}{414} = 0.0024\text{ sec} = dt.$$

conditions must be accepted as they are for fractional loads. The mean velocities should be estimated for three-quarters load and for half load to make certain that they do not fall below the theoretical requirement. At one-quarter load and friction load, discharge takes place at very low plunger velocity, but departure from the best conditions is not so harmful when only a small quantity of fuel is burned.

Cams with a shear instant drop are often proposed, but obviously the roller cannot follow a cam of that sort. In practise such cams are soon rounded by wear to some more natural contour.

It is distressing to analyze a cam by careful triangulation and differentiation only to find that it closely follows a constant-acceleration profile. Cam drawings should be dimensioned as shown in Fig. 51 and should not be approximated by arcs and tangents. The approximate profile contains the cumulative errors of draftsman's and tool-maker's layouts, one of which could have been avoided. The draftsman's error is considerable unless the work is done with vernier dividers on sheet zinc. If the shop requires such layouts from the drafting room, the work will be improved by blacking the zinc with dilute muriatic acid rubbed on until a satisfactory mat is obtained. Only metal tools should be used (square and protractor), as wooden, vulcanite or celluloid draftsman's apparatus is not dependable.

Pump cams or some driving member should be slotted so that the setting may be varied to suit the fuel. Usual practise is to set the cam so that a maximum cylinder pressure of 600 lb per sq in. results, although 750 lb or more may be required for good combustion at high engine speeds.

The diameter of the base of the cam may be fixed by rule, but it is better to develop back from the roller design. Roller pins of hardened steel will carry from 2000 to 4000 lb per sq in. if properly designed and finished. The pins should be dimensioned for 50 per cent overload to allow for over-pressure due to plugged spray tips and design allowance for cam wear and leakage which results in over-pressure in new or recently overhauled pumps.

Assuming, then, 6000 lb normal fuel pressure and 9000 lb maximum over-pressure, the largest pin ever required will be one worked at 2000 lb unit pressure. Its projected area will be $4\frac{1}{2}$ times the plunger area. If diameter and length of roller pin bearings are equal these dimensions will take the value

$$\sqrt{(\text{Diameter of plunger})^2 \times 0.7854 \times 4.5}$$

or

$$\text{Diameter of plunger} \times \sqrt{0.7854 \times 4.5}$$
$$= 1.88 \times \text{plunger diameter}$$

At 4000 lb unit pressure this factor is reduced to 1.33 × plunger diameter.

The former value is obviously impracticable except for slow and cumbersome engines. The latter results in a normal unit working pressure of $\frac{2}{3}$ of 4000 or 2667 lb, which is quite safe. It is at once apparent, however, that it is altogether desirable to avoid the higher fuel pressure.

The cam and roller should be machined with $\frac{1}{16}$- to $\frac{1}{8}$-in. 45° chamfers on each side as in Fig. 51 to prevent breaking down of edges should the roller loading concentrate because of slight lack of parallelism.

The cam roller should be three pin diameters in size, and the cam itself is best made two roller diameters at the base circle. Smaller rollers often fail to turn as the friction moment is too near the traction value. Smaller cams present too much roller interference preventing correct design of profile for the motion desirable. The flat follower has been used on very small pumps but cannot be favorably regarded. The unit loading of injection pump parts is so heavy that nothing but the best of construction should be used.

Various materials are used for roller and cam. If unit pressures are low they may be of chilled cast iron or S.A.E. 1020 V steel case-hardened. If the latter, double heat treatment and draw S.A.E. 1020 must be used or the case of the cam profile will part from the core and shell-off under the action of the roller. This is due to enfoliation.* For high speeds and duties it is best to make both cam and roller of high-carbon chromium steel. For pressures up to 3000 lb per sq in. the roller may be bronze bushed using phosphor bronze S.A.E. 65 preferably chilled. If the unit pressure 4000 lb is approached it is far better to run hardened steel against hardened steel. Both pin and roller bore should be lapped after grinding or seizure is almost certain to take place. Pins may be lapped on a Bethel-Player machine if available. If not, the cast-iron lap may be used. The lead lap and fff emery may be used for the roller. The pin should be at least 0.001 in. per in. of diameter smaller than the hole to accommodate the oil film. The clearance should not, however, exceed 0.0015 in. per in. of diameter when new. The pin should be grooved on the

* Steel and Its Heat Treatment" by Bullens, page 155, and Chapter VIII entire.

side away from the thrust as shown in Fig. 51. This is done by traversing the grinding wheel while the work is held stationary on the cylindrical grinder. A certain amount of oil should be supplied to roller and pin, preferably circulating. It may be introduced into a hollow pin; a small jet from the forced lubricating oil system may be played between rocker and roller with good results or the assembly may be immersed. Needle bearings are being applied to this service with excellent results.

Some means of relieving fuel pressure in case of complete stoppage of nozzle holes is always provided to prevent destruction of pump parts or piping. Earlier pumps were fitted with spring-loaded relief valves. Later engines use quite long and rather highly stressed studs in the flanges of the high-pressure piping. These studs are sufficiently elastic to allow the unions to part and relieve pressure above 9000 to 10,000 lb per sq in.

Several basic requirements may be laid down for valve design:

1. All valve seats should be replaceable.
2. The valve itself should be wing guided in the seat bore.
3. Valve seat and valve seat bore should be finished at one setting, as should the valve itself and its wing guide.
4. Valve and seat should be hardened.
5. Inner and outer diameters of valve and seat should register to prevent formation of shoulders by wear or grinding together.
6. The valve lifts should be limited by stops (usually to $\frac{1}{32}$ in.).

Early pumps had the valve seats formed in the pump body. They were difficult to machine and could not be hardened. During regrinding much abrasive escaped into the pump, being difficult to remove. Inspection was possible only with artificial light or light and mirror.

Many of these valves were guided in the valve cap. It is possible by tool-room methods to produce accurately centered valve guides and seats in which the two are in separate parts screwed together, but the process is expensive, requires highly skilled men and can be avoided by proper design.

The wing guide has replaced all other forms. It may be milled from solid metal or merely drop-forged. Four wings should be used, as three-wing valves are difficult to gage. The hollow cylindrical guide once used because it is easily made on a screw machine has lost favor, for it is so often stuck by a particle of foreign matter between the guide portion and the seat bore.

The ball valve has been abandoned. Its natural frequency of movement is low as it does not lend itself readily to spring loading. It is not easily guided, and if not well guided it batters the seats badly.

For finishing valve seat and valve seat bore at one setting the Rivett grinder with the swivel chucking head is ideal for either flat seats or angle seats. For a like operation on valves the Bath grinder with centers and ways swinging about the column which supports the grinding head is an excellent tool. The valves should be driven by a driver on center or face plate in such a way that the valve need not be turned end for end for the two operations. Even work produced on dead centers is rarely concentric if reset between operations.

S.A.E. 1020 steel case-hardened is the most common material for valves and seats, particularly if the valves are drop forgings. On account of the thin sections, carburizing is rarely prolonged over one or two hours and double heat treatment S.A.E. 1020 V must be used. Crucible steel of 0.65 to 0.90 carbon may be used to good advantage and hardened by water quench, then drawn to about 600°F. Brittleness must be avoided in the treatment of either material, but the valves and seats should not be less hard than Rockwell C 55 to 60.

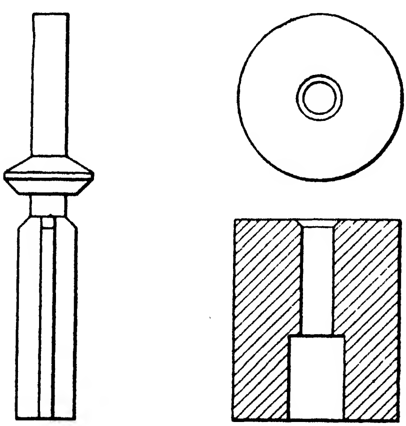


FIG. 52

Examples of valve and seat are shown in Fig. 52.

Commercial valves of good quality are now on the market as supplies.

Valve seats are commonly compressed into the pump body against flat copper gaskets. It is better to lap the sealing faces and set them up metal to metal for pressures 4000 lb and above.

The pump body is usually a solid steel forging or drop forging. Foundry losses in bronze and cast-iron bodies are heavy, and such bodies often prove defective only after a period of operation has loosened inclusions or broken down strained sections. S.A.E. 1035 steel is a satisfactory material for bodies. The softer steels are readily "galled," that is, threads of screwed parts weld together. The material should be forged, as sections of rolled bars and blooms are often surprisingly porous.

Plungers are made of the same materials as valves and are hardened, ground and lapped. The plunger bushings are of cast iron or cast phosphor bronze S.A.E. 65 or hardened steel and are ground and lapped. Bushings are good for two or more years of service if well made and particularly if the fuel oil is centrifuged. The plungers must be well guided preferably by a crosshead with a floating connection to the plunger as in Fig. 48. The bushing must be relieved of any side thrust.

Since the advent of high-pressure solid injection the by-pass valve used for governing is of the "balanced" type, Fig. 48. The stem is fitted into a hardened and lapped bushing. The seat is made very narrow. A wide seat brings about need of an unduly heavy seating spring by reason of the leakage effect described by Reuleaux (see Chapter IV) and taxes the governor. The "balanced" valve is seated by a 50-lb spring and is quite durable if made of good materials, preferably crucible steel of 0.65 to 0.90 carbon. It greatly lightens the reaction on the governing mechanism.

In applying lapped bushings special attention must be given to the elastic effects due to the compression of follower nuts and

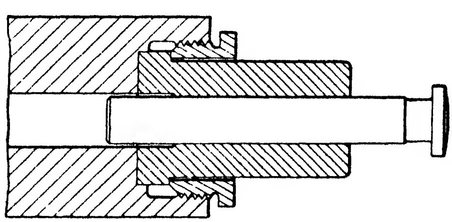


FIG. 53

the like. It would seem quite improbable that the bore on Fig. 53 could be appreciably distorted by the compression of the nut $\frac{3}{8}$ in. or more radially distant, yet it bulges inwardly enough to seize the plunger if the lapped portion of the bore is carried through. For this reason the bushing is counterbored at the upper end. Fuel pumps provide many of these interesting exhibits of the direction of elastic movement of materials under stress. The gage-like closeness of the fits and the high pressures employed exaggerate all such phenomena. To reduce the mechanical loads all openings into the pump body should have the least possible diameter and all gaskets or ground seats should be narrow.

COMMERCIAL FUEL PUMPS

Pump units either single or multiple are now accessory manufactures, and few engine builders design or make their own pumps and nozzles. This, however, does not relieve the engineer from responsibility for careful application of purchased units for his engine. The analysis outlined in this chapter should be applied to each adaptation.

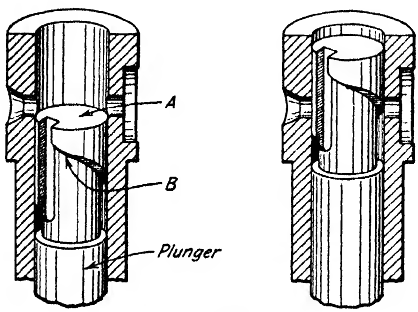


Fig. 54

For small diesels the poppet-valved pump with poppet by-pass is difficult to design as large clearances are inevitable. This brings into prominence ported pumps like the Bosch, Fig. 54, in which the suction port is cut off by the end of the plunger *A* and variable by-pass is effected by the spiral edge *B* as it re-opens the suction ports. The plunger is rotable for governing.

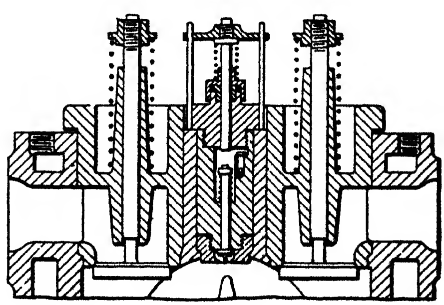


Fig. 55

Improved materials and precision grinding and lapping have made these pumps quite practical when protected by the best fuel filters available.

A further aid has been the very desirable reduction of fuel pressure to 1500 or 2000 lb, made possible by use of whirl chambers and the pintle nozzle.

The unit injector, in which pump and nozzle are in tandem in a single body, also is beginning to appear. This idea was shown schematically in the Weidmann patent D.R.P. 175,932, July 8, 1905, as in Fig. 55.

CHAPTER VI

GOVERNING

The governor may be considered as a tachometer. In its active or useful range it assumes a certain definite position for each engine speed. The lever that transmits the movements of the revolving weights, like the tachometer hand, indicates the engine speed. The controlling device which acts on the fuel supply is attached to this "hand" in such fashion that as over-speed or under-speed is indicated fueling is corrected by the same movement.

All governors possess some degree of internal friction and if driven at absolutely uniform velocity have two equilibrium positions for that speed. In moving toward any true equilibrium position the governor comes to rest before that position is reached or when friction and centripetal force balance the centrifugal force.

It would seem desirable to make governors quite frictionless if this were possible. This is not quite true since prime movers of the crank motion type do not turn at constant velocity. Their crank rotation is distinguished by a cyclical irregularity, a "lope" limited in degree by the flywheel. This cyclical change of angular velocity is transmitted to the governor through its drive. The result is, in a well-chosen governor application, a very slight and continuous oscillation of the governor, the pulsations of force being a little greater than necessary to overcome the internal friction of the governor at each power impulse, leaving it periodically free of friction to assume a new mean position corresponding to the mean engine speed.

This may be considered from the opposite viewpoint. Internal-combustion engines, particularly those with few cylinders and light flywheels, have a very irregular turning effort; consequently it is not always possible to fit them with the highly sensitive knife-edge governors used with turbines. A governor of this type would be thrown into violent cyclical movement by the motion of the engine. It is common to fit rather insensitive governors trusting to their internal friction to limit the governor oscillation.

This is well enough if the amount and quality of the internal friction are carefully considered, but friction is by no means a panacea for governor troubles. Unfortunately, the friction of rest is greater than that of motion, and a governor damped in this fashion moves when called upon to do so as though projected by released force instead of following the speed changes with just enough lag to make the required force available. An abnormal speed change is required to overcome the friction of rest. Once this is exceeded the governor mechanism springs toward a false position called for by the excessive speed change and overtravels or "over-regulates" as it is termed. Unless the load is a very stable one sustained hunting may be set up.

A better expedient is to choose a governor of a sensitive type moderating the torque impulses by interposing a flexible driving member between the driving shaft and governor spindle.

Weight-loaded governors are rarely used. The dead loading on the pivots contributes to friction and thus to insensitiveness. The inertia of the mass makes the governor more sluggish than usually desired. The upward rate of movement is slower than the downward rate. In ships, on shovels, dredges, locomotives or even on foundations, the weighted governor acts as a seismograph. The weight tends to maintain its position in space while the engine and its carrier vibrate about it. This causes "indications" of the governor lever that are due to the movement of the engine as a whole rather than to any speed change. Spring-loaded governors of the plain centrifugal type are preferable.

If low internal friction is a requirement the method of transmitting the centrifugal force of the weights to the centripetal spring is of first importance. The usual application is that shown in Fig. 56, in which the centrifugal force is resisted directly by the spring. This relieves the pivots of all unnecessary loading. With this design it is of course impossible to change the main spring tension while running. When such adjustment is required a stationary spring may be used if applied as in Fig. 57. Governors of this type do not suffer from lost motion as all pivots are held in contact. The spring tension should be carried through a ball-bearing collar to avoid loading the splines in the sleeve by an excessive collar friction torque. The links are long in order that the link pin friction moment may be of the least possible consequence. The pins themselves are not closely

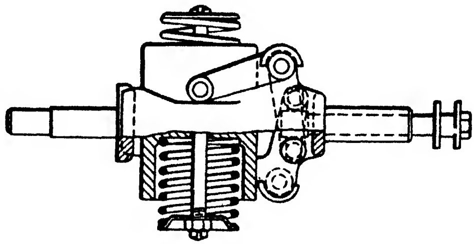


FIG. 56

fitted to the links, Fig. 58, but actually are about $\frac{1}{16}$ in. in diameter smaller than the link eyes to provide rolling contact. Where the governor oscillates continually this is preferable to knife-edge

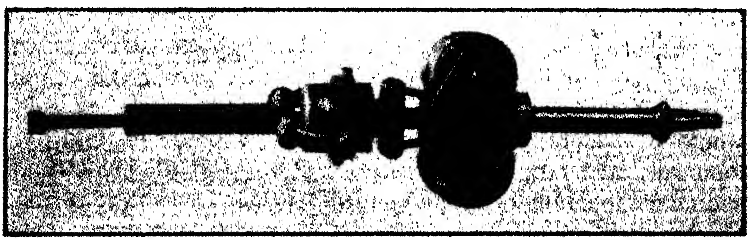


FIG. 57

joints. It provides a good wearing contact with very little friction. The governor weights should be pivoted on ball or roller bearings, Fig. 59.

Friction usually is developed in the running condition of the governor. At the fitter's bench a governor may appear to be

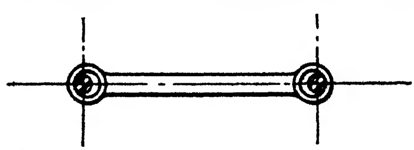


FIG. 58

perfectly free when moved by hand yet it may give evidence of sticking when in use even though the spring tension is applied directly to the revolving masses and theoretically there is no pivot loading. A common cause is the inertia of the weights acting tangent to their orbit. When the velocity changes they

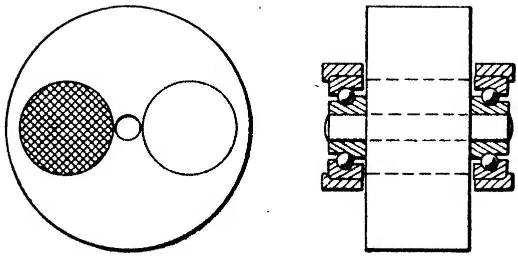


FIG. 59

do not hang vertically on their suspensions but tend to lead or lag. This causes binding on the weight suspension and on the link pins. Ball-bearing mounting of the weights obviates the trouble entirely.

Unbalance also causes internal friction. If the structure in Fig. 60 were a rigid one, that is, if the sleeve A were forced on the spindle B, and the pintles C, C' and C'' were driven home, it would be possible to balance the assembly statically and dynamically about its axis of rotation XX. But all pivots and sliding fits in governors require at least enough freedom to permit the building up of the oil film of minimum resistance. This freedom is increased by the necessary tolerances of manufacturing and by wear so that the statically or dynamically balanced device is free to assume one of two unbalanced positions, the weights running in different orbits as shown much exaggerated in the sketch. One or the other of these positions will be sustained in operation.

This results in a greater centrifugal force being produced by one weight P than the other P'. The springs F exert equal centripetal force on both weights. Therefore the weight having the major orbit, finding its centripetal force insufficient, tends to lift the sleeve while the excessive centripetal force acting on the other weight tends to reduce its orbit and to depress the sleeve.

Under the influence of this couple the sleeve cocks and carries the unbalanced load in tension between the bearing points assumed to be at the edges b and b'. If f or f' represents the unbalanced load at either side, p or p' may represent its horizontal component and pp' multiplied by the coefficient of friction measures the effort required to move the governor from its equilibrium

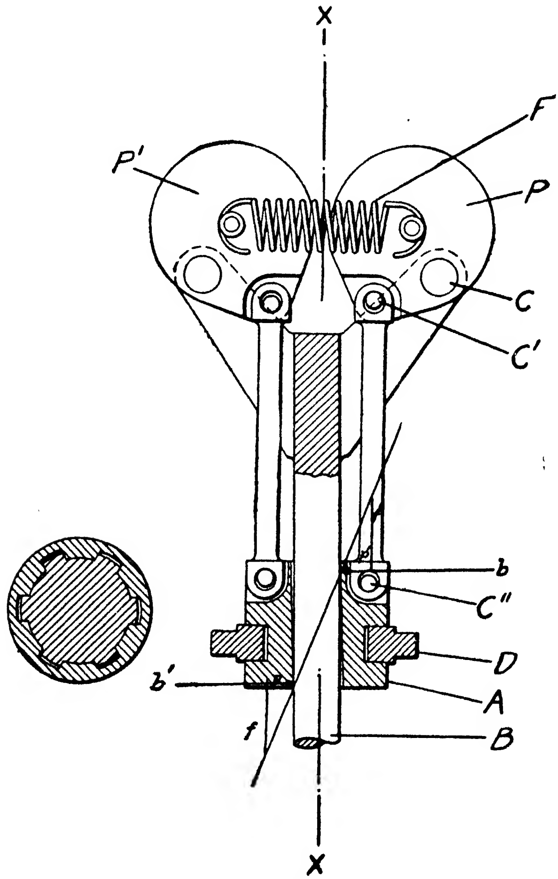


FIG. 60

position when a change of speed makes it necessary for the governor to function.

If the spindle is large and the sleeve relatively short, this clamping action will exert a ruinous influence on governing action.

It is desirable to keep the sleeve short, and a stiff spindle also is a requirement, but splining the shaft and broaching the sleeve as shown in the section, Fig. 60, offers a satisfactory solution. In this case the width of the splines relative to the sleeve length rather than the spindle diameter and sleeve length determines the friction to be overcome before the governor can function.

The splined sleeve has improved governors having every other refinement known to the art including anti-friction bearings on all pivots and spring loading so arranged as to impose no reactions on the mechanism.

It is quite usual to fit a single key spline to the governor sleeve to prevent twisting of the linkage under the drag of collar *D*. This, however, does not suffice, multiple splines being effective in all directions as is necessary if the sleeve is to slide with minimum friction under tipping loads.

Governors that derive their energy through high speed rather than large effective mass usually are too quick in action. This may be offset by introducing inactive weight into the system, that is, extra weight that must be moved by the governor when a change of position is required.

In Figs. 57 and 59 this is effected by using two weights mounted at their center of gravity in place of the usual weights mounted on lever arms. The active mass is introduced by loading one side of the weights with lead. This method is more desirable than weight added in the form of a counterweight as it is free from gravity influences and it avoids loading the collar and linkage with dead weight or forces due to inertia.

The counterweight is sometimes used as in Fig. 61. In this case it serves the two functions of increasing stability and counterbalancing the dead weight of the sleeve. This construction is satisfactory for stationary plants, but the seismographic action of the weight makes it undesirable if the engine is in a floating or rolling plant.

Gravity as it affects the equilibrium of the moving parts of the governor must be considered. In the horizontal position the governor weights have a gravity moment sum of zero. In the vertical position the action of gravity on the weights may result in a positive moment (acting with centrifugal force) or a negative one or it may be mixed. It is desirable that the gravity moment if present should increase as the weights move outward. This tends to offset the undesirable tendency of governors toward isochronism in the outer part of their range. This quality is due to the variation of centrifugal force with the square of the speed while the spring force is a straight-line function of the radial position of the weight. The gravity effect of the sleeve is constant if the governor is worked in the vertical position and zero for the horizontal position.

Many kinds of flexible drives have been used for governors. The leather belt is the oldest of them but it is now little used.

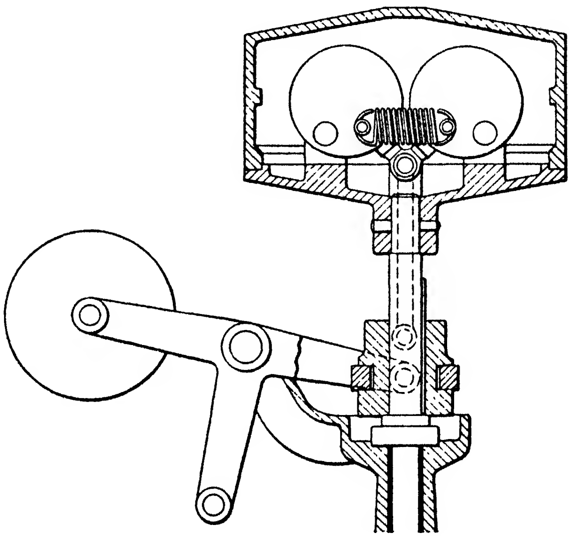


FIG. 61

A jockey pulley or sprocket on the tight side of a silent chain drive is excellent. The pulley rises and falls with the engine impulse, causing the chain to transmit a comparatively even torque to the governor sprocket.

After years of experiment with the known forms of spring-actuated drives all of which were unsatisfactory because of internal friction and occasional cases of resonance a most satisfactory solution has appeared. It is the funicular coupling shown in Fig. 62. One spider floats on the governor shaft and is attached to driving gear or chain sprocket. The other arm is keyed on the governor shaft. The ends of the two spiders are loosely linked together with common roller or block chains. When the velocity of the driver is constant the chains, under their own centrifugal force, assume the form of a circle. With velocity changes one pair of chains arcs out and the other or opposing set tends to straighten thus providing for a change of angular velocity between the two members and a restoring force due to the difference in the toggle advantage of the two sets of chains. As the increments of force vary at different rates for outward and inward displacement of the chains from their circular orbit, harmonic vibration of the governor as a whole on its

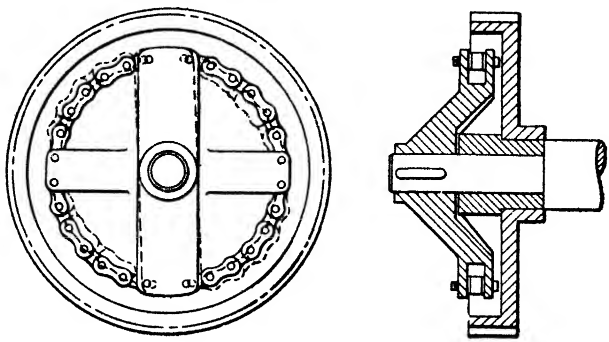


FIG. 62

drive is not possible. This latter is a dangerous reality with spring drives. Thus far no mathematical solution of the chain weight has appeared to be necessary. Half inch pitch chain is in common use, but to liven up a sluggish governor a heavier chain may be used. The device is not patented.

Dashpots are rarely used. For steady loads they may be satisfactory, but when full load is thrown on or dumped by the opening or closing of a switch the dashpot invariably causes trouble by reason of its great resistance. Powerful inertia governors are an exception to the rule as a great additional force is called into effect when a sudden change of flywheel velocity takes place.

The centrifugal governor at best is a weak instrument. A dashpot may be set to damp out small oscillations. If then a sudden movement of wide range is called for, the resistance is too great since the flow of oil through the dashpot orifice varies only as the square root of the head upon it. Relief valves may be fitted to the dashpot, but their effect on regulation leaves much to be desired. When they are opened by over-pressure due to the demand upon the governor it usually over-regulates. This calls for a sudden reflex movement that is not possible until the relief valves are again opened with the result that the whole performance is disgraceful compared to that of a governor with inherent stability.

THE CONTROL DEVICE

The common-rail type of engine is controlled by a wedge, Figs. 37 and 38, introduced between the fuel cam follower and the fuel valve lifter. In the standing position the wedge is in, taking up all lost motion but the small amount required for clearance to insure seating of the valve. The governor should be capable of withdrawing the wedge so far that the fuel valve receives no motion.

Engines provided with individual cam-operated impulse pumps are governed through a by-pass which is kicked open at some point in the pump stroke determined by the governor. This is illustrated in Fig. 48. The governor is connected to the eccentric shaft *a* which acts as a floating fulcrum for the lever *b*. The other end of this lever is attached to the pump plunger actuator *c*. The governor sets *a* in the required position to effect cut-off of fuel and discharge back to the supply rather than to the fuel nozzle. The by-pass valve *d* may be balanced as in the M.A.N. system in which the valve seat is quite narrow, the poppet being held down by the spring, Fig. 48*e*. This relieves the governor and linkage of the rather heavy blow to which it would react and cut off less sharply. The eccentric shaft is used because of two excellent properties. The friction moment of its larger diameter stabilizes the mechanism during the impact of cut-off. The declining rate of the rise of the fulcrum point due to its arcuate travel may be used to advantage to offset the decreasing stability of the governor as it approaches the friction load position.

The total possible movement of the governor should be sufficient to cut out completely the pump or fuel valve acted upon. On the other hand, when the required control movement is effected by a small part of the governor stroke the accuracy of regulation may be seriously affected. For example, if the linkage of a worn governor and control apparatus operates with $\frac{1}{16}$ in. total lost motion and the governor controls with $\frac{1}{4}$ in. of motion at least 25 per cent of the required total speed variation may be used in idle movements through the lost motion range. The same is true of the false positive and negative positions due to governor friction which may bear a needlessly exaggerated relation to the total speed change. In either case, units in parallel operation may make a decidedly bad showing on the switchboard on account of interchange of current within the limits permitted by the governor.

The governor sleeve movement should not be less than 1 in. for small engines and it may well be made 2 in. or more for larger engines, all of it except a moderate amount of "safety travel" being used for complete regulation, which means provision of a double supply of fuel for starting and complete shut-off of fuel under over-speed. The safety travel is reserved for slight misadjustment of linkage which might cause an engine to run away or at the other extreme to starve for fuel at full load. In the case of cam-operated pumps the governor travel is utilized to cut out the excess pump displacement provided for fuel compressibility and for the overcharge used for starting so the part of its stroke useful in regulation during normal operation will be relatively less than that available for common-rail engines.

The remarks regarding friction apply as much to the control mechanism as to the governor. All shafts should be mounted in anti-friction bearings; and where control shafts are divided into short units for each cylinder, connection into one continuous shaft should be through flexible disk couplings of the Thomas type which do not acquire lost motion.

The governor power required is difficult to estimate as the control mechanism is almost frictionless and the same governor is used for a whole series of engines comprising many cylinder sizes and their multiples. The necessary compromise is quite satisfactory. Most diesel engine governors have a mean centrifugal force of from 50 to 150 lb. A mean force of 100 lb will adapt itself to almost any situation with a proper drive and control gear. This force may be reached by mass, speed or radius in any combination. The heavier governors usually operate at engine speed, the lighter ones at about twice engine speed. Governors operated at camshaft speed are rarely lively enough to do well under all conditions. The centrifugal masses vary from $2\frac{1}{2}$ to 6 or 8 lb, the orbit being as small as the construction will permit. These figures apply to ball-bearing-mounted governors operating ball or roller bearing control mechanisms having little or no reaction upon the governor.

PARALLEL OPERATION

A fairly wide speed variation, as much as $2\frac{1}{2}$ per cent from the normal speed, is essential to good operation of a-c generators in parallel. With a total speed variation of 5 per cent from full load to friction load and a long-stroke governor the control mechanism positions for each power output are quite definite and slight faults in governor condition or functioning do not effect great changes in fueling.

Some form of synchronizer is required to bring incoming engines to the speed of loaded engines and to keep frequency to the standard number of cycles per second.

This change of governor adjustment may be effected by varying the tension on an auxiliary spring by hand or by means of a

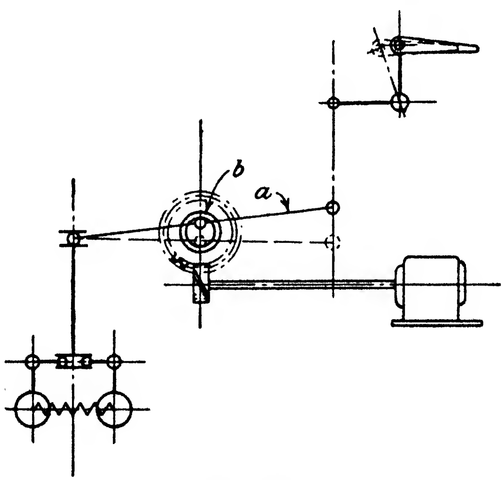


FIG. 63

motor working through a small speed reducer upon the spring adjustment.

The auxiliary spring has the disadvantage that operators tend to neglect main spring adjustment, depending upon the auxiliary springs to compensate for excessive or deficient tension. Consequently a battery of engines may be running at a given frequency, some being under control of the main governor spring only and others using in addition some synchronizer spring tension through all or part of the governor range. This results in different load-speed characteristics for the various units, a condition altogether undesirable if the total load fluctuates widely. The general result is that some one unit does most of the regulating for the plant, the others tending to vary less in their output with the demand on the station.

A more satisfactory form of synchronizer is that shown in Fig. 63. A floating lever *a* is carried on an eccentric fulcrum *b*. This eccentric shaft is rotable by worm gear and reversible motor so as to change the relative positions of governor collar and wedge or control device. Thus at any synchronous speed the control apparatus may be adjusted independently of the governor varying the fuel input which, however, still remains under governor control.

THE GOVERNOR SPRING

The spring calculation for governors like Fig. 56 is very simple.

First estimate the radii of the inner and outer weight orbits corresponding to the full-load and friction load fuel requirements.

One weight or fly ball only is considered as the other furnishes the reaction.

Let

Radius inner orbit	$= r_1$ ft
Radius outer orbit	$= r_o$ ft
Force at inner position	$= F_{ci}$ lb
Force at outer position	$= F_{co}$ lb
Weight	$= W$ lb
Speed of governor at inner radius	$= N_i$ rpm
Speed of governor at outer radius	$= N_o$ rpm

Then

$$F_{co} = 0.00034\, W r_o N_o^2$$
$$F_{ci} = 0.00034\, W r_i N_i^2$$

The required spring scale equals

$$\frac{F_{co} - F_{ci}}{12(r_o - r_i)}$$

or simply the difference in centrifugal force at the two positions divided by the difference in radius (the spring compression) in inches.

COMPENSATING SPRING SUSPENSIONS

Governors like Fig. 56 grow less stable as they approach the outer equilibrium position.

A governor head made like Fig. 61 may be so designed as to give a straight-line speed-lift curve since the spring suspensions rise and increase the spring moment as well as its extension as the governor lifts. This is done at the expense of pivot loading which again may be offset by using anti-friction pivots.

Although anti-friction bearings are mentioned frequently in this chapter as being essential to good regulation, their life is short on account of the oscillating motion of the pivots. Bearings of ample capacity should be used preferably with rollers in support rather than balls.

More extensive use of pumps using ported intake and plunger cut-off like Fig. 54 has greatly increased the friction work on governors. In regulating, all the fuel pump plungers are rotated by means of splined gears on the plunger shanks and racks connected to the governors. When the number of cylinders is great as in recent developments using 12 and 16 cylinders the friction is high. Hydraulic relay governors of the type used for water-wheel governing are now available in compact form for this work. Governor heads designed on the principles laid down in this chapter control the valves of a servo motor which does the work of positioning the pumps.

These governing units are capable of governing with little or no speed variation, a property of less value than their ability to maintain speed to a close degree on any fixed load thus avoiding interchange of current between parallel units. Some load-speed variation is essential to successful parallel operation, and since frequencies must in any event be corrected by operation of the hand or automatic synchronizer the variation might as well be sufficient.

CHAPTER VII

CYLINDER HEAD, VALVES AND VALVE GEAR, STARTING

Inlet and exhaust valves for four-cycle engines are made to the same dimensions. Valve capacities are compared on the basis of "mean gas velocity" through the valve. This value is determined thus:

$$\frac{\text{Piston area}}{\text{Valve area}} \times \text{mean piston velocity} = \text{mean gas velocity}$$

The "mean piston velocity" equals twice the stroke multiplied by the revolutions per minute and is expressed in feet per minute.

The valve area is taken at full lift, and pressure difference is ignored, so the expression lacks mathematical accuracy but its use is quite as satisfactory — or more so — as the involved method of plotting piston velocities and corresponding valve areas to derive a curve of gas velocities. The latter is never realized in practise owing to pressure changes and particularly to the inertia of the gas columns in intake pipe and exhaust pipe.

Much testing has failed to discover any advantage in reducing mean gas velocity below 9000 ft per min. A decline in efficiency and capacity sets in when the velocity much exceeds 12,000 ft per min.

Cylinder head ports should be enlarged outside the valve throat, the area being increased 50 per cent if practicable without introducing casting difficulties.

At higher speeds large valve areas are needed and are acquired best by the use of dual inlet and exhaust valves. The smaller dual valves offer many advantages. Except in the case of very large engines, water cooling of the poppet is avoided. The valves may be so far separated that the center section of the head is stronger and better cooled, Fig. 64. It is not necessary to mill valve clearance in the cylinder liners.

The exhaust valve poppet is made of a heat-resisting material. Many of these are available. Diesel engine exhaust temperatures are lower than those of the gas engine, so materials do not show such marked differences in performance. These metals are

FIG. 64

mostly chrome nickel irons or steels sometimes alloyed with silicon, tungsten, copper, manganese, etc.

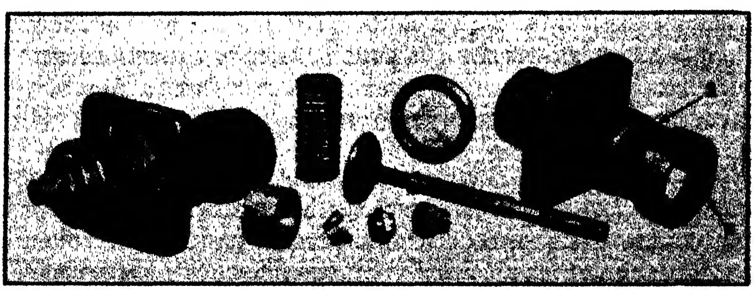


FIG. 65

The inlet valve usually is made of one of the forging varieties of chrome nickel steel.

If the material selected for the valves cannot be hardened, hard-steel nuts, caps or lock heads may be used as in Fig. 65.

Valve seat inserts of the same heat-resisting materials as are employed for exhaust valves are being used extensively. They are pressed or shrunk into the heads. If they are to be reliable a high degree of skill in design and manufacture is essential.

When valves operate in cages, as they should whenever the engine dimensions are such that cages can be incorporated in the cylinder head (about 12-in. bore and up), the valve seats should be separate as in Fig. 65. These seats usually are of cast iron, sometimes of cast aluminum bronze.

Copper	89 per cent
Aluminum	10 per cent
Iron	1 per cent

The exhaust valve cage should be water-jacketed, carrying the jacket as far down as possible. The struts *s*, Fig. 65, which support the seat flange are to be calculated for a compressive stress not to exceed 4000 lb per sq in. The seat flange *f* itself should be calculated as a continuous beam carrying a uniform load equal to 50 per cent more than the explosion pressure against the valve and seat. Its stress should not exceed 4000 lb per sq in. The greatest skill and care are required to compromise stress, jacketing and gas flow area through these cages. If the job is well done failure is rare.

The bending stress of 4000 lb may seem rather high, but the loading is static being set up by the studs and the assumption of a uniformily loaded beam is favorable. Really there is concentration of load at the supports. Steel mix having a tensile strength of 30,000 lb per sq in., or more, should be used for the cages.

Valve cages of unsymmetrical cross-section, particularly those having one side outlet instead of the hour-glass form shown in Fig. 65, are troublesome. When bolted in position they often deform causing the valve to seat on one side only. Valve cages having much uncooled length exposed to exhaust gas often buckle owing to expansion and break their lugs or studs.

The gasket between seat and head should be of solid copper or folded copper. A rubber ring may be used in a chamfer around the top to prevent the blowing of gases into the engine room. Cages should be coated with plumbago when installed to facilitate removal. Cage studs should be as long as possible, the stud body being turned down to the thread root diameter. This allows for expansion of the lower part of the cage without over-stressing other parts, the studs being able to stretch appreciably without great increase of tension.

VALVE TIMING

Extended research into valve timing reveals no notable difference in performance if a few basic principles are observed. The exhaust valve should close at an interval so far removed from the top dead center that a little wear, expansion or workmanship tolerance will not cause it to close before the exhaust stroke is completed. Actually it may close not less than 10° after dead center and need not close later than 20° "late." The opening time of the exhaust valve often is governed by starting conditions. A six-cylinder four-cycle engine of the direct reversible type must start from any crank position. The minimum duration of starting air admission is then $2 \times 360 \div 6 = 120°$. It is desirable to prolong this somewhat to allow for wear of the starting-valve mechanism and consequent delay. It may not be prolonged to such an extent, however, that both starting valve and exhaust valve are open at one time. This would result in needless loss of stored air. Stationary engines are built with the exhaust valves opening as much as 60° before outer dead center. This is obviously impossible for the reversing marine engine as 120° of starter action and 60° of exhaust valve lead account for the whole stroke with no interval to allow for overlap of starting impulses and wear. If valves and passages are large enough it is quite satisfactory to open the exhaust valves 30° before the outer dead center, thus allowing the starter action to be prolonged 10 or 20° during which two starters may act in unison, this overlap being very desirable.

For stationary engines it is customary to open the inlet valve 10 to 20° before upper or ~~outer~~ inner dead center. Thus both inlet and exhaust valve are open at once and with a carefully designed exhaust system a scavenging draft may be induced through the clearance. Repeated tests have shown this action to be economical, but where the length of the exhaust pipe cannot be controlled or where the exhaust must be muffled as in many classes of ships, it is perhaps advisable to open the inlet very little before dead center, not more than 20° in any case and if the engine is fast running and much muffled certainly not before dead center.

Inlet closure takes place from 30 to 45° or more past the lower or outer center. For most engines operating up to 1200 piston ft, 30° lag is enough. If it is desired to establish this angle accurately it should be varied and the compression with each setting plotted. The setting that gives the greatest compression naturally gives the most complete charging and should be adopted.

In a long series of tests, varying all the opening and closing angles of the valves, the greatest difference in economy recorded was 0.02 lb per bhp-hr, and the influence on rating was indeterminate. Naturally the variations tried did not exceed the limits set by common sense or much larger losses might have been shown.

CAMS

Constant-acceleration cams of the type described in Chapter V are most satisfactory. For moderate speed full opening is realized in 50° of camshaft travel. For very high speeds this may be prolonged to 60° to reduce valve gear acceleration.

If the valve settings are so chosen that the inlet and exhaust functions use equal angular intervals, the cams may be alike. Thus with satisfaction in many cases we may use the following settings:

Inlet opens	20° before dead center
Inlet closes	35° after dead center
Exhaust opens	35° before dead center
Exhaust closes	20° after dead center

Both cams act for 235° of crankshaft travel. If then the keyseat in the cam is placed as in Fig. 66 a single keyseat in the camshaft serves for each cylinder station. The exhaust cam is like the inlet cam but is so assembled that it leads its key while the inlet cam follows it. If so made all inlet and exhaust cams for a reversible engine are alike and they are the same for port and starboard engine or for stationary engines driving either clockwise or counter-clockwise generators.

The idle part of the cam profile may be relieved as in Fig. 66, the concentric contour being joined to the rise by a transition radius r which takes up the valve clearance gently before the valve is accelerated from its seat.

Cams should be drop forgings, although sometimes they are made of chilled cast iron or of steel castings case-hardened. The drop-forged cams made of S.A.E. 1020 steel are given the heat treatment S.A.E. – H.T. – 1020 V. Their hub strength

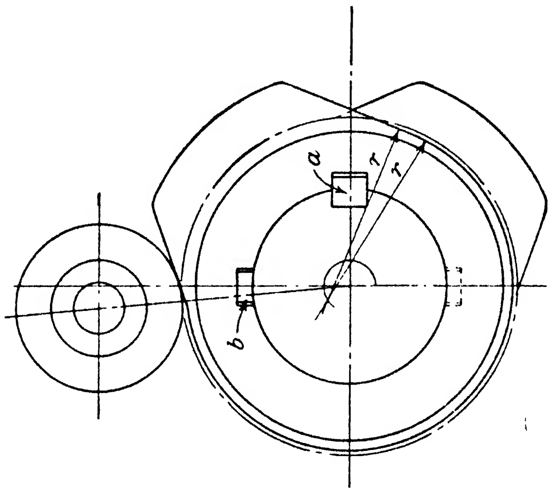


FIG. 66

through the keyway is great on account of the refined core. Chilled-iron cams require larger hubs to withstand keying, and so do case-hardened steel castings as the carburizing gas penetrates too deeply into the porous casting to leave a very tough core.

Cams are secured to the shaft in many ways. When pressed or shrunk on they may be provided with a different bore for each seat on the shaft. This is a nuisance in manufacture, stores and service. It is far more satisfactory to use turned and ground shafting for the camshaft, making the cams a slip fit. The cams are then secured with a shear key, Fig. 66*a*, and a wedge key, Fig. 66*b*.

CAMSHAFT DRIVE

The once conventional camshaft drive is shown in Fig. 67 although some designers preferred bevel gears.

Housings for the bevel and spiral gear drives reached an admirable though expensive state of perfection. The lower gears in each set ran in oil, and pockets were arranged to intercept the oil thrown off whence it was conducted to the bearings.

From a manufacturing viewpoint the results were far from satisfactory. The use of spiral or bevel gears made necessary special provision for thrust, and great skill was required to align the drive.

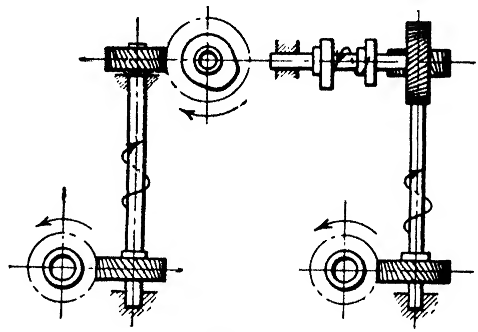


FIG. 67

Owing to necessary manufacturing limits the meshing of gears was variable and similar units exhibited notable peculiarities in sound except when skillful matching and fitting were attempted. To avoid the grief heaped upon them by such details in the building of hundreds of thousands of motors yearly, automotive engineers introduced the silent chain for valve gear drives, greatly simplifying design and manufacture and certainly eliminating any annoying tendency toward individuality in the operation of motors of the same origin. For the same reason European designers of large engines of the diesel type adopted the roller chain for valve gear drives using vertical flights of many feet and pitches as large as 3 in. Special chains of even greater pitch were designed for engines of high power. Their success was universal, and such drives now are established in this country.

Housed in an engine frame and lubricated with a stream of oil applied inside and near the approach to a sprocket, properly designed roller chain drives have shown no considerable wear after four years of use. The accuracy of timing effected is equal to any requirement. It is far better than that of right-angle spiral gears when they have developed lash, which they almost invariably do.

In designing camshaft drives a torque curve is plotted from which the maximum chain pull is determined. While factors of safety as low as 20, or lower in certain cases, have been used successfully, for long chain life it is better to select a chain on the basis of pin load rather than breaking strength.

The design is based on a wear factor, not a safety factor. Neglecting the effect of centrifugal force about the sprocket the pin load should not exceed 900 lb per sq in. of projected area for heavy jobs. It never should be greater than 1200 lb per sq in. This makes the ratio of breaking strength to working load equal 30 to 45 depending on the design of the chain.

For this type of drive the chain velocity should be limited to 900 ft per min where possible and should not exceed 1200 ft. No sprocket in contact with the chain should have less than 19 teeth, preferably never less than 21.

Multiple chains, Fig. 68, of the double, triple or even quadruple variety are used to provide adequate strength with small pitch thus making possible the required number of teeth with reasonable pitch diameters and allowable linear velocities.

Every chain is provided with an adjustable jockey sprocket having a minimum of 19 to 21 teeth. It should be so located that at least 3 of its teeth are in engagement at any position. Jockey sprockets are best provided with anti-friction bearings. In determining the position of the jockey, for any drive, it usually is advisable to locate it on the inside of the chain and thus obviate backward bending.

Chains are preloaded by the manufacturer, that is, they are stretched on a draw bench under a pull exceeding the operating load, thus practically eliminating the initial or seating elongation due to wearing in.

The camshaft diameter may be made a simple proportion of the cylinder diameter. This proportion is often $1/4\,D$. Analysis of the camshaft for stress and deflection even under the loading of impulse pumps yields rather ridiculously low values, yet a sweet-running engine requires a camshaft of quite ample proportions.

THE VALVE GEAR WEIGHT

It is common practise to fit valve springs of such initial tension that the valves cannot be opened by any possible vacuum that may be produced in a cylinder. The usual requirement is from 12 to 14 lb of spring tension per sq in. of valve area when the valve is seated. The valve gear mass should then be such that this spring tension will preserve cam and cam roller contact at any speed required of the engine, reserving at least 50 per cent

FIG. 68

excess spring tension over the maximum acceleration force for friction normal and abnormal. If, as it often happens, large engines operating at comparatively high speed require excessive spring tension when the push rod type of valve gear is contemplated, the push rods must be eliminated by raising the camshaft to the cylinder head level, Fig. 37. For very high speed even the valve rockers are eliminated by placing the cams directly above the valves.

The push rod variety of valve gear, Fig. 38, is inherently a low-cost type of construction. The frame provides seating for the camshaft bearings so that no separate housing is required. Lubrication is automatic from oil thrown by the cranks, and no special system of oil guards or drainage is required. The overall dimensions of the engine particularly at the top are somewhat reduced by housing the camshaft in the crankcase.

Owing to the expansion of cylinder on which the fulcrums of the valve operating levers are supported the valve or cam clearance never can be maintained accurately so the push rod engine with its indeterminate cam and roller clearance and its many joints may not be as quiet as the engine with the elevated camshaft. Nevertheless, its popularity will persist with both builder and owner on account of its simple construction and cheapness.

The substitution of light drop-forged rockers for castings and the use of tubular rods have done much to improve the running of the push rod type.

VALVE SPRINGS

When constant-acceleration cams are used the spring calculation is a simple one. Thus if the valve is lifted in 60° of camshaft travel, acceleration is known to take place during half of this period or 30° of camshaft travel. In four-cycle engines this equals 60° of crankshaft rotation or $\frac{1}{6}$ revolution.

At 300 rpm, $\frac{1}{6}$ revolution equals

$$\tfrac{1}{6} \times \tfrac{60}{300} = 0.0333 \text{ sec}$$

If the lift is 1 in. and acceleration endures to half lift, or

$$\frac{0.5}{12} = 0.0417 \text{ ft}$$

then in

$$S = \tfrac{1}{2}\, at^2$$

$$a = \frac{2\,S}{t^2} = \frac{2 \times 0.0417}{0.0333^2} = 75.1 \text{ ft per sec}^2$$

As the acceleration due to gravity or that of 1 lb force on 1 lb mass equals 32.16 ft per sec^2, the valve spring for the above case should provide $75.1 \div 32.16 = 2.3$ lb force for each pound of mass in valve and gear including one-half the spring mass all of which should be reduced to the valve or cam centers.

Large-diameter low-frequency springs having not less than twelve coils suffer least from vibration, and high-carbon spring steel worked at the low stress of 40,000 lb by the Begtrup formulas is advisable. The scale of the spring may be such that axial compression equals to three times the valve lift is required to secure the loading when the valve is at rest. The alloy materials of very high elastic limit are remarkable when homogeneous, but steel makers claim that they are difficult to manufacture and thus account for epidemics of spring failure.

STARTING VALVE GEAR

Most of the conventional semi-automatic starting valve designs in use have one common property — the starting air-admission valve in the cylinder head goes out of action automatically when the cylinder that it serves starts to fire. This obviates any danger of firing back into the air tanks. It permits starting with fuel on, which is always much more certain than the old method of rolling the engine on air, then throwing the hand gear that puts the starter out of action and the fuel valves in. The latter method is wasteful of air and altogether too slow for marine work. With the semi-automatic valve the main air throttle may be left open and the valve-operating mechanism may be allowed to continue operation after the cylinders are firing. No harm will result, as the admission valve cannot open against the pressure of explosion.

The commonest of these devices, and a very good one, is that which originated with Burmeister & Wain, Fig. 69. A balance piston opposes the automatic opening of the valve when air is admitted to the common air main to which all the starting valves are connected. As this piston is the same size as the inside of the valve seat, the valve is not quite balanced, because air tends to work in between the valve-seat faces and lift it. A spring makes up this difference and a little more to insure a tight seat. The spring also serves to hold the valve in place when the air pressure is shut off. A small hole is drilled up through the valve stem to

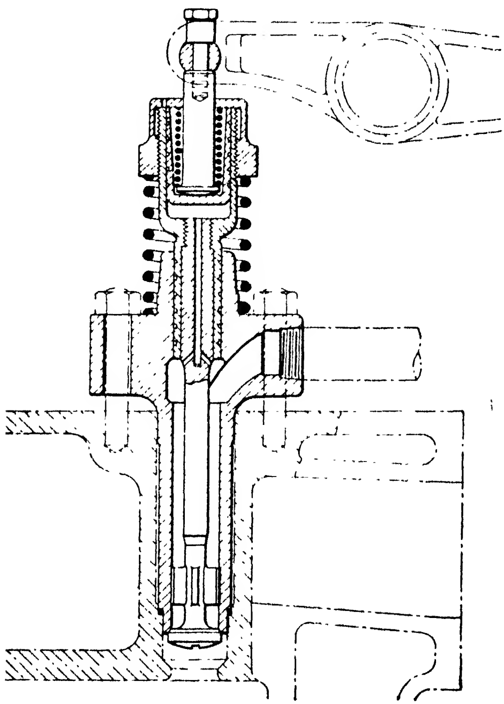


FIG. 69

the cushion cylinder on top of the stem. In this cylinder is a free piston which is forced up against the cap when air is admitted to the system. This operation extends the starting valve stem so that it comes in contact with the starting-valve operating lever at one end, depressing the other end into the path of the cam. Thus, as soon as air is thrown on for the purpose of starting, all valves are thrown into the operating position. The engine begins to turn and air is admitted to the cylinders in sequence. As soon as firing begins, the explosion pressure holds the valve so firmly to its seat that the air pressure within the cushion cylinder, being only one-third or one-fourth the explosion pressure, is not sufficient to force the valve open. The piston in the cushion cylinder then merely oscillates up and down without unseating the valve, until the operator shuts the air off.

Another commonly used system, Fig. 70, employs the valve *B* in the cylinder head as a check, or non-return valve only. A second cam-operated valve *A* for each cylinder is located under the camshaft. .When air is admitted to the air main by opening the throttle, the mechanically operated valve is raised by the

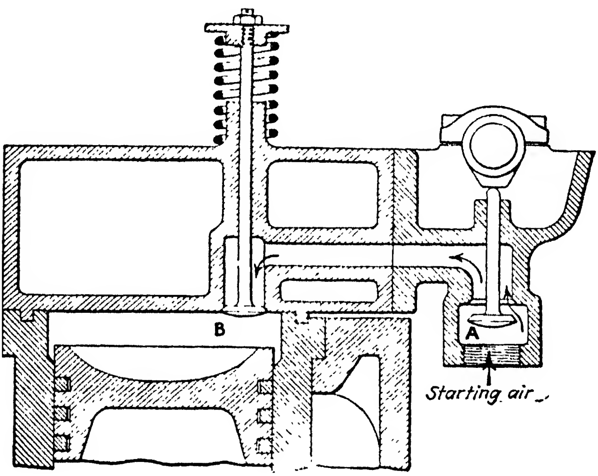


FIG. 70

unbalanced air pressure so that the stem comes in contact with the cam. The several valves admit air to the cylinders in sequence through the non-return valves *B* in the cylinder heads until the explosion pressure holds the latter shut and prevents further admission of air.

Yet a third system is used, mostly because it requires no cam and lever on the cylinder head and because its cam-operated mechanism is quite compact and can be placed at any convenient position. In the system the valve in the cylinder head, Fig. 71, is balanced and spring loaded exactly like the first one described. The valve is operated by an air ram, however, instead of being merely thrown into contact with its operating lever by the action of an air piston. In other words, the valve is pneumatically operated by admitting air to and exhausting air from an air ram mounted above it. The air is distributed to the valve operating rams by pilot valves, Fig. 72, there being one of the latter for each cylinder. The pilot valve itself is a little D slide valve *A* with outside admission and exhausting through the space

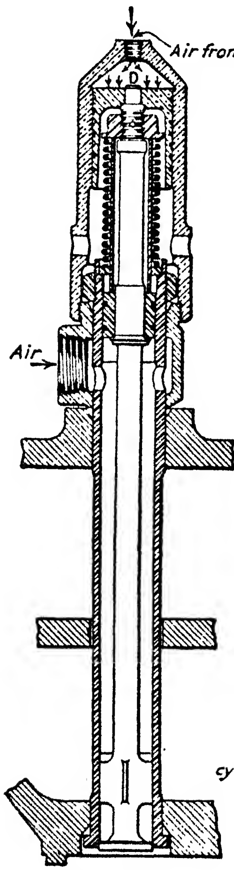


FIG. 71

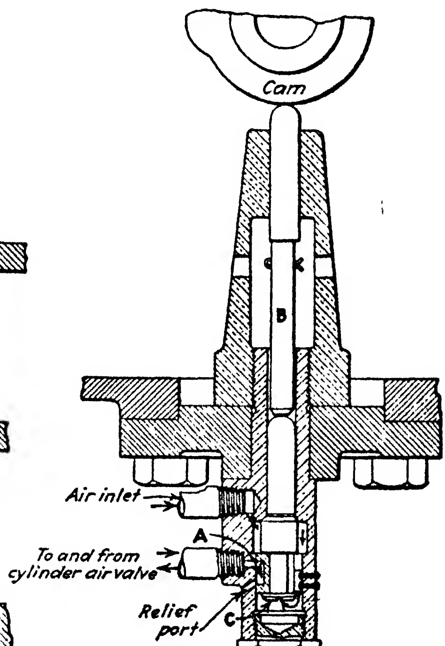


FIG. 72

under the valve. A slide valve is chosen rather than a poppet valve because it combines both inlet and exhaust functions in one valve and because it is not easily unseated by grit or débris like the poppet valve. It is thrown into action by the admission of air to the system which moves the stem *B* upward and into the cam path by reason of the unbalanced area of the stem at *C*. It drops out automatically when air is shut off. The piston *D* that operates the cylinder admission valve, Fig. 71, is too small to open it against the pressure of explosion.

This system has the same properties as the other two discussed. It is common practise to use a reducing valve in the pilot-valve air main, set for some pressure approximating the lowest useful pressure in the air tanks, such as 150 lb. The air ram is then designed to function best at 150 lb pressure, and the pilot valves and ram work under a fixed condition, regardless of the tank pressure.

All these are what may be called "four-stroke-cycle" starting systems; that is, they admit air every other revolution. Older gears admitted air every revolution as in two-cycle engines even when used on four-stroke-cycle engines. This was done by throwing the inlet valve out of gear and causing the exhaust valve to make an extra movement on what is normally the compression stroke. Air under pressure was admitted on what should be the power stroke, and on the intake stroke as well. This mechanism had the questionable advantage of making a start with a very low air pressure.

Before starting, engines with less than six cylinders usually are "set up" or barred by hand to a crank angle at which one starting valve is in the cam-contact position. Six-cylinder engines start from any position. Eight-cylinder engines in which the starting impulses overlap considerably are very prompt in their getaway and are least likely of all to prove unreliable when wear alters the starter timing, as it usually does, by increasing clearances and shortening the angle of admission.

Starting systems for stationary engines usually are quite free from troubles. On ships engaged in canal, river and harbor work, these mechanisms assume great importance. Instead of being used once a day or once in weeks or months the starter may be used several hundred times in 24 hours and must function perfectly each time if collisions are to be avoided. For this class of service the starting system must be built with just as much attention to detail as any other part of the engine, and careful maintenance is absolutely essential.

Perhaps the most distressing and often mysterious occurrence the operator faces is air locking. Imagine two cylinders of a six-cylinder engine standing with their cranks at equal angles off center, one on the starter cam, the other on the compression stroke, but both cylinders charged with air at the pressure of the starting receiver. If this seems a remote and improbable condition, it is not so difficult to imagine two cranks ranged at some angle above that of exhaust valve opening, with the cylinders charged with air at a pressure such that the turning effort of the one neutralizes that of the other. If there is pressure air in

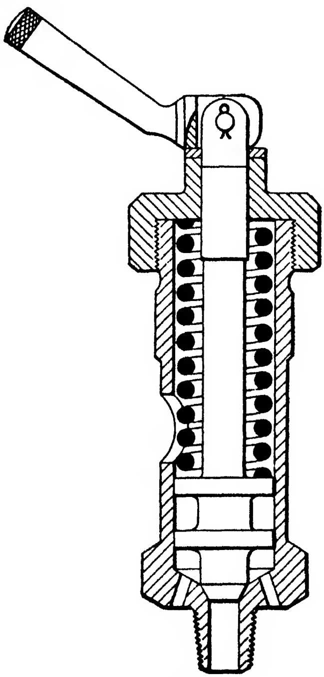


FIG. 73

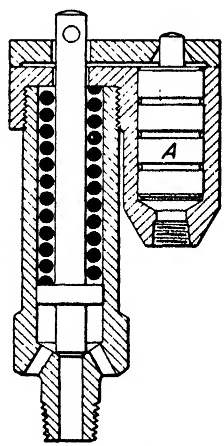


FIG. 74

two cylinders and no momentum on the engine, the cranks will seek this position. It then will be impossible to move the engine by the use of starting air until the dead cylinder, the one not connected to the starter cam, is relieved of its pressure. There are so many possible and almost unavoidable causes for this condition that it is a serious design problem.

If the engine is a stationary one, it is necessary only to open the cylinder relief valves, Fig. 73, or vent valves one by one. Air will be relieved from two of them.

But the venting of cylinders by use of individual hand-operated relief valves is a process too slow for marine engines, although it does well enough for the stationary plant where minutes may not be so important as seconds on the water. Marine engines are fitted with two general types of mechanisms for dealing with this contingency. The first and best opens all exhaust valves each time the valve mechanism is thrown from the position for one direction of rotation to the opposite position.

A device that is coming into use for the same purpose is the air-operated cylinder relief valve. An American example is shown schematically in Fig. 74. The little air cylinder *A* mounted alongside of the relief-valve spring case lifts the relief valve by lever action when air is admitted either manually through a three-way cock or automatically by interconnection with the air system used for reversing.

STARTING VALVE TIMING

Stationary engines with less than six cylinders, which have to be barred to starting position, may, for the sake of saving air, be given a comparatively short starting-valve cut-off. The valve may remain open as little as 60° and still give practically as good a starting impulse as with a longer admission. Again the period of admission may be 120° or more in order that the setting may be standard for all engines of a certain manufacture, regardless of the number of cylinders and regardless of whether they are intended for stationary or marine service.

It is quite usual to give the starting valve some lead but seldom more than 10°. This should never be excessive for marine engines or other engines that are intended to start from any position without barring.

In the case of the pneumatically operated starting valve, the setting of the pilot valve does not indicate the true timing. There is a lag in both opening and closing due to the capacity of the piping between the pilot valve and the starter-valve air cylinder and the capacity of the air cylinder itself. This action must be developed by the use of the indicator applied to the air cylinder of the starting valve itself. The operator, of course, makes all his settings by adjustment of the pilot valve timing. This is done by disconnecting the small pipe between the pilot valve and the starter-valve cylinder. Air is turned into the pilot-valve line and the engine is barred over until air starts to discharge from the little pipe. This indicates the opening of the pilot valve. The pipe is then reconnected and the engine barred further until the pilot valve begins to exhaust. This indicates the closing of the starting valve.

As has been indicated, the timing for six-cylinder direct-reversible marine engines must be set very closely. For this purpose a completely adjustable starting-valve cam such as is

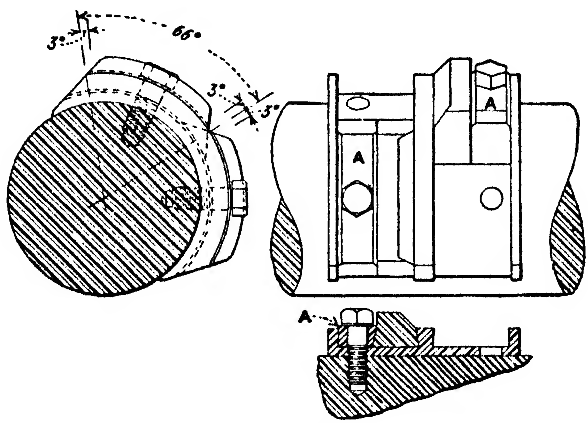


FIG. 75

shown in Fig. 75 is used. By loosening the clamp segment *A* the nose of either cam may be moved any desired amount. The two cams are for ahead and astern motion.

When engines have been operated for a long period without shut-down on land or sea, it is quite common for a deposit of burned oil to form about the starting-valve head and wing guide, sealing it against opening. Where effective starting is necessary to service or safety, the starting valves should be cared for by the exercise of a rigid routine.

On coming into a port each engine should be maneuvered to determine whether the valves are in working order. If not, it will be possible to make the necessary correction without the mischance of having the engine fail to respond in traffic or when docking. It is becoming more and more evident that all starting valves should be in cages, and also should be fitted with hand-operating gear, to permit freeing of the valve by hand working without the necessity of dismantling.

Aside from the operating precautions described, starting valves in service where their working is imperative, as in tug-boats and the like, are cleaned sometimes daily and certainly once a week. Every possible design measure should be adopted to reduce this necessity.

REVERSING MECHANISMS

Most reversible engines are maneuvered by shifting the camshaft endwise so as to present to the cam rollers the set of cams required to time the functions of starting, intake, fueling and exhaust, for the desired direction of motion.

Small engines use cams with inclined sides, Fig. 76, and chamfered or rounded roller edges. The camshaft is forced ahead or

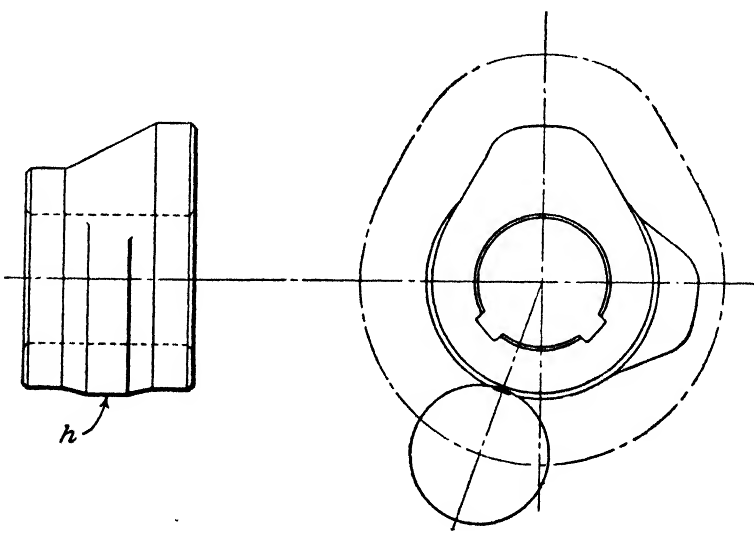


FIG. 76

astern by a hand lever or air ram to the position required. Any roller which happens to be on a cam at the time slides down the inclined side and if the position requires rides up the inclined side of the substitute cam to its new position.

Larger engines use a more elegant mechanism. The valve lever fulcrums, Fig. 77, are made eccentric. By revolving these fulcrums half a turn the valve levers are lifted and being held down at the valve end by bars all rollers are lifted clear of the cams. The camshaft is then shifted, putting the second set of cams in position to function, and the eccentric fulcrums complete their revolution lowering the valve levers with their roller ends into place. These operations are controlled by a cam and rack,

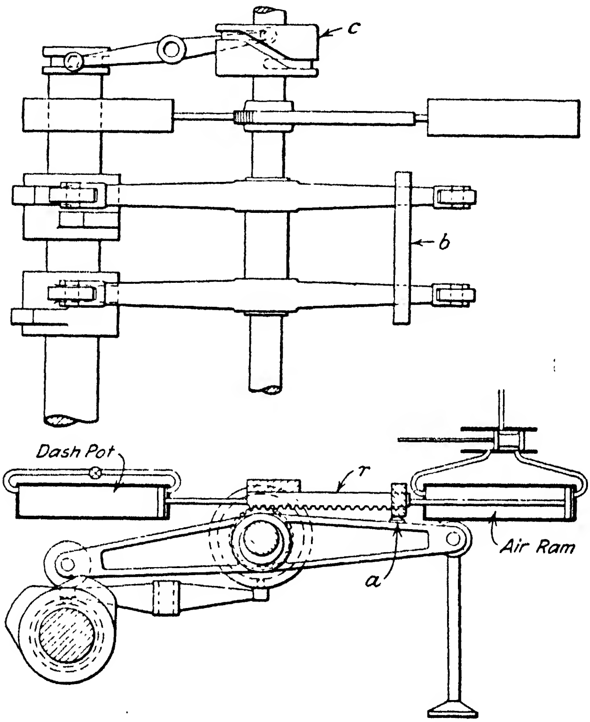


Fig. 77

Fig. 77, *c* and *r*, operated by an air ram. An oil-filled cylinder in tandem with the ram acts as a dashpot to prevent irregular or violent shifting of the gear.

Practically all American direct reversible engines of the solid injection four-cycle type use the common-rail system of fueling. This system adapts itself well to marine work and especially to reversing. Gear for maneuvering high-pressure fuel pumps is cumbersome and adds much to the complication of the mechanism while the low lift spray valve cams are easily substituted, ahead cam for astern cam, as the shaft slides without the necessity of lifting any mechanism out of the way. The governing wedges are withdrawn while reversing, leaving the spray valve mechanism free to slide up and over the chamfered sides of the fuel cam should any one of them be in the roller path when the shaft is shifted.

A variety of mechanical interlocks has been devised to prevent admission of starting air and fuel while shifting cams or to prevent accidental maneuvering when the fuel lever is in the running position.

A very simple one, proof against ignorance or panic, is shown in Fig. 78.

The single lever is capable of two kinds of movement. It may be rocked on pin *a* along slot *b* without turning the shaft, or it may be moved through slot *c* or *d* turning shaft *e* with it.

A shifting pin *f* engages slide bar *g* when the lever is in the transverse slot *b*. The slide bar operates a piston valve which admits and exhausts air from the ram, Fig. 77, that maneuvers the main valve gear. Thus ahead or astern movement of the lever through the transverse slot causes the camshaft to slide to the corresponding position.

The rock shaft *e* which is stationary while the lever is moving in the transverse slot carries three arms. Arm *h* operates the fuel valve wedges. Arm *i* carries a trip mechanism for operating a pilot valve which controls the pneumatically operated starting air throttle. Arm *j* actuates a pilot valve in the body which controls the air-operated power cylinder relief valves shown in Fig. 74.

In Fig. 78 the lever is shown in the stop position with the camshaft set for astern rotation. If the telegraph indicates "Full Speed Ahead," the lever is shifted ahead through the transverse slot carrying the piston valve with it. The camshaft under the influence of air pressure directed by the piston valve through port *k* follows to the position for ahead rotation, the air pressure on the other side of the double-acting ram exhausting over the end of the valve *m*. The lever is next moved at right angles to its first movement through the ahead slot. Arm *i* picks up the air starting pilot valve *n* by means of latch *o*, and arm *h* moves the fuel cam wedges in, setting the fuel valves in motion as the engine turns over on air. Further movement of the lever causes

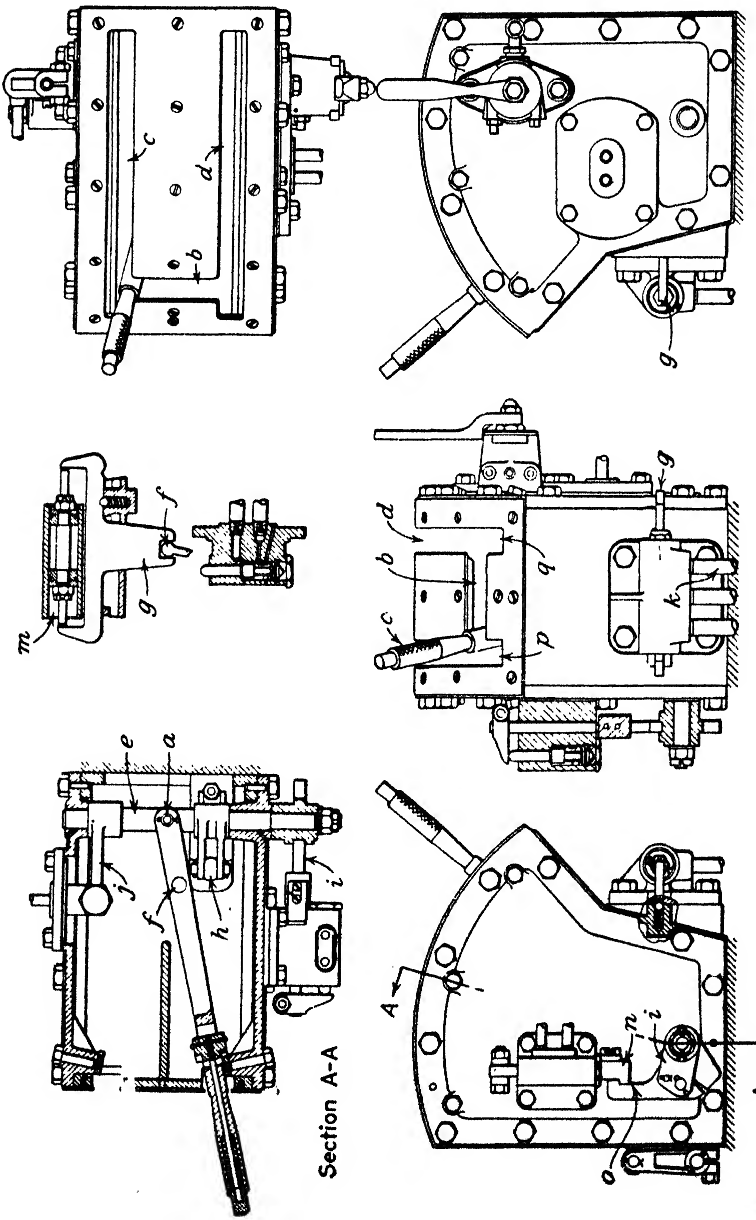


FIG. 78

the air valve latch *o* to trip, shutting air off as fuel admission is increased. The engine is now in operation.

The latch mechanism on the air starter pilot valve is provided to prevent re-admission of air when the lever is returned to stop position.

The arm *j* operates cylinder relief valve pilot valve when the control lever is dropped into notch *p* or *q*. This function is called upon only when the engine is airbound and refuses to turn on starting air as described in this chapter.

Another means of relieving air pressure in cylinders at each maneuver is more often used and far simpler. The exhaust valve levers are provided with a cam surface at *a*, Fig. 77. This engages the hold-down bars *b* when the eccentric fulcrum shaft lifts the levers off the cams, opening all exhaust valves a fraction of an inch during the act of reversing.

For non-raising gears which simply force the camshaft laterally the hump *h* on cam, Fig. 76, serves the same purpose.

PROPORTIONS OF VALVE GEAR PARTS

The shafting, links and levers of the valve mechanism are required to work against the valve spring resistance and a terminal gas pressure of about 30 lb per sq in. of valve area for exhaust valves. However, all gear should be designed for safety when lifting the valves against the air starting pressure. This condition is encountered when braking reversing engines to a stop by applying the starting air and occasionally in stationary engines with leaking starting valves.

As the starting air pressure usually is 250 lb per sq in. and combined spring and gas pressure seldom exceeds 45 lb per sq in. the ratio between abnormal starting or maneuvering stresses and normal operating stresses is $250/45 = 5.55$. If the valve gear were designed with ordinary factors of safety for the heavier load it would be quite ponderous. For this reason it is customary to use rather high stresses, such as 20,000 lb per sq in., in designing these members with the starting pressure as a basis. The normal operating stress will then be $20{,}000 \div 5.55$ or about 3600 lb per sq in. This results in the ample stiffness essential to quiet operation.

Aside from this it is not possible to derive general rules for valve gear design since proportions vary so greatly with the speed and weight limits set for the engine. There are gears capable of operating at several thousand revolutions per minute and gears with a useful life of twenty to thirty years, each good and sound engineering in its own class. The designer's judgment and the materials employed determine valve gear proportions very largely.

ELECTRIC STARTING

The electric starting systems used with small diesel engines are replicas of those in automobiles. Twelve-, eighteen-, twenty-four- and thirty-six-volt outfits are provided although the available twelve-volt apparatus scarcely suffices to start even the smallest four-cylinder diesel at temperatures below 32°F unless the jackets are first filled with hot water.

With the thirty-six-volt systems, six-cylinder engines of bores up to 6 in. or more have been started with oil (S.A.E. 1020), air and jackets at $-10°$ to $-20°F$ if an efficient electric resistance heating unit is installed in the intake manifold. The compression exponent is very low in small engines when the combustion chamber walls are below 32°F, and the compression end temperature rarely is high enough to insure spontaneous ignition unless some form of preheating is used. Instead of the intake air heater, glow plugs in the combustion chamber may be used Fig. 21 or punk sticks may be inserted in plugs with breech lock threads. Electric apparatus is preferably installed outside the combustion chamber, however. The electric preheater draws from 600 to 2000 watts for one to five minutes prior to starting.

Starting motors are of the series type. They should be capable of supplying a rolling torque at the crankshaft equal to twice the normal full-load torque of the engine when cranking at 200 rpm. The ratio of bendix or motor pinion to flywheel ring gear is usually 10 to 1. It is useless to make this very much greater. At 20 to 1, for example, the starting motor would be required to turn 4000 rpm to crank the engine at 200 rpm. This high speed is too near the limit for the unbanded motor armature for safety as a considerable margin must be allowed for speeding up before the bendix disengages. Cranking at less than 150 rpm is not effective since too much time is available during each cycle for heat loss to the cylinder walls.

The expedient of relieving compression to make starting easier is no longer in general use. Once the engine is turning most of the work of compression is recovered in re-expansion, while the work of wiredrawing air through the valves is not recoverable. Engines usually require least battery energy when starting on full compression.

The full battery energy may be made available in very cold weather by housing the cells in insulated or heated chests.

A peculiar flywheel problem is introduced by electric starting. Naturally the cyclical variation of speed is great in an engine designed for 1000 to 2000 rpm when it is turning at 200 rpm. This variation may be sufficient under light and intermittent firing to disengage the bendix and re-engage it with great shock. Other requirements being fulfilled the wheel should be designed for a coefficient of regularity better than 1/5 of the starting speed of 150 or 200 rpm.

CHAPTER VIII

TWO-CYCLE ENGINES

LABORATORY METHODS

Glass cylinder models are useful to guide the student of scavenging systems. These may be full cylinder size and served by turbo compressors at the normal scavenging pressure or they may be very small. Many ways have been devised to make the process visible and some ways for making photographic records. The commercial systems are developed so well that little of this kind of research is required.

Invention in this field is best checked by models before much money is spent for patents or construction. The patent records include thousands of drawings showing by flights of arrows the imagined or desired path of the air streams in scavenging a cylinder. The simple truth is that scavenging is not a very orderly process. Practical port dimensions call for undesirably high air velocities to introduce the fresh charge in the time allowed by the speed of rotation. The over-energized stream is stopped by the cylinder wall or the cylinder head or by convergence in the case of opposed piston engines, loses its velocity head and displaces or more accurately crowds the neutrals out of the exhaust ports.

The entering stream of scavenging air sets up whirls by frictional contact with the neutrals. The stream toward the exhaust ports often acts with the scavenging air stream forming a couple to intensify these whirls. Eddies cause further mixing.

All these actions are made visible in small cylinders filled with tobacco smoke and scavenged by a hand pump of proportionate size through ports and deflectors modeled in wood. Larger models use pyrotechnic powders burning in the scavenging air, or the cylinder may be filled with water colored with aniline dye and scavenged with clear water. A more practical method because it is continuous is to put into the glass model, in stages, a number of radial wires to which long silk threads are fastened. When the blower directs air through the modeled ports the threads assume the directions of the air currents in every part of the cylinder.

Gas sampling from the cylinder and at the exhaust ports through timed valves coupled with chemical analysis of the gases collected has been tried repeatedly. Scientific accuracy is required, and even then a doubt as to the efficiency of sampling always lingers.

It is very difficult at best to develop the scavenging system of a two-cycle engine simultaneously with the fueling system and combustion chamber. The great overload capacity of the oil engine and its comparatively flat economy curve in the range from three-quarter to full load make it hard to determine the value of small improvements with accuracy. The fuel economy, too, varies with the character of the fuel, no two shipments of which are exactly alike. These variations of economy are in no wise proportionate to the heat value of the fuel, in fact very heavy fuels having the greatest heat value often burn at reduced thermal efficiency.

The gas engine with its fixed maximum mep and its very small overload capacity (power in excess of that developed with the mixture giving best thermal efficiency) immediately indicates any change in the charging efficiency. The injection type of gas engine in particular, avoiding loss of fuel gas at the exhaust ports, indicates immediately and directly at the Prony brake the effect of any experimental modification of the scavenging system.

Taking advantage of this property, when an ample supply of natural gas is at hand the development of a two-cycle diesel exhaust or scavenging system may be accelerated by fitting a cylinder head provided with a gas injection valve and an enlarged clearance space giving suitable compression for operation as a gas engine. This method has another use. It insures convertibility often desired by clients. It adds to the diesel line a gas engine design which may itself find ready sale because of the possibility of conversion for diesel operation should the gas supply fail.

Several factors enter into efficient scavenging. The major influences are

1. Scavenging air pressure.
2. Porting.
3. Exhaust system.

These factors are linked inseparably.

SCAVENGING PRESSURE

The first assumption made by the inexperienced is that air at high scavenging pressure must necessarily sweep out the products of the previous combustion more effectively than a comparatively gentle flow from the ports.

There might be some justification for the use of high-pressure scavenging if an ample quantity of air were available. The scavenge volume, however, is limited as the pumps at best are bulky and the negative work required to compress scavenging air always is a handicap.

The least pressure that will insure delivery of the air volume provided through ports of reasonable (or possible) size effects the best scavenging. High-pressure air sets up eddies, and if the profile of its path is favorable the high-velocity air stream may be diverted to the exhaust ports. The lowest scavenge pressures used result in velocities far exceeding the wind velocities during hurricanes and cyclones.

SCAVENGING PUMPS

The pump design depends upon the size and class of the engine. Although not the oldest, "crankcase compression" as in Fig.

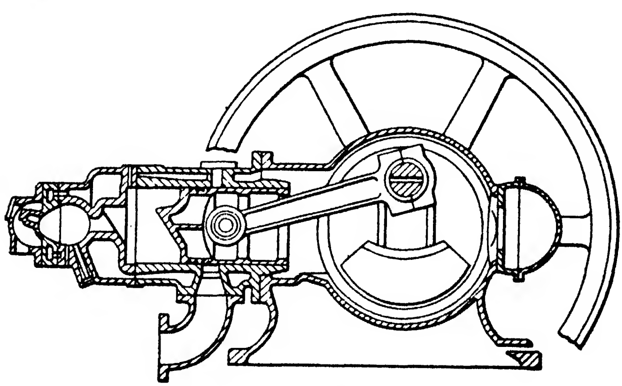


Fig. 79

79 was the form most prevalent for many years. It is still in use for small engines.

Fig. 80 shows power piston pumping in which the crank end of the power piston incidentally displaces scavenging air in the open cylinder end and in a chamber in the bed plate instead of in the whole crank chamber. These bed and cylinder clearances determine the scavenging air pressure and act as a receiver.

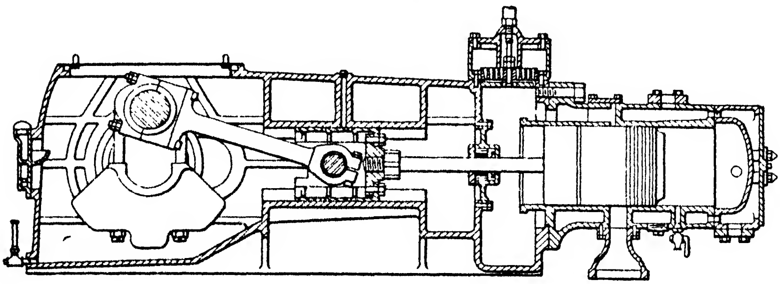


FIG. 80

With well-designed inlet valves and discharge ports this produces a card like Fig. 81. If anything hinders the discharge of the air from the ports the card takes the form Fig. 82, showing low volumetric efficiency due to back expansion from the large

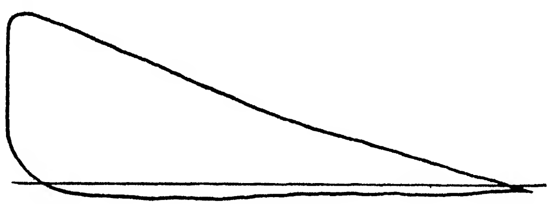


FIG. 81

FIG. 82

clearance. This clearance is necessarily great since a small clearance would result in abnormally high scavenging pressure.

Fig. 83 shows the usual form of scavenge pump for large engines. This pump is fitted with both suction and discharge valves. It discharges into a receiver or manifold of great volume to minimize pressure fluctuations. This manifold extends length-wise of the engine near the inlet port level and serves to distribute the air to the various cylinders through suitable short connections. With this type of pump any change of resistance at the ports is met by an equivalent rise in scavenging air pressure since air entering the pump and passing its discharge valves must go through.

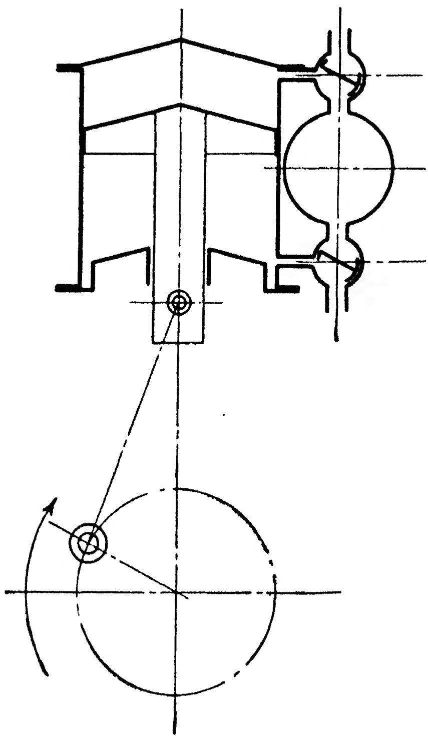


FIG. 83

PUMP VALVES

Engines pumping air by displacement of the power piston have inlet valves only. These may be automatic plate valves, in which case they are duplicates of the forms used in air compressors.

Engines using forced scavenging employ double-acting pumps fitted with suction and discharge valves. The valve and aspiration noises of these great low-pressure pumps serving as many as six or eight cylinders are responsible for the tom-tom beat often associated with two-cycle engines. As the working pressure is very low, simple forms of oscillating valves may be substituted for the automatic type with pleasing results since they are quite noiseless in operation. A single valve, Fig. 83, may serve the purposes of both suction and discharge. Indicator cards, Fig. 84, were taken from a pump fitted with such a valve. The mean air velocity through the valves should not exceed 6000 ft per min.

The properties of an oscillating valve are given in the Bilgram diagram, Fig. 85.

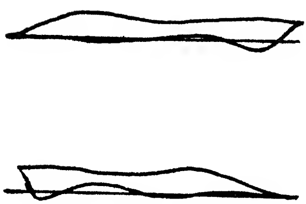


FIG. 84

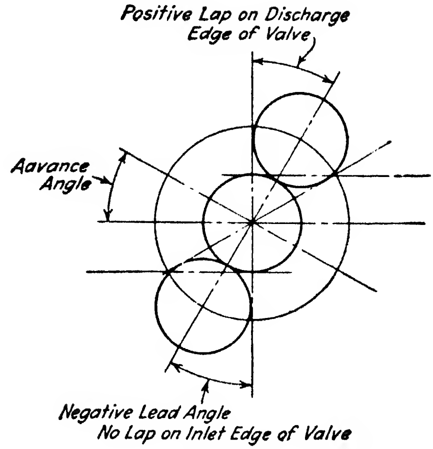


FIG. 85

BALANCING

Scavenging pumps are a disturbing element in balancing as they add to the balanced six- or eight-cylinder system a single-cylinder or twin-cylinder crank system concentrated at one end.

Single-cylinder pumps should be counterweighted. Either type of pump should be fitted with the lightest possible reciprocating parts. Aluminum pistons are used to advantage. No rings are required as no harmful amount of leakage can take place at the low working pressure.

For high-speed engines the Root-type blower has completely replaced the positive pressure pump. The noise due to air impact at aspiration is somewhat modified by making the lobes helical and by intake mufflers, but a reasonably quiet blower is still sought. Another disadvantage of the Root blower is that it is not simply reversible like the piston pump with automatic valves.

SAFETY PROVISIONS

Pumps and manifolds or trunks should be protected by spring-loaded hand hole covers at many points to prevent damage in case of oil explosions due to back-firing through the inlet ports. Flame guards should be fitted to the covers to prevent burning of the operators if they should lift.

EFFICIENCY OF SCAVENGING

The late Professor Hopkinson constructed the curve, Fig. 86, which indicates the efficiency of scavenging on the basis of complete mixing of each increment of scavenging air with the mixed volume of air and neutrals preceding it into the cylinder.*

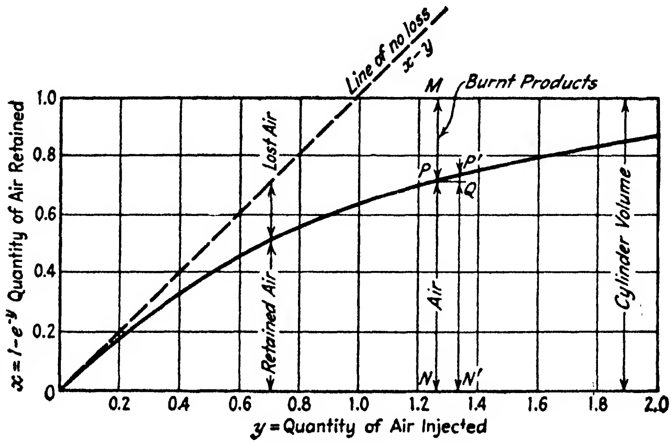


FIG. 86

Although this assumption is not very optimistic and better results should be attained, it perhaps represents average performance a little too well.

Scavenge pump displacement usually is made about 50 per cent greater than the total cylinder volume. This seems to represent a fair compromise between good scavenging and excessive negative work.

DEFLECTORS

Much ingenuity and more imaginative effort have been wasted on the profile of the deflector on the piston head. A simple arc, Fig. 87, of a radius equal to the inlet port length extended perhaps 10 per cent by a straight line seems to serve as well as anything. Fig. 87 shows a semi-diesel head.

If the deflector is used at all it is best to extend it all around the piston as in Fig. 88 to preserve the symmetry of the crown.

* *Trans.* N. E. Coast Inst. Engrs. and Shipbuilders, Newcastle upon Tyne, Vol. XXX, Part 6, page 436.

Lugs as at *A* are provided to prevent short-circuiting of the air to the exhaust ports.

The deflector in entering the head as in Fig. 87 creates a not

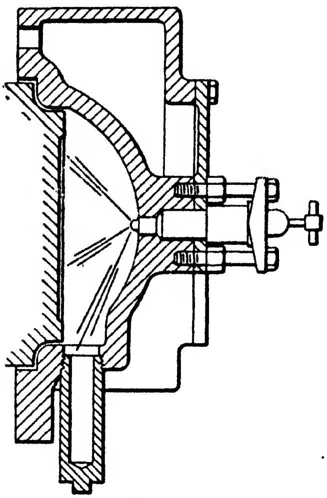


FIG. 87

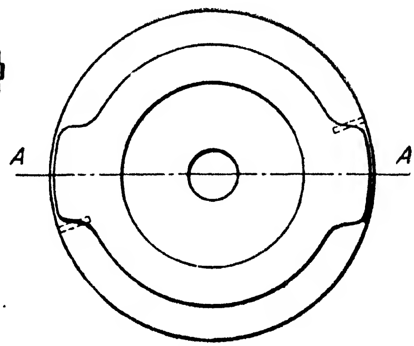
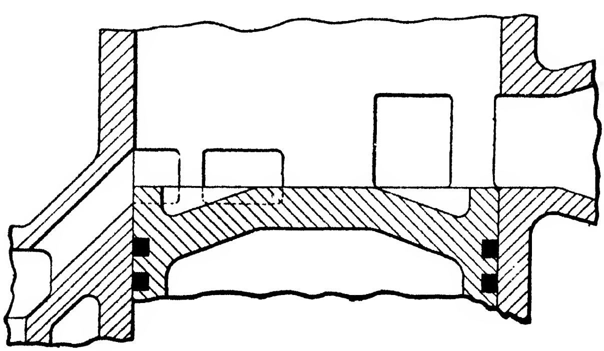


FIG. 88

undesirable turbulence effect which is useful chiefly to take up fuel that might skid off the piston.



Most diesel engine designs wisely avoid the deflector altogether. To this end the ports are inclined 45° upward as in Fig. 89. The piston fails to uncover the port completely at the lower-most position by about one-third of the port width which is essential to proper action. This arrangement is quite as efficient as any deflector and has the great advantage of allowing the use of a conventional diesel piston design.

PORTING

Perhaps the first rational treatment of inlet and exhaust port design is that published by Adam Kreglewski.*

The port openings are plotted against crank angles θ about the outer dead center.

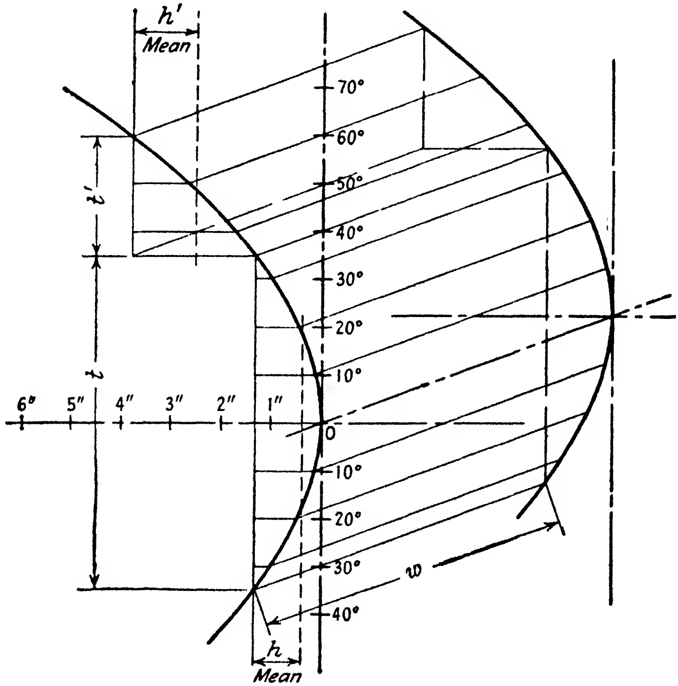


FIG. 90

The graph, Fig. 90, results. The mean port openings h and h_1 are next determined by the planimeter or by averaging abscissas.

Crank angles are reduced to time in seconds and a value (hwt) is used as a measure of port capacity.

* "Der Oelmotor," Vienna.

In this expression

h = the mean port opening during time t.
w = the total circumferential width of port.

The unit of port capacity, the square inch second (hwt), is equated by Kreglewski with an expression for adiabatic flow of gases through the port orifice. On account of the usual uncertainties as to the temperature of the air and exhaust gases, the variation of temperature and pressure of the latter as the cylinder is exhausted, cooling at ports, orifice coefficient, etc., a constant from practise is introduced. This being the case, it seems quite rational to embrace the entire flow term of the equation in a constant the pressure range being narrow.

For exhaust ports then

$$\frac{hwt}{V} = K_e$$

V = total cylinder volume in cubic feet.
K_e = required exhaust port capacity in square inch seconds per cubic foot of cylinder displacement.

For engines developing mean indicated pressures below 60 lb in the power cylinder

$$K_e = 0.10$$

For fully scavenged cylinders developing high ratings

$$K_e = 0.12 \text{ to } 0.15$$

The required inlet port capacity (unit = K_i) varies with the scavenging air pressure.

Extended experiments indicate that the value of K_i for any given pressure is $2/p$ for each cubic foot of scavenge pump displacement when p equals the maximum gage pressure of the air supplied by a system in which the scavenging pressure falls from the maximum developed in crankcase or chamber to atmosphere during each cycle.

For forced scavenging with valved pumps and receiver (Fig. 83)

$$K_{if} = \frac{1.2}{p} \text{ for each cubic foot of scavenge pump displacement.}$$

The expression relating the air flow to pressure developed out of a search for values for the constants in Kreglewski's formula in which a great many (24) cylinders were modified in various ways and worked under different scavenging pressures.

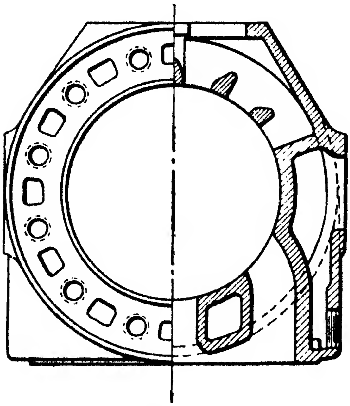


FIG. 91

In the past fifteen years it has been used in the design of a large number of engines having cylinders from 8- to 18-in. bore, speeds from 180 to 1000 rpm and scavenging pressures from 2 to 9 lb per sq in. Both power piston scavenging and forced scavenging were used, and the porting always has been such as to show excellent pump cards from the primitive systems and a close approximation to the desired receiver pressure in the forced systems.

An example in design follows.

First a transverse section of the cylinder, Fig. 91, is laid out to determine the possible circumferential width of inlet and exhaust ports consistent with a sufficient number of bridges to

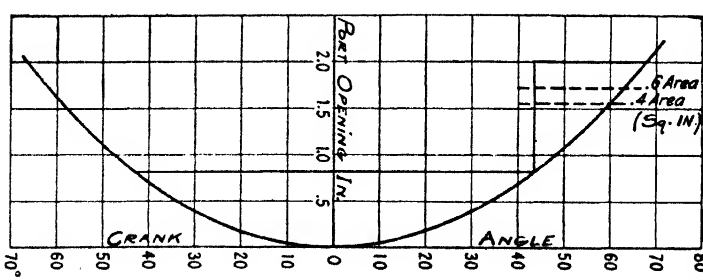


FIG. 92

guide the piston rings, proper exhaust port bridge cooling and sufficient tensile strength to resist the load applied by the pressure on the cylinder head.

In Fig. 91, which is a section of an actual 8-in. cylinder, this circumferential width is 8 in. for the inlet port and 6 in. for the exhaust port.

Fig. 92 shows the actual length of the inlet port to be $\frac{13}{16}$ in. This results in a mean length of 0.53 in. and a duration of port opening of 84° of crank travel.

At 600 rpm, 84° of crank travel requires 0.0233 sec of time.

The displacement of the scavenge pump is $1\frac{1}{2}$ times that of the cylinder, or 0.35 cu ft.

$$\frac{hwt}{V} = \frac{1.2}{p}$$

$$\frac{0.53 \times 8 \times 0.0233}{0.35} = \frac{1.2}{p}$$

$$p = 4.25 \text{ lb}$$

This is a fairly satisfactory solution. Port capacity is limited physically. If this were a large slow-running engine of the best type it would be desirable to operate with only 1.75 to 2.00 lb scavenge pressure, but for a simple little high-speed engine it is necessary to compromise.

In any case the inlet ports must be designed first. By reason of space or strength limitations the port calculation is made to learn the probable scavenging pressure, or in more favorable designs a scavenging pressure is chosen and the port is designed by trial and error to fit it.

Next, the exhaust port: after several trials we arrive at a port 2 in. long of which 2 in. $-$ $\frac{13}{16}$ in. $=$ $1\frac{3}{16}$ in. is useful in discharging the cylinder to atmospheric pressure before scavenging begins. The mean opening $h_e = 0.64$. The angular duration of discharge is 25°. The time duration is 0.007 sec.

$$\frac{hwt}{V} = K_e$$

$$\frac{0.64 \times 6 \times 0.007}{0.23} = 0.11$$

which is near enough to 0.1, the value assigned to K_e, for less-sophisticated engines. This might well be made somewhat larger if the engine is to be used only at 600 rpm. If it is expected to operate much of the time at some lower speed as in oil-well pumping it is preferable to design the ports for the mean speed as very large exhaust ports are a source of loss both in "dead stroke" and charging.

The diagram, Fig. 92, shows the limitations of the ordinary system of porting. Exhaust must be completed before scavenging begins so the exhaust function is superimposed on that of charging. On the return stroke the time between the closure of the inlet ports by the piston and the closure of the exhaust ports is wasted.

It is desirable and even necessary in large engines, since possible port capacities grow only as the square of the cylinder diameters and volumes as the cube, to prolong exhausting toward dead center and to use an equal period on the return stroke for charging. This is accomplished notably by the Sulzer system, Fig. 4, in which port *a* is piston-controlled while port *b* is valved, opening and discharging after the cylinder pressure has fallen to nearly that of the atmosphere. This auxiliary port *b* may prolong charging until the exhaust port is closed or it may be used for supercharging.

THE EXHAUST PIPE

Exhaust pipe pressure waves have been known to exert an influence on four-cycle engines, but their effect is of such moment in two-cycle engine operation that the exhaust piping system is a problem as serious as any in the design of the engine itself.

FIG. 93

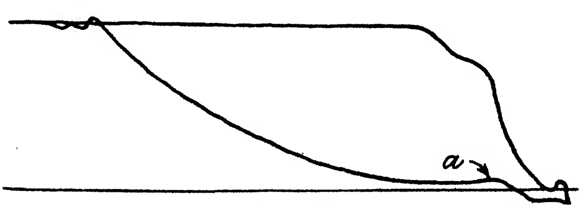


FIG. 94

An exhaust pipe pressure diagram is shown in Fig. 93, a light spring card from the power cylinder in Fig. 94. Steady back pressure usually suspected as the cause of all exhaust system evils is lacking. Discharge from the cylinder builds up a pressure wave to about 10 lb per sq in. This seeks outlet, pulling a vacuum in its wake. A return wave from the atmosphere fills this vacuum, and its kinetic energy builds up pressure again at the ports and within the cylinder.

The period of these waves is a straight-line function of the length of the exhaust pipe.

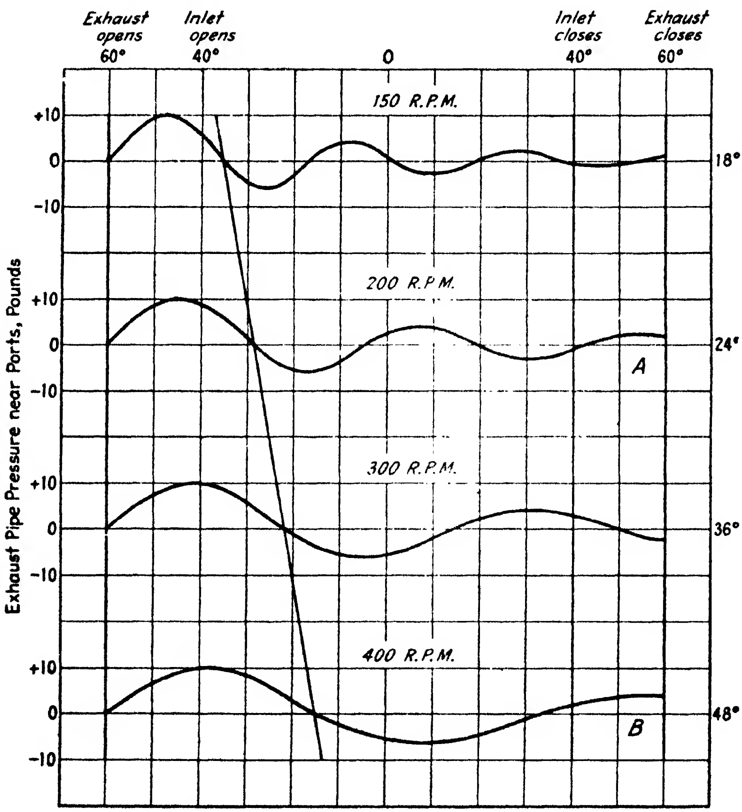


FIG. 95

Careful indicating of the exhaust pipe at various engine speeds as plotted in Fig. 95 seems to prove that the best advantage to be secured from the unavoidable pulsations is realized when charging is completed against a return pressure wave, restoring the cylinder pressure to atmosphere, following which the wave tends to supercharge the cylinder during the interval after inlet port closure and before exhaust port closure.

The wave effect is unfortunate when scavenging is completed during an outward pulsation associated with a sub-atmospheric

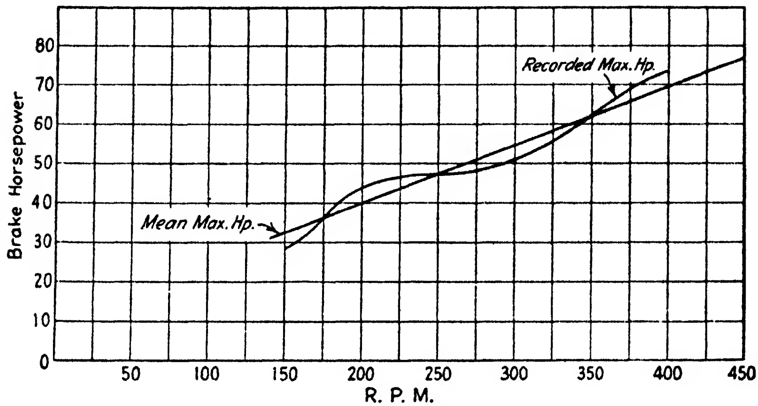


FIG. 96

pressure at the ports during the period after the inlet ports are closed, but before exhaust closure.

Thus it is evident that for each speed there is one correct exhaust pipe length which results in one vacuum phase and one return pressure wave tending to supercharge the cylinder from the exhaust ports as they are closed by the piston as in Fig. 95*B*. By reason of two complete pipe wave cycles completed in the same time this length is favorable also at half speed as in Fig. 95*A*, or if the environment demand it the half length of pipe may be used, giving two short-frequency waves again like Fig. 95*A* during the period of port functioning rather than one but terminating in either case in pressure rather than vacuum against the ports as they are closed.

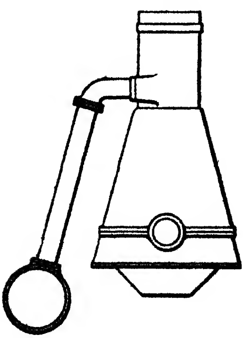


FIG. 97

This action results for gas engines in a sinuous horsepower curve (Fig. 96).

Multicylinder engines once were piped as in Fig. 97 in respect of this phenomenon. This system was not appreciated, being bulky and cumbersome, and had to give way to manifolding, Fig. 4. The manifold being equal in diameter to the cylinder damps out all waves. Rather larger ports are required as the flow acceleration due to vacuum is lost.

A single pipe of half cylinder diameter for each cylinder is used universally for horizontal engines. The correct length approximates $2500/N$ ft, where N equals the speed in revolutions per minute or half or twice that length. This pipe should discharge into a great receiver which may be common to all cylinders. Its volume should be about fifty times that of any one cylinder. The receiver discharges through a stack of a diameter equal to that of the cylinder.

OPPOSED PISTON ENGINES

The revival of interest in the opposed piston engine introduces one possibility of saving dead stroke or of supercharging. The cranks operating the opposed pistons are not set 180° apart but are 10 or more degrees offset with the exhaust function leading thus avoiding that spill and loss of compression stroke caused by the closure of inlet ports before the exhaust ports are sealed. Extreme displacement of cranks has not proved profitable on account of power loss when the torques of the cranks are opposed about the firing center.

VALVED EXHAUST FOR TWO-CYCLE ENGINES

Engines with piston valves in the cylinder head for exhaust are merely practical modifications of opposed piston engines.

Engines with poppet valve exhaust using ported and piston-controlled inlet have the advantages of exhaust valve timing independent of piston movement in the highest degree. Since the period of valve movement is far shorter than in four-cycle engines, acceleration forces are high and the duty on the valve gear is heavy. Careful kinetic analysis is essential. The valve sizes may be determined from a curve of valve displacement just as port sizes are determined from the curve of piston movement.

CHAPTER IX

INJECTION OF GASEOUS FUEL

Since the gas engine has been introduced into this volume as the most likely means of studying the actual efficiency of scavenging it seems desirable to include a chapter on this subject. Altogether aside from its use as a means for developing two-cycle diesel engines, the performance of the injection type of gas engine throws some light on the subject of liquid fuel injection.

The commercial success of the Otto engine which was patented in 1872 set men to prospecting for other methods of working without awaiting the expiration of the Otto patent. This led to a series of two-cycle engine inventions beginning with the Alexander Rider patent 245,218 filed May 12, 1881, and patented Aug. 2, 1881. Claim L of this patent specifies:

> A gas engine provided with a motive cylinder constructed to exhaust the burned air and air and gas supply pistons and cylinders, the parts being constructed and arranged to cause the charge of air entering the motive cylinder to rapidly expel the last portion of the air therefrom, after which the air is confined in the motive cylinder and the charge of gas then admitted after the exhaust opening has been closed, substantially as set forth.

Engines of this sort are classified by the late James T. Allen in his "Digest of U. S. Patents of Air, Gas and Calorie Engines" as follows:

Subclass — Two-Cycle, Separating Air and Gas Pumps

> Two-cycle recompression internal-combustion engines in which air and combustible in the gaseous form are supplied to the working cylinder or combustion-chamber by separate and distinct pumps, each operated by and in unison with the engine, one at least of the constituents being recompressed within the working cylinder before the ignition of the combustible mixture. The air is ordinarily supplied to the working cylinder of the engine before the gas, and such air

> is frequently designed to more or less completely scavenge the working cylinder. In such cases a part of the air thus supplied remains in the cylinder to form with the gas subsequently supplied thereto the next following charge. The air and gas may, however, enter the working cylinder simultaneously. The air and gas are ordinarily intermixed within the working cylinder before the completion of the compression-stroke, so that the complete charge is recompressed in the working cylinder before the ignition thereof and the beginning of the working stroke; but this subclass also includes engines in which one only of the constituents of the charge is recompressed in the working cylinder, in which case the other is supplied thereto under pressure during the working stroke.

From 1883 to 1905 a number of patents were placed under this classification. The following is a copy of one claim of the first Baldwin patent.

PATENT 276,750. CYRUS W. BALDWIN. Filed Jan. 5, 1883. Pat. May 1, 1883.

CLAIM 5

> The combination of the cylinder, its exhaust port, piston, and air inlet valve and gas pump all arranged as set forth to permit the air to traverse the combustion chamber as the piston completes its forward movement to compress the air and force a measured charge of gas into the latter as the piston returns, and to condense the combined gases on such return movement of the piston substantially as set forth.

Fig. 98 shows a section of the Baldwin engine.

Engines charged by simultaneous air and fuel induction whether into the scavenge pump or the power cylinder had superseded designs with air pumps and gas pumps so completely that the original commercial history of the latter is totally obscured.

This is due to the fact that for many years two situations reigned in the gas engine field. Industrial centers were supplied with manufactured gas or carefully conserved natural gas at prices so high that only the most efficient four-cycle types survived.

In oil fields and adjacent territory natural gas was available as a by-product at prices as low as 5 cents per thousand cubic feet. It cost the oil operator little or nothing at all. It had no market value, and its use, even wastefully, spared him the criticism of conservationists. For oil country work then very primitive types of two-cycle engines served well on account of their simplicity and reliability.

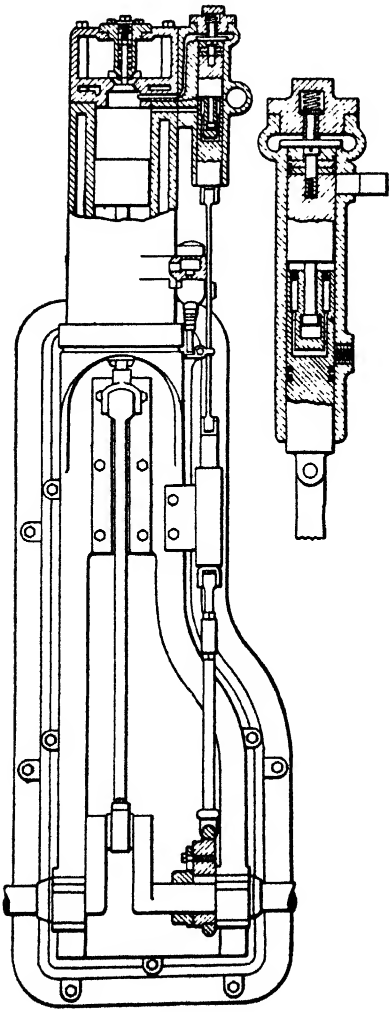


FIG. 98

Gas, however, became a valuable commodity as capital was found to build great pipe lines and as salt-water encroachment, water drive and air repressuring in remote fields threatened the gas supply. Economies were sought, and gas injection was reinvented as a measure to allow increased compression and to avoid port loss in two-cycle engines which previously had been scavenged with the gas-air mixture used for charging.

Two schools of thought developed. One followed Diesel hoping to succeed where he had failed, which was not illogical as his solid injection experiments were later refuted. The other, perhaps little informed as to the ideas of Rider and Baldwin which antedated theirs, proposed only to obviate port loss by scavenging with pure air and injecting gas after the cylinder had been isolated from the atmosphere.

GASEOUS FUEL IN DIESEL ENGINES

The Rider and the Baldwin engines had been forgotten if they ever had any success. The more recent experiments of Diesel (1894–7) were available in printed form as well as the comments by Lüders.*

Diesel, despairing of injecting liquid fuel satisfactorily, had turned to gaseous fuel. He learned that the entering gas stream was not immediately dissipated through the air but rather "blew"

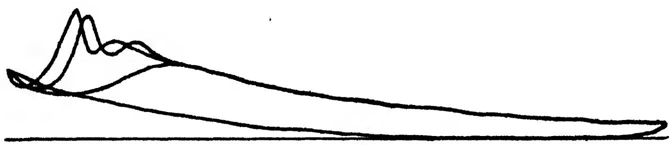


FIG. 99

the air out of the way, creating a space for itself. This resulted in erratic performance — some good cards, many misfires and many cards like Fig. 99. Gas retained in the charge from previous misfires pre-ignited by heat of compression producing terrific explosions dangerous to machine and operator.

Oil introduced with the gas made ignition certain but this added complications. Introducing gas through fine multiple-orifice nozzles helped much, but even then after-burning continued to the middle of the power stroke because of poor mixing.

Air injection of the gaseous fuel produced better results, but this required two high-pressure compressors, an unwarranted complication in a day when even one injection compressor was a serious problem.

The governing of the gas charges at a pressure of 60 atmospheres or more introduced serious difficulties, and as can be imagined leakage of fuel valves under the high pressure brought in a real and ever-present hazard.

HIGH-COMPRESSION GAS ENGINES WITH FUEL INJECTION AND ELECTRIC IGNITION

Later diesel gas engines duplicated the inventor's experience although more leisurely research developed the fact that gas introduced toward the end of the compression stroke may be fired

* "Der Diesel Mythus."

successfully by an ignition plug if a definite time interval between the termination of injection and spark production is allowed for mixing by turbulence and diffusion.

This interval is defined to the thousandth of a second, a phenomenon of great interest in the laboratory but utterly impracticable in commercial operation. If the spark is too early, misfires result. If the engine is speeded up with set spark, misfires again result because of reduced mixing time unless the spark is *retarded* at a rate proportional to the speed. The ignition range is limited to so few degrees before dead center since the gas must be introduced late to avoid spontaneous ignition and the rate of flame propagation is so low owing to indifferent mixing that cards like Fig. 99 develop.

These unfortunate properties recently have caused a second abandonment of the high-compression injection gas engine. Seemingly it should be a perfect type and it may yet be, but the explosive phenomena of vaporization and dissociation of liquid fuels result in a rate of mixing as yet unattainable with the simpler gaseous fuels.

OTTO-CYCLE GAS ENGINES WITH FUEL INJECTION

The ideas of men who had been concerned with low-compression engines centered on improving the constant-volume type of two-cycle engine. Injection of fuel immediately after closure of the exhaust ports as recommended by Rider was proposed again in order to allow the greatest possible time interval for mixing. Engines of this sort were an almost immediate commercial success.

In gas compressing stations, gas at the injection pressure of 15 to 20 lb is available through reducing valves and no special compressor is required. Fig. 100 shows a compressor with "economizer" or gas admission valve in the cylinder head.

For commercial use when operating on gas metered to consumers at 4-oz pressure an engine-driven gas pump as in Fig. 101 is required.

It is quite possible to realize the slogan of "two-cycle engines with four-cycle economy" if the scavenging system is designed to operate at the lowest possible pressure. With the rather high scavenging pressure ruling in two-cycle gas engine designs the fuel consumption is slightly greater than that of four-cycle en-

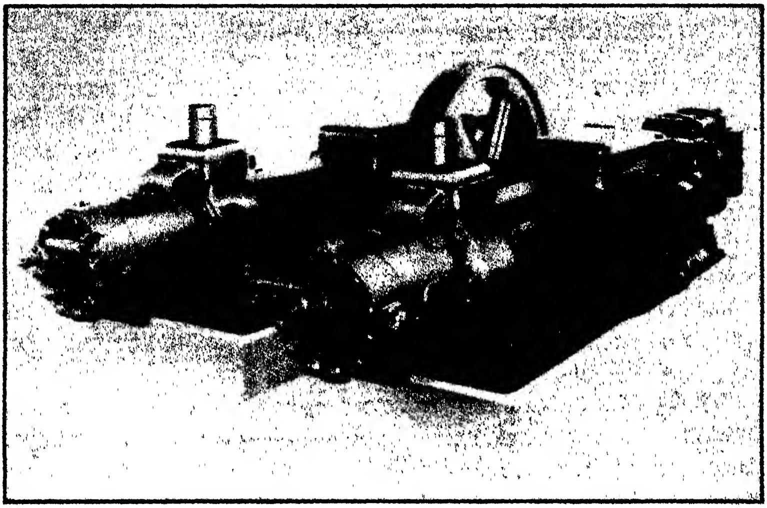


FIG. 100

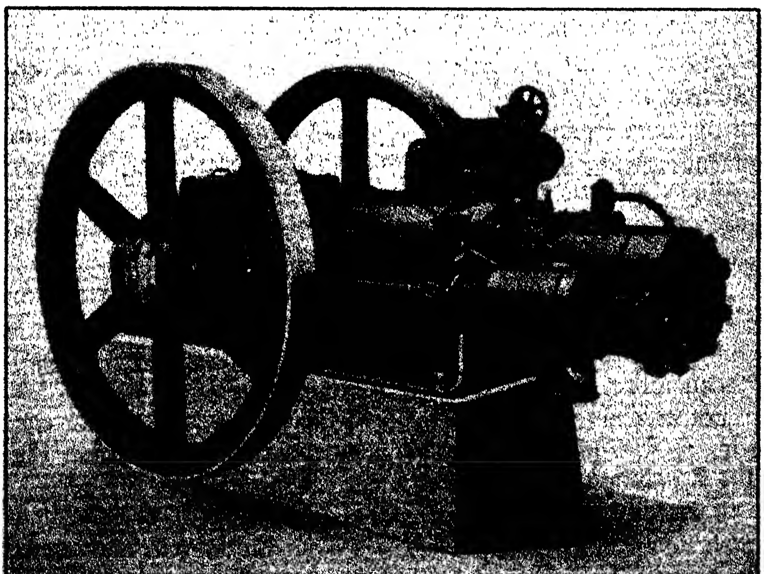
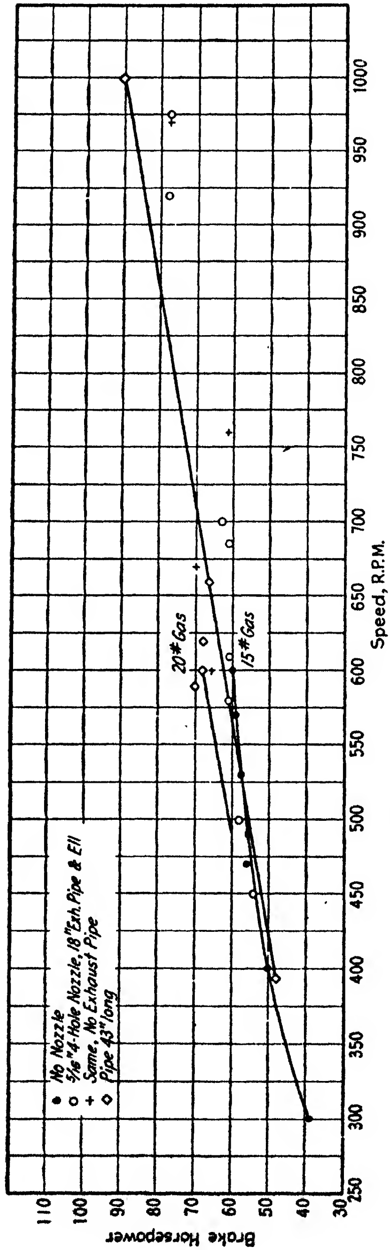


FIG. 101



• No Nozzle
○ 5/16" 4-Hole Nozzle, 18" Exh. Pipe & Ell
+ Same, No Exhaust Pipe
◇ Pipe 43" long
20# Gas
15# Gas
Brake Horsepower
110
100
90
80
70
60
50
40
30
250
300
350
400
450
500
550
600
650
700
750
800
850
900
950
1000
Speed, R.P.M.

FIG. 102

gines of the same size and compression but the difference probably is offset by the losses in four-cycle engines due to the average condition of the valves.

Adoption of forced scavenging improves the engine output, and the injection of gas after port closure supercharges about 8 per cent resulting in increased horsepower and increased speed range as shown in Fig. 102.

THE GAS NOZZLE

In most commercial designs gas is injected through an inwardly opening poppet having the considerable lift of $\frac{1}{4}$ to $\frac{3}{8}$ in. The poppet is actuated by a cam or eccentric when station pressure is used for injection as in Fig. 100. It is automatic when the

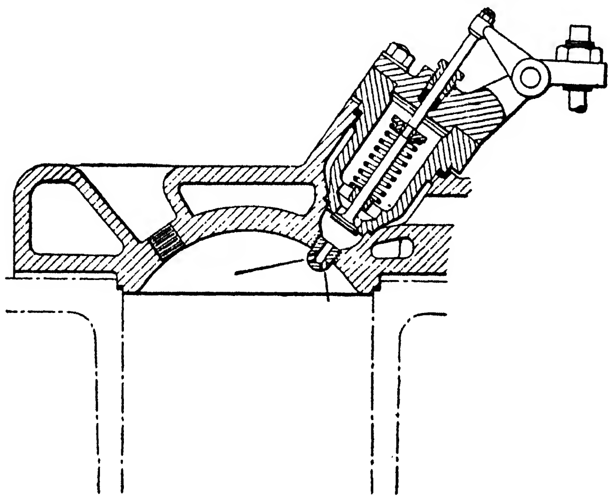


FIG. 103

engine is fitted with a pump for auto injection, Fig. 101. These simple nozzles do very well for slow-moving engines (200 rpm). At higher speeds when less time for mixing is available it is necessary to use drilled nozzles, Fig. 103, or to limit the lift of the valve so that the gas is jetted through the seat. Much experimental work will be required to put gas injection apparatus on the same sound design basis as that occupied by oil fuel nozzles.

TIME REQUIRED FOR MIXING

When a simple high lift gas admission valve is used without any special form of nozzle, about $\frac{1}{20}$ sec. seems to be required for adequate mixing. Allowing 60° for port closure and 30° for ignition advance, 90° or 1/4 revolution is available for diffusion of the gas through the air. This 1/4 revolution must be completed in $\frac{1}{20}$ sec. or 1 revolution in $\frac{1}{5}$ sec. This limits the engine speed to $5 \times 60 = 300$ rpm. Jetting the gas through coarse nozzles reduces the mixing time to $\frac{1}{40}$ sec allowing a speed of 600 rpm.

It also appears that when the exhaust piping system is favorable gas may be admitted after closure of the inlet ports with little or no loss, extending the time of mixing. Experiments have been conducted to 1000 rpm with jetted gas, and skillfully designed turbulence effects are due to make further advances.

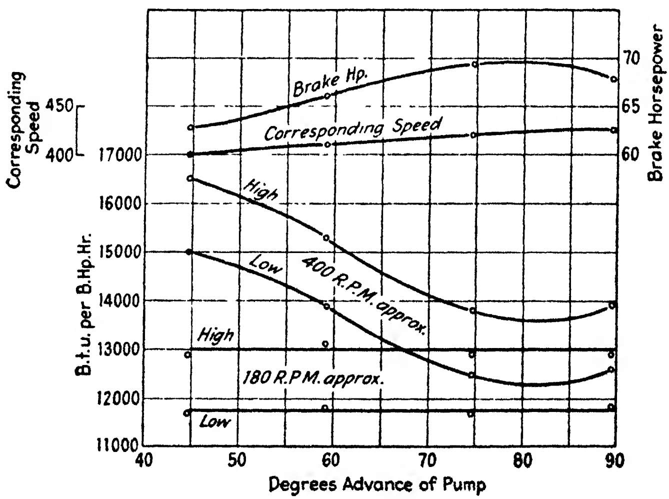


FIG. 104

Fig. 104 shows the effect of advancing the injection pump crank of an engine at 180 and at 400 rpm. No change in fuel consumption is noted at the lower speed. At 400 rpm the best position allowing sufficient mixing time is decidedly marked. These tests were for a rather inefficient example as the scavenge pressure was abnormally high and the injection pump not of the best design.

An electrical contact on the automatic admission valve by recording on the indicator drum showed that gas entered a few degrees after the dead center.

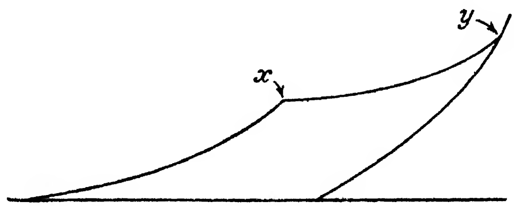


FIG. 105

Fig. 105 shows an injection pump card from this engine. Admission begins at x and ends at y. The relation between main and pump cylinder clearance determines the point $= y$.

THE GAS PRESSURE

The maximum injection pressure used with commercial designs is about 30 lb gage. On refinery gas or any gas of very high heating value 15 to 20 lb gage seems to be very practicable.

If the gas pressure is so chosen that the critical velocity of flow rules the discharge from valve or nozzle the rate of injection is governed by the injection valve closure rather than the rising compression pressure within the cylinder and the condition of minimum nozzle size or least possible injection interval is attained.

INSTABILITY AT PEAK CAPACITY

When the engine with fueling by gas injection is operating at its maximum capacity any slight additional load or upward fluctuation of gas pressure due to governor or pressure regulator action causes more gas to be admitted enriching the charge so much that weak impulses or complete failure of ignition ensues.

The immediate result is further opening of the governor valve owing to loss of speed, and the engine stops unless an attendant is near by to reach the throttle in time. For this reason at least 10 per cent overload capacity should be allowed when establishing the rating.

ENGINES WITH LIQUID FUEL INJECTION AND ELECTRIC IGNITION

A number of attempts has been made to revive the use of spark ignition for low-compression oil engines fueled by injection into the power cylinder. These engines exhibit some of the characteristics of the gas engine with fuel injection and in other ways are more like the diesel engine.

A very definite interval of time must be allowed to elapse between the completion of injection and the passage of the spark or ignition will fail. Advancing or retarding the spark a very few degrees from the position of best operation but still within the firing range brings about a striking change of performance.

Early firing causes black smoke to issue from the exhaust on account of the isolation of the fuel in a sheath of dense mixture or products of combustion. Correct firing results in clear exhaust. Later firing results in white clouds of unburned vapor passing out of the exhaust. Fuel, igniting late, is not consumed while the higher temperatures reign and naturally fails to burn as expansion lowers the temperature.

These phenomena confirm the deductions in Chapter II regarding the necessity of the delay between the beginning of injection and ignition.

Such engines are thought to be promising in the aircraft field. For other work they are as yet of little use. The spark gap either must be in the path of the spray or the mist eddying near it, or the sprayed fuel must be brought to the spark by turbulence as in the Hesselman engine. Consequently, plugs require frequent cleaning when heavy oil is burned.

Often the line of demarkation between spark-ignition engines and those igniting spontaneously by reason of carbon deposits on the antennas of the plug is very narrow. The same engine may auto-ignite continuously or with a slight change of condition require ignition apparatus in working order at all times.

It is quite feasible and perhaps desirable to develop these systems for burning selected fuel distillates heavier than gasoline. For the run of market fuel oils the high-compression auto-ignition engine is infinitely more satisfactory.

There is still opportunity for research and invention in this field. Derangement of the nozzle valve when burning gasoline is fatal to cylinder lubrication; consequently, extreme care must be used for designing this detail.

CHAPTER X

BEARINGS AND LUBRICATION — PISTONS AND PISTON RINGS

THE CRANKSHAFT BEARINGS

The main journals of the crankshaft are carried in concentric shells, lined with babbitt. These shells rest in bored seats in the bed plate. With this construction it is possible to roll the bottom shell out after the cap half is removed without disturbing other bearings. The shells are prevented from turning in place during operation by the doweling of the top half to the cap.

Older engines were fitted with brass shells, babbitt lined. Brass is preferable as it tins perfectly causing the babbitt to adhere as a unit when cast in place. Its coefficient of expansion is nearer to that of the high-tin babbitts so that the junction is under less strain than when steel backing is used. Necessary economy in construction, however, has forced brass out and forged or cast steel has largely replaced it. Cast iron was once much used but it neither tins nor electroplates well enough to be reliable. Needless to say a shell is not dependable if the babbitt is not perfectly soldered to it in the pouring process. A shell, to pass inspection, should ring clearly when sounded by striking it lightly.

Small steel shells are made from strip in a die. Large forged shells usually are made by trepanning a bar. The resulting tube is bored inside, set up on a centrifugal casting machine while hot, the babbitt being poured as the shell revolves at a high rate of speed. The centrifugal force causes the babbitt to line the shell which is allowed to spin until the metal congeals. Shells so lined are free from air bubbles and dross as the dense metal seeks the outside, the inner portion being bored away.

The next operations after boring the babbitt are the turning and facing of the shell, after which it is cut in two with a saw somewhat narrower than the desired shim space. This is to provide finish for a milling operation on the joint to true it up. The saw curf itself is rarely free from wind.

Grooving is done most uniformly and cheaply on the milling machine. A very satisfactory form of transverse groove is that shown in Fig. 106. The older method of relieving bearings for 20 to 30° each side of the joint is satisfactory for purely normal loading but the centrifugal force of the crankpins and arms throws some side loading on the bearings which is better carried on the

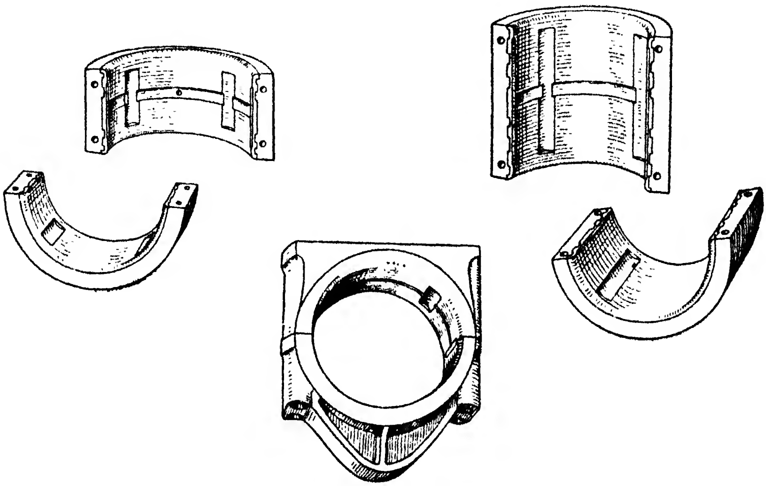


FIG. 106

lands left between the joint and groove. The annular groove puts the oil lead from the lubricating oil main in communication with the hole in the crankshaft which carries oil to the crankpin. Needless to say, shell joints and transverse grooves should have feather edges to prevent scraping oil off the journal and thus defeating the purpose of the groove. All oil passages should be of quite generous proportions.

The thickness of large shells should be about one-sixth the shaft diameter or more. The shims at the joint should clamp the shells together to prevent them from working and curling inward against the shaft. This latter action is always possible with loose shells after slight heating and greatly aggravates that trouble.

It is very difficult to machine the joints so that they match exactly, hence it is well to relieve the surface in bed and cap as in Fig. 107*a*. The shims are sufficiently flexible to accommodate

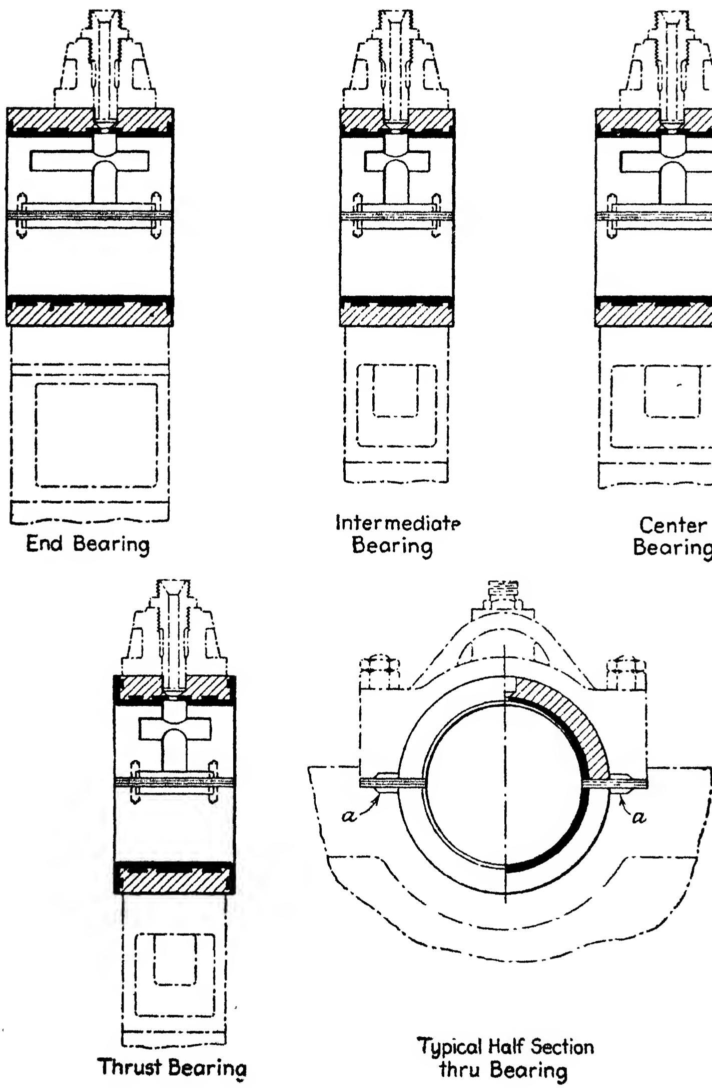

End Bearing
Intermediate Bearing
Center Bearing
Thrust Bearing
a
a
Typical Half Section thru Bearing

FIG. 107

themselves to the slight difference of levels. The shells are clamped firmly in place by the cap, and the cap itself is relieved of bending which would result in its closing the shell in. If heavy self-supporting shells cannot be used on account of weight limitations it is then best to go to the other extreme and use quite thin ones (about $\frac{1}{16}$ diameter). Thin shells cannot put any appreciable elastic pressure on the shaft if deformed by pounding or heat, but even they are best secured by riveting in place as their curling interferes with lubrication and may be responsible for piston slap.

Whether or not it is advisable to machine anchorage in the backing is questionable. If the tinning process is of that excellence practised by the specialists in the business it is not necessary. For the average shop and particularly for the repair shop it is safer to do so. The slots may be turned and broached, and the end facings should be dovetailed.

The cap should be designed as a free beam loaded with the inertia force of the reciprocating parts and the centrifugal force of the crank and rotating end of the rod. An assumption of 200 lb per sq in. of piston area will satisfy most cases for the upward load. For the bending moment it is customary to use the formula

$$M_b = \tfrac{1}{6} Wl$$
$$l = \text{distance between studs}$$

This results in a value midway between that for a concentrated load at the center and that for uniform loading. On these favorable assumptions the stress in semi-steel caps should not exceed 4000 lb per sq in. although it is relieved by the clamping action of the studs.

Shells are rarely flanged, dependence being placed on the caps to hold them in place. This removes the necessity of facing the seats in the bed, a costly and rather useless procedure. One-eighth to one-fourth inch lateral clearance is allowed between crankarms and shells to permit the shaft to shift endways from expansion, deflection or the wear of marine thrust bearings. These latter locate the shaft endwise in marine engines. In stationary engines one shell is babbitt faced and closely fitted between a pair of crankarms or between the after-crankarm and a collar turned on the shaft to serve as thrust collar and oil thrower. These faces should be generous if the engine is to be used as an auxiliary in a ship or on rolling equipment where it may be required to operate when tipped 30°, more or less.

THE CRANKPIN BOX

This usually is a babbitt-lined steel casting, Fig. 106, which has dovetailed babbitt facings since it fits closely between the collars turned on the crankarms. It is grooved like the main bearing shells, and its cap is designed in the same way except that for steel the stress may run to 6000 lb per sq in. without undue deflection. Two-cycle engines of the single-acting type do not subject the crankshaft or crankpin bearing caps to any loading since compression exceeds inertia, but it is well to design them on the same basis since a cylinder may be required to operate with fuel nozzle or valve removed in time of emergency.

It should be possible to remove the upper crankpin box by hoisting out through the cylinder bore while attached to piston and rod or at least to remove the rod, which usually has the same diameter at the foot as the box. This often makes it necessary to use four small bolts through the box rather than two large ones and in a measure determines the bolt stress and material. Except for very high-speed light-weight engines heat-treated acid open-hearth carbon-steel bolts are quite satisfactory if drawn to 1200°F after oil quenching. They should not be heat-treated for high tensile strength but rather for high elongation and reduction of area. It is well to keep the stress down to 6000 lb per sq in. although it may be necessary to exceed this for want of room.

The bolts should be roughed out and heat-treated. After this they may be ground and the threads milled. Except where necessary for centering the boxes, bolts should be relieved to a diameter slightly less than that of the thread root to prevent concentration of stress in the threads. Fine threads to the S.A.E. schedule are in common use.

Rods for small high-speed engines often are drop-forged and fitted with shells. This construction may be required to save weight and expense if the production warrants the die cost. It is not so popular for large engines as operators sometimes find it convenient to put a shim between the rod foot and box to raise the compression of a worn engine to avoid ignition failure when starting.

In order to keep the crankshaft short and stiff, crankpin box and intermediate crankshaft bearings, Fig. 106, are often made the same length. The crankpin box may be designed for a unit pressure of 1500 lb per sq in. of projected area based on 600 lb per sq in. piston pressure. When the piston pressure exceeds 600 lb in the case of high-speed engines the unit bearing pressure will increase in direct ratio. If an attempt is made to use low unit pressure in high-speed engines the shaft will lack torsional stiffness due to excessive length and the rotating weight will be too great. It is very desirable not to exceed 600 lb maximum pressure in any engine.

The end, center and intermediate crankshaft bearings, Fig. 107, are sometimes made to different lengths to compensate for their different loadings. As this loading is compounded of cylinder pressures, centrifugal forces and inertia — all of which are variable with speed and mep — it is quite doubtful whether an adequate solution of the problem can be found. It is perhaps just as well to make them of a single length except the flywheel end bearing and center bearing which may be made 50 per cent longer than the intermediate bearings.

In general practise the tin-base bearing metals are the most satisfactory lining.

HARD BEARING ALLOYS

High-speed four-cycle engines tend to pound out babbitt bearings if not carefully designed and maintained. This has caused the return of leaded bronze and copper lead mixtures for shells and linings. Needless to say these alloys should not be loaded more heavily than babbitt metal. After all, the permissible unit loading is a matter of the endurance of the oil film rather than the metal.

The leaded bronzes will not melt out, but the lead sweats out under excessive heat and the rate of crankshaft wear is intolerable.

THE PISTON PIN AND ITS BEARING

Hollow pins of S.A.E. 1020 steel, case-hardened, are the rule. The pins are given the usual double quench and draw H.T. S.A.E. 1020 V but should be reasonably hard (Rockwell C 55 to

60). They are always ground and preferably lapped after hardening.

Some theories have been put forward regarding the deformation of the pin from cylindrical shape under load. These neglect the shape of the oil film pressure curve and are of somewhat doubtful utility. If very light pins are required experiment is the best guide. Deformed bearing surfaces will show high burnished spots. For general practise the pin bore should not exceed two-thirds of its diameter.

Very little grooving is required for four-cycle engine wrist-pin bushings. An annular groove at the center may be provided to carry the oil around the pin. Transverse feeders are sometimes useful when pins are very snugly fitted. The grooves should be chamfered carefully to prevent their scraping the oil film of the pin. Two-cycle single-acting engines require more grooving as the load is constantly downward and the oil must be wiped over the surfaces by the motion of the bushing. In this case the lower half of the bushing usually carries straight transverse grooves about 30° apart in addition to the annular groove around the bushing at the center. Oil is transmitted to the bushing from the crankpin supply through the bored connecting rod, preferably, or through a tube secured alongside the rod. If the rod bore is large to reduce weight a tube is centered inside by washers welded in place at intervals (Fig. 37) to reduce the amount of oil required to fill the system when starting.

The bushing usually is made of phosphor bronze S.A.E. 65, preferably chill cast.

The piston pin of either the fixed or "floating" variety may have a diameter equal to $1/2\,D$. When the pin is fixed in the piston the length of the bushing in the rod is fixed by the allowable unit pressure. Twenty-four hundred pounds per square inch is a satisfactory value. If, as it usually is, the maximum cylinder pressure is 600 lb per sq in., the projected area of the bushing is

$$\frac{\pi}{4}D^2 \times \frac{600}{2400} = \frac{\pi}{16}D^2$$

Since the diameter equals $1/2\,D$ the length of bushing is $(\pi/8)\,D$. Much side clearance is allowed to avoid the disagreeable "slap" of a loose rod against the bosses and to allow for misplacement of the crankshaft thrust bearing due to wear or accident. This clearance may be made 1/64 D on each side.

For two-cycle engines the construction shown in Fig. 108 is often preferred. The continuous nature of the loading of two-cycle wristpins makes a lower mean unit pressure advisable if not absolutely necessary. The projected area of the wristpin bearing may equal that of the crankpin bearing if possible.

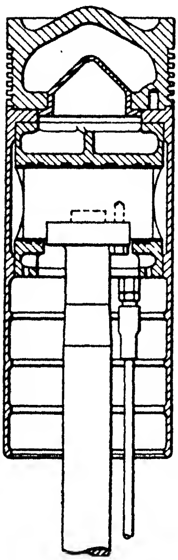


FIG. 108

When aluminum-alloy pistons are used it is not considered practicable to secure the piston pin in the piston on account of the greatly differing coefficients of expansion of the two metals. It is true that pins are successfully inserted into and removed from smaller pistons by gently heating the piston in boiling water, but this light shrink, loosening at operating heat, is not secure for diesel engine temperatures and pressures. Even if it were secure the design would not be desirable. Being restrained along one diameter by steel having a coefficient of expansion of about one-quarter that of aluminum the piston barrel assumes an oval form and seizes unless given an undesirable amount of clearance.

When the piston pin is clamped in the rod end, the piston bosses are bushed, with the piston alley for aluminum pistons and with bronze for cast-iron pistons either of which under good forced lubrication may carry 2400 lb per sq in. of projected area in four-cycle engines.

The floating pin free in both rod and piston is often fitted with a pilot at each end made as shown in Fig. 109 with a slotted steel shank cast into aluminum-alloy heads turned to the radius of the cylinder bore. The pilots slip into the bore of the pin and prevent it from coming into contact with the liner. The pilots seem to wear well and have no ill effect on the cylinder bore.

The full floating pin suffers from combustion gases and burned cylinder oil unless a complete closure like Fig. 110 is fitted. With a closure, it has some very desirable features. It obviates bolts in the small end of the rod. The piston pin wears round. Repair is made by rescraping to an over-size pin rather than fitting and aligning new bushings and pins. The headroom required to remove piston and rod is reduced since the pin may be slipped out as soon as it appears above the cylinder head studs. Burned cylinder oil cannot reach the pin bushings, nor can sulphurous acid, a product of condensed water of combustion and sulphur anhydride. Oil from the pressure system supplying the rod is

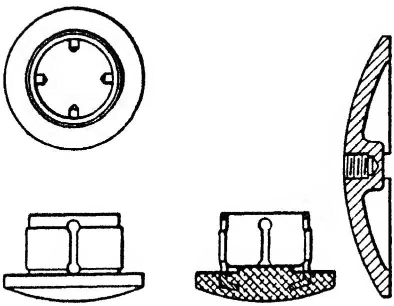


FIG. 109

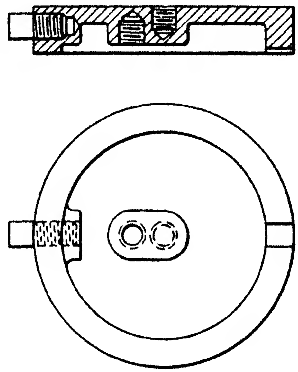


FIG. 110

kept from the cylinder bore, and danger of a scored cylinder due to a loose pin is entirely absent.

It is necessary to carry a unit pressure of 3000 lb per sq in. on the bushing in the small end of the connecting rod when the floating pin is used, on account of space limitations. Hard phosphor bronze (S.A.E. No. 65) sustains this very well as the pin turns in the rod at reduced velocity, if at all, during the heavier pressure intervals owing to the lesser friction of the piston bushings. Chilled phosphor bronze of the same composition is even more desirable, and case-hardened steel ground and lapped has its advocates.

NEEDLE BEARINGS

For two-cycle engines the needle-bearing wristpin has been in successful use for some years. The pin is designed to essentially the same proportions as for bronze bushings. It is surrounded by 2 or 3 rows of short needle rollers $\frac{1}{8}$ to $\frac{3}{16}$ in. in diameter running in a hardened and ground bushing in the rod eye.

THE CONNECTING ROD AS A PUMP

The column of oil in the rod is subjected to inertia forces due to the acceleration and retardation characteristic of crank motion. During the upper or inner half stroke the oil is thrown violently into the wristpin bushing grooves. During the lower or outer half stroke the oil column sets up a pressure counter to that in the lubricating oil main and that due to centrifugal force on the column of oil in the passage from main journal to crankpin. The system can be made to pump oil by grooving only the upper half of the crankpin box as in Fig. 106. When the crank is in the lower half of its travel there is then no communication between the rod bore and the oil passage in the crankpin and arm so back-flow is prevented. Communication is effected when the crankarm passes the 90° position and oil is pumped through, assisted by its own inertia.

THE PISTON BARREL OR SKIRT

The length of piston skirt below the ring belt varies from D to $2D$. The question of unit bearing pressure under the connecting rod thrust is not important. Pistons are made long mainly to avoid "slap" and to remove the piston pin from the heat zone. In the case of the Butler piston, Fig. 10, they are made long to reduce the bending of the struts between crown and bosses. Pistons are made short to reduce weight, a necessary measure for high speeds and to reduce the over-all height and weight of engine. It is scarcely necessary to make piston skirts longer than $1.5\,D$ and they should rarely be shorter than $1.0\,D$.

The skirt thickness usually tapers from top to bottom; $1/32\,D$ is a common value for thickness at the top if lightness is required, $0.04\,D$ is more common for engines of moderate speed. The thickness at the lower or outer end of skirt usually is three-fourths that of the upper and inner end. The open end often is reinforced with a bored flange, Fig. 10, to give strength and to provide for centering on lathe and other machine tools.

The above expressions are not strictly rational but very nearly so. Their variation from rational form is in the right direction, the smaller pistons carrying unit compressive loads of about 5000 lb per sq in. based on 600 lb per sq in. maximum cylinder pressure. Deflection under thrust is provided for by the rigidity of upper and lower ring belts and other circumferential ribs.

It is difficult to give a rational expression for the piston pin bosses. They usually are made eccentric, and for cast-iron pistons with pins of diameter equal to $1/2\,D$ a thickness of boss equal to $1/8\,D$ on the top and $1/12\,D$ on the bottom seems to suffice if stayed by a single rib to the crown and well filleted to the skirt. One or two circumferential ribs should pass from boss to boss. A diaphragm should be provided in the ring belt to prevent contact of the lubricating oil with the under side of the piston

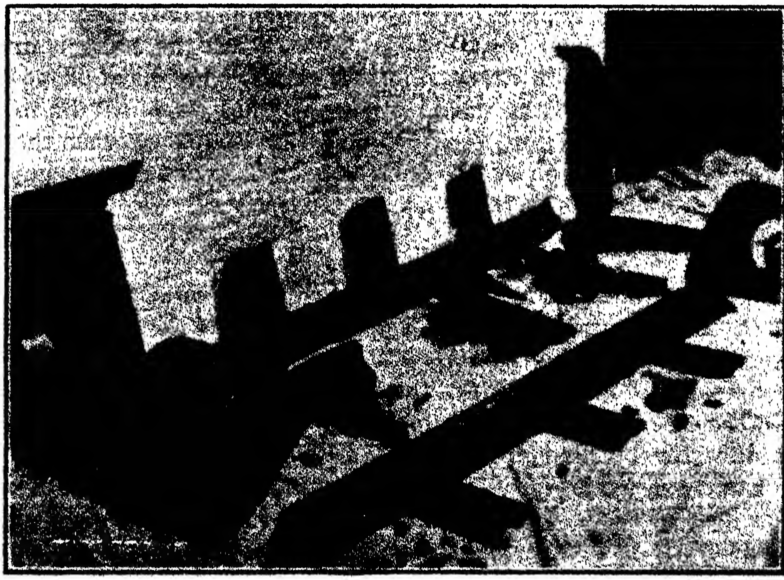


Fig. 111

crown. This avoids deposits of coked oil in the lubricating system. More important, if possible, is the avoidance of crank-case explosions due to spontaneous combustion of the lubricating oil. The result of such an explosion is shown in Fig. 111. The necessary core outlet is sometimes closed with a pipe plug or a bolted plate. Both have been known to come adrift with alarming frequency. A spun or drawn cup, rolled in place and electrically welded, is far safer. Steel cups are used with cast-iron pistons and aluminum sheet cups with aluminum-alloy pistons. The latter really do not require a diaphragm as they cannot be operated at a temperature sufficient either to ignite oil spontaneously or to crack oil.

SPECIAL ALUMINUM-ALLOY PISTON DESIGNS

The necessity of light pistons for high speeds has resulted in a number of special designs of aluminum-alloy pistons. Most of these are patented, and the claims are concerned with means of reducing the "slap" due to the greater clearance required by the alloys on account of their high coefficient of expansion.

Automotive pistons of the spring skirt type have found no favor in diesel engine service. The split or spring side of the skirt sometimes takes a set and becomes ineffective.

The Butler piston, Fig. 10, is quite successful. It is described in the following patent claim 1 from patent 1,532,121 issued to Chas. Butler, April 7, 1925:

> A metal piston having a skirt which is divided into a plurality of separated segments, and one or more continuous rings of metal (steel) having a smaller coefficient of expansion than that of the metal of which the piston is made, said rings being located in the wall of the said skirt with piston metal against them both radially inward and radially outward thereof, there being one such ring near the head of the skirt.

It is quite usual to make aluminum-alloy pistons to the same general dimensions as cast-iron pistons so that they may be fitted to the regular engine line for special service at higher speed.

Light metal pistons are very useful in extending the possible diameter of uncooled pistons, in slightly raising the critical range of torsional vibration and in keeping inertia values within bounds for larger engines.

PISTON FITTING

All pistons should be "wabbled" or relieved by eccentric turning about the pin hole as in Fig. 112. Each relief area should cover 90° of the circumference. The greatest depth of relief may be $0.004\,D$. Lands must be left at the top and bottom of skirt for side guidance, but they need not be longer than $1/16\,D$ unless the piston projects from the bore at the stroke end. In this case the relief should not leave the bore. If it does a pronounced click may be heard once per revolution.

The fire end of the cast-iron piston is almost invariably turned $0.005\,D$ smaller than the bore with a tolerance of ± 0.0025 in. Those of aluminum pistons may be turned $0.009\,D$ small with the same tolerance. These allowances may not seem to be properly related but it will be remembered that the aluminum piston runs far cooler, of necessity, than the iron one.

The ring lands are tapered or stepped up in diameter to that of the barrel, which is usually straight although sometimes tapered one or two-thousandths, being smaller at the fire end.

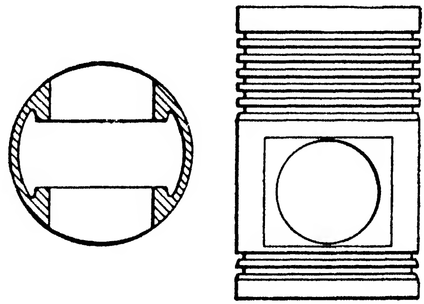


FIG. 112

The traditional barrel allowance of 0.001 in. per in. of cylinder diameter suffices for iron pistons at moderate speed (750 to 900 piston feet) as long as the engine is kept in good condition and not overloaded either totally or in some particular cylinder. It is sometimes better to fit all pistons 0.00125 in. loose per in. of diameter. Up to 1200-ft piston velocity they are well given an allowance of $-0.0015\,D$. At 1600 ft they will require at least $-0.002\,D$. The tolerance is ± 0.001 in. Allowances are made on the actual least dimension of bore. The barrel allowances for simple aluminum pistons should be $-0.004\,D$ if the service conditions are hard and the speed high.

Cast-iron or Butler pistons fitted to the above allowances will be just audible at full load and will "slap" somewhat although not disagreeably at light loads. It is quite unreasonable to expect pistons of ordinary design to be quiet under all conditions. Operators who have witnessed many disastrous piston seizures due to tight fitting will agree that the sound of an easy-working piston is reassuring rather than annoying. The aluminum piston rarely scores a liner but the seized cast-iron piston does so almost invariably.

PISTON RINGS

The plain snap ring or Ramsbottom ring is still the most reliable fire ring. It is rarely made eccentric, as means have been found to make rings of actual or approximate circularity and equality of wall pressure never approached by the eccentric ring.

Peined or hammered rings have many friends. Rings with rough cast inner wall are highly regarded. Excellent service may be had from most commercial rings.

Circularity is measured by spanning the ring freely in a wire or band. The ring is then measured with micrometers to ascertain its departure from true circular form.

Wall pressure is measured by a jig. The rings are forced into circular form with closed gap by calibrated springs. Circularity is indicated by the rotable dial gage at the center of the fixture.

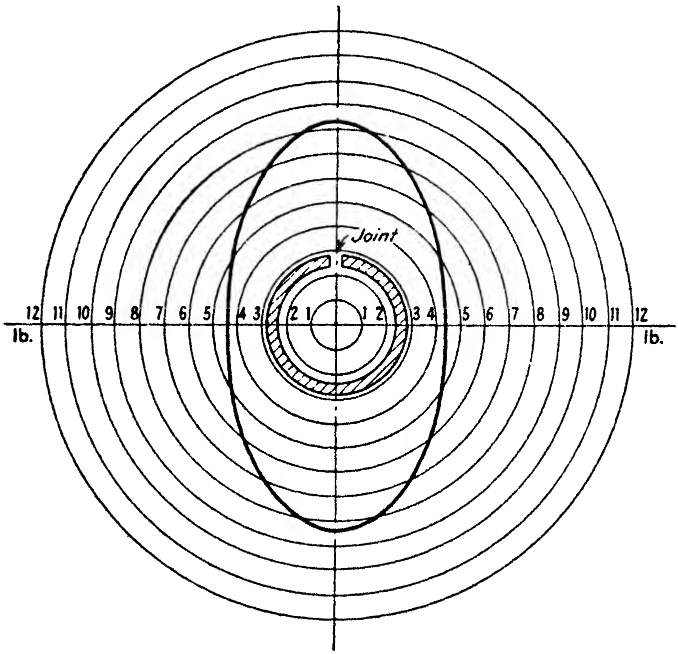


Fig. 113

The spring tensions are charted as indicated in Fig. 113 which shows the chart of a peined ring. Fig. 114 shows that of a ring made by the Kistner process, interesting as an almost ideal uniform tension ring.

The Gorlitzer process which is in most common use, is that described by Hartman in the *Zeitschrift* V.D.I., 1907. In making the pattern a solid circular ring is first made having the same width, thickness and outside diameter as the piston ring when in place in the cylinder, but with finish added all over. This ring is then split and sprung apart to allow the insertion of a straight piece of like section and of a length equal to the "cut-out." The operations are shown in Fig. 115.

Castings made from the pattern are, after edge grinding, "gapped" exactly where the insert was put into the pattern, sprung together, clamped on an arbor and turned. A good ring is

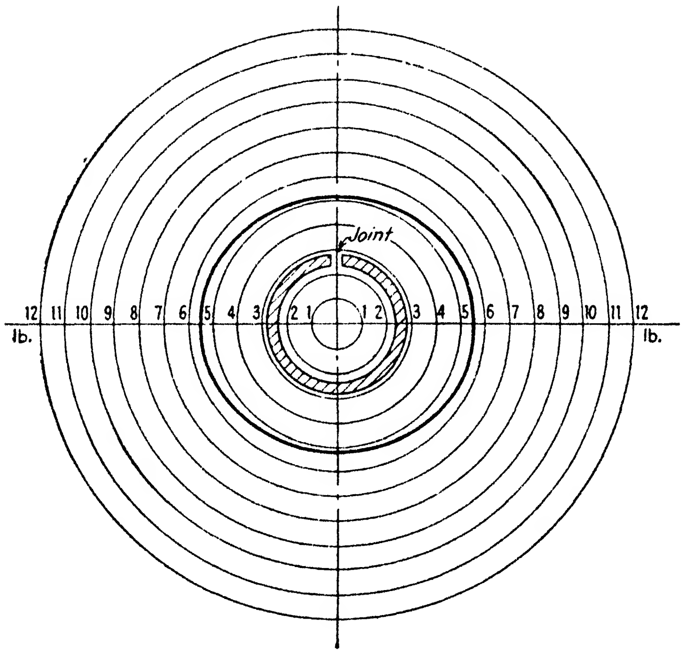


FIG. 114

produced, but not equal in circularity and uniformity of wall tension to the Kistner ring, description of which follows.

Patent application, Serial No. 746,220, filed by Herman E. Kistner, Oct. 27, 1924, specifies in Claim 2:

> The method of making piston rings, which consists of producing a ring blank of elongated circular form, the foci of which are separated along the major axis a definite amount, removing a section from one side of said ring and of a length about six times the distance between the foci of said elongated ring, contracting said ring and then finishing said ring to true circular form.

A model drawing for the Kistner ring is shown in Fig. 116. The pattern dimensions are derived thus:

A constant is chosen for the thickness. This depends on the desired tension. A good value is 0.038 D where D equals cylinder diameter.

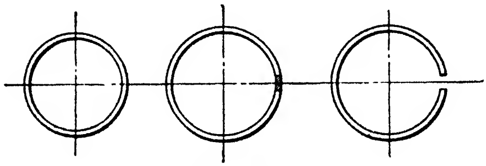


FIG. 115

Next a constant is chosen for the "cut-out." This factor may be 0.090 D.

The finish allowed is 0.060 in. on each side. The casting shrinkage averages 0.015 D.

The so-called correction factor, which, with shrinkage and finish, determines the minor diameters, is one-fifth the cut-out, or

$$\frac{0.090}{5} D = 0.018\,D$$

The inserts are of a length equal to one-sixth the cut-out or 0.015 D, the amount by which the major exceeds the minor diameter. Thus the minor outside diameter of the pattern is expressed

$$d_0 = D + 0.018\,D + 0.120 + 0.015\,D = 1.033\,D + 0.120$$

The major outside diameter, D is equal to the minor plus the insert length, or D_0 equals

$$D_0 = (1.033\,D + 0.120) + 0.015\,D = 1.048\,D + 0.120$$

The inner diameters are, in both cases, equal to the outer diameters, less twice the wall thickness and four times the radial finish allowance. Thus the major and minor inside diameters of the pattern are expressed

$$d_i = 0.957\,D - 0.120$$
$$D_i = 0.972\,D - 0.120$$

The correction factor is probably derived as follows: The cut-out equals a constant times D or KD. The separation of the foci equals $KD/3$. The difference of $KD - (KD/3)$ equals 0.667 KD must be made up by enlarging the diameter of the blank. It is therefore equal to 0.667 KD divided by π or 0.21 KD, which is nearly enough 1/5 KD. The error is no doubt

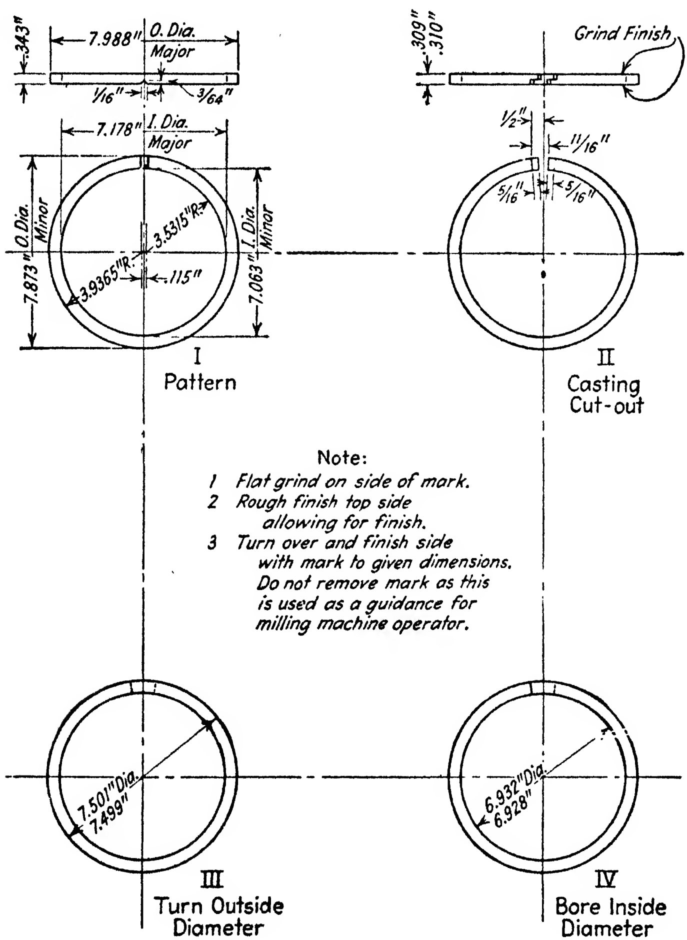


FIG. 116

covered by finish and shrink allowances. The slight variation in dimensions from formula in Fig. 116 is due to using the nearest fractional value for the cut-out, which is quite permissible.

OIL PUMPING

As far back as 1890 Wm. Bole of the Westinghouse Machine Company vented pistons to allow escape of the lubricating oil scraped downward by the upper piston rings and confined in the narrow annulus between piston and bore. If the piston ring is a perfect fit to the bore, practically without joint clearance, it is obvious that terrific oil pressure will be built up below the lower

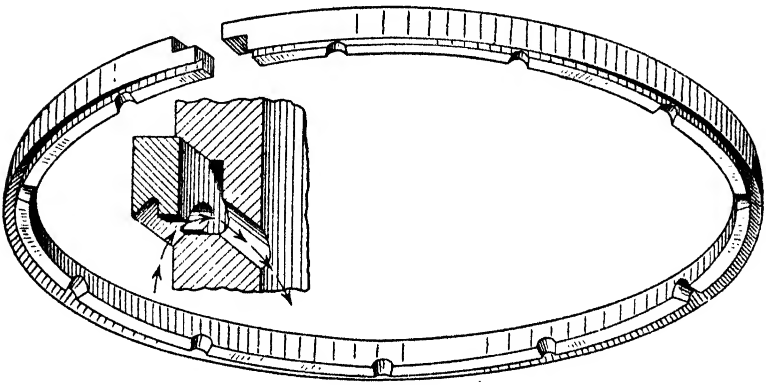


FIG. 117

ring as it scrapes the oil from the bore. Under practical conditions the ring in unvented grooves acts as a valve, passing the oil collected from the bore and preventing its return. Great quantities of oil may be pumped from the crankcase to the combustion chamber in this way. This condition cannot be endured in oil engine practise. White smoke appears in the exhaust, the combustion chamber is fouled, valves are pitted and the engine governs badly owing to the quantity of lubricant irregularly consumed as fuel.

Automobile engine practise has required close diametral and side fits on all crank bearings to limit the amount of oil escaping and thrown to the cylinder walls. The diesel engine operates under more rigid requirements. At shipment it is already "broken in" and ready to assume full load when installed. It may be set to work immediately driving a centrifugal pump in constant full-load, full-speed operation. Oil is flushed through bearings fitted to a good working clearance serving as a coolant as well as a lubricant, and the piston and rings must be capable of returning to the crankcase the great excess thrown on the cylinder bores by cranks and rods. To do this it is well to use never less than two and for high speeds four of the best oil control rings available. They are vented in the ring groove and below the ring groove with an ample number of drilled holes as in Fig. 117.

These rings are made in various ways. Circularity or uniform wall pressure is of first importance. After this the type of ring is to be considered. The Teetor ring, Fig. 15, in the author's experience has never been surpassed. By the grooving and venting of the ring itself, oil pressure sufficient to retract the ring cannot be built up.

PISTON LUBRICATION

While entire dependence must be placed in the splash of oil from the cranks upon the cylinder wall to lubricate small pistons and rings the practise is not to be commended for large engines. It is fairly satisfactory for gas or gasoline engines, but the cylinder-wall oil film of the diesel engine is constantly fouled by the decomposition of the lubricating oil itself and the addition of fuel oil residues. This results in a black sludge which works downward and in the case of the trunk piston engine falls into the crank pit. Much of this matter may be removed by daily or continuous centrifuging or filtering of the oil bath, but no single process except washing by agitating with a coagulant and settling while hot will completely remove all the sludge. The ½ to 1 per cent by volume of sludge remaining after centrifuging contains free or amorphous carbon so finely divided that it is quite harmless to bearings but it will deposit in ring grooves.

A certain amount of make-up oil must be added daily, and there is no better place to administer all or part of it than through a force feed lubricator to the cylinder walls. The cylinder lubricating oil taps are fitted with checks. One tap is usually sufficient for four-cycle engines of 12-in. bore or less. For larger bores to 18 in., two should be used at diametrically opposite positions. Two-cycle engines should have twice as many taps. The oil should be admitted above the ports being delivered midway of the ring belt when the piston is at the top of the stroke. For single-acting four-cycle engines the taps are placed so that they meet the second ring from the top when it is in the lower-most limit of its travel. Timed delivery of oil is not necessary for single-acting engines. The accuracy of delivery of timed pumps is sometimes questionable.

MARINE THRUST BEARINGS

For engines up to about 500 hp and particularly for high-speed engines double-coned roller bearings are available, Fig. 118. If properly selected and protected from salt water they are prac-

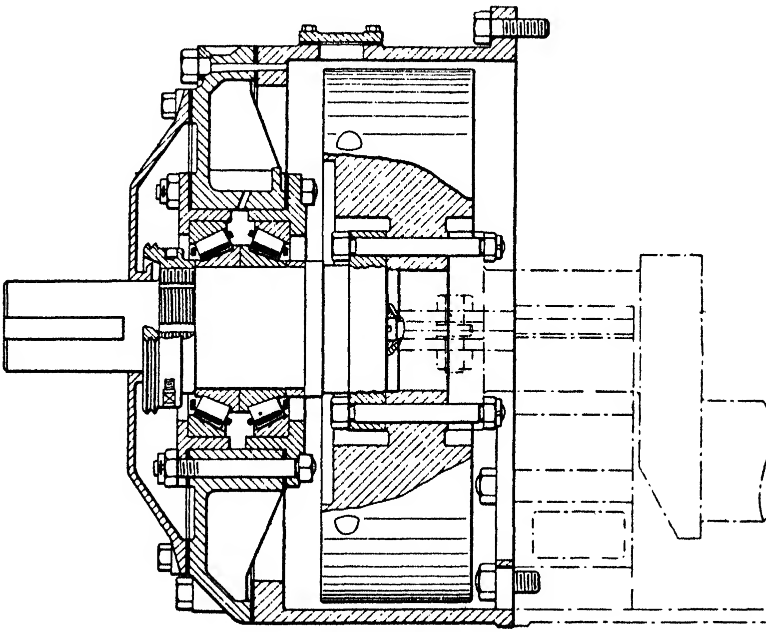


FIG. 118

tically indestructible. They are desirable in that they provide a thrust and steady bearing in one unit and thus keep the engine short. They are, of course, not readily replaced in foreign ports, and replacement requires disconnecting the line shaft and moving it aft. This is sometimes inconvenient on account of the closeness of the after engine room bulkhead. It is also a rather dangerous procedure at sea. The desirable forged-on-flange coupling for the intermediate shaft is also ruled out by the use of roller bearings. In spite of these handicaps the type is popular and deservingly so. The flange carrying these bearings must be adjustable vertically to compensate for crankshaft bearing wear.

The Kingsbury thrust, Fig. 119, is used almost exclusively for larger engines. A part of the lubricating oil supply is diverted over this bearing for cooling. The dimensions of the bearings

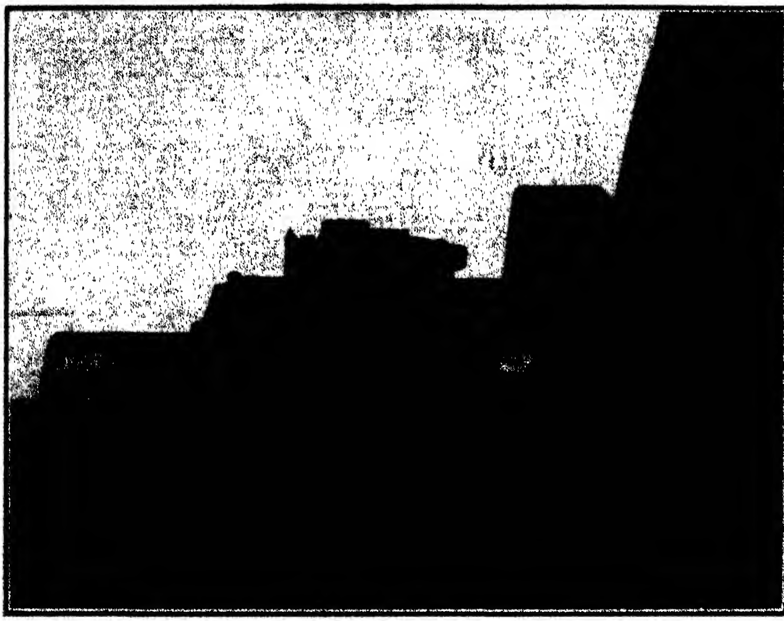


FIG. 119

should be approved by the makers. The shaft collars are lapped or honed with an oil stone to a mat surface and the shoes are babbitt faced. The shoes are supported at the back by a hardened button on which they swivel readily to form a perfect wedge-shaped oil film between collar and bearing surface.

ANTI-FRICTION BEARINGS

Single-cylinder crankshafts, camshafts, jockey pulleys on chain drives, pump shafts, governor shafts, in fact most small mountings, are fitted with ball and roller bearings with flood lubrication. Their dependability is a great relief to the operating staff and they quite often effect manufacturing economies as well.

THE OIL RESERVOIR

The use of the engine base as an oil reservoir still prevails for small engines, but the working supply of oil for larger units usually is kept in a tank below the level of the engine-room floor. Oil is picked up from this tank by a pump, either motor driven or directly connected to the engine, and is forced through the cooler and thence to the main that serves the bearings. The pump is fitted with a by-pass discharging into the tank. This relieves the system of the excess that is not passed by the bearings. A small tap from the pressure line or main leads through a heater to the centrifuge or filter providing for continuous purification on the by-pass plan, the clean oil returning by gravity to the tank. The oil collected in the crankcase is either pumped out or drained to the tank by gravity, the latter preferably if possible. When engines are installed in small craft it is pumped out as the engine is on a sloping foundation, the oil all draining aft where there is not sufficient room between engine base and timbers for piping and certainly none for tanks. The scavenge pump should be of somewhat greater capacity than the force pump, and it should be self-priming as its suction is often open to the air.

THE LUBRICATING OIL PUMP

This unit is quite often of the "rotary" type. The minimum capacity of the pump in gallons per minute varies from $\frac{1}{10}$ to $\frac{1}{20}$ the bhp of the engine. This simple expression makes rough allowance for increased oil flow from bearings and increased total friction heat at higher speeds.

When part of the pump discharge is used to cool a marine thrust bearing the capacity should be increased accordingly.

Rotary pumps are never very efficient and some of them are quite inefficient when pumping heavy oil. This difference is due to the trapping of oil at the bottom of the teeth or vanes. This oil is pocketed and cannot escape to discharge by the usual channel. It is often by-passed back to suction by escape ports in the end housings or by drilling through the gear hubs at the bottom of the tooth spaces. In both cases the escape ports register with the tooth only at the period of revolution when the tooth or lobe is bottoming in an opposed tooth space.

The pump efficiency is of little moment when the pump is engine driven. It may be a more serious consideration when the pumps are driven by motors taking power from the limited capacity in auxiliary generator sets as in ships. All such pump units should be tested on oil of the viscosity preferred for the main engines. This oil should be chilled to the least temperature likely to prevail in the tanks and the motor load should be

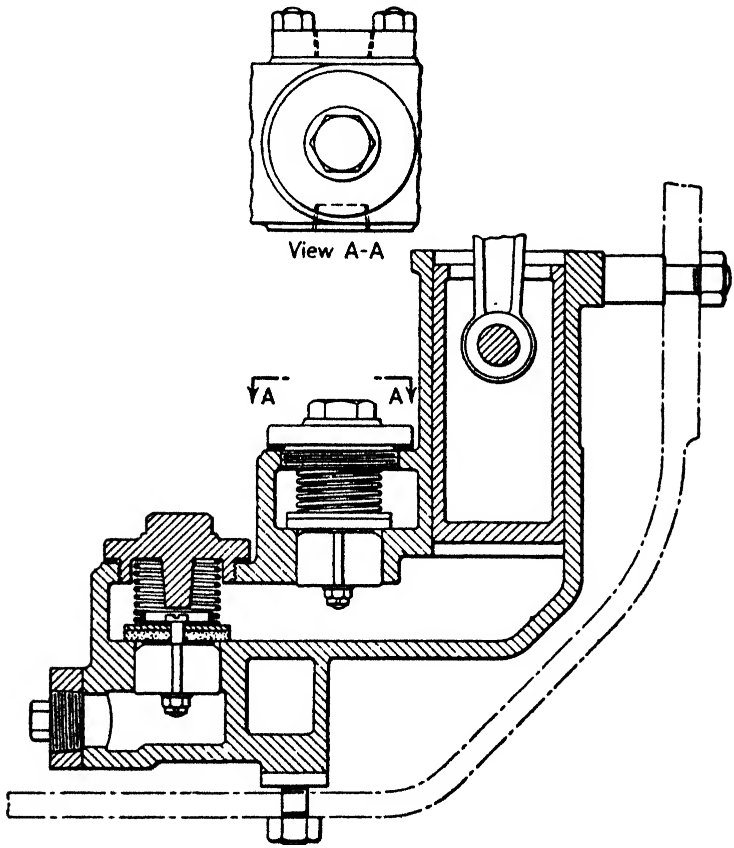


FIG. 120

checked against its rating. Very serious overloading may result if pumps are not carefully designed and adequately motored. The maximum pump efficiency to be assumed in selecting motors is 50 per cent. Commercial pumps have shown as little as 25 per cent efficiency when tested on heavy oil.

Rotary pumps are made reversible, either by the use of double suction and discharge connections provided with non-return valves or by the ingenious application of some floating member that shifts when the rotors change direction, by mere contact with them. These devices, however, are not extensively used. Plunger pumps are in favor when it is necessary to use direct-driven pumps on reversible engines. If large suction and discharge air chambers are used they operate well at fairly high speeds, 200 to 300 rpm being quite common for single-plunger pumps. The valve seats may be made of leather, as in Fig. 120, which enables them to handle a reasonable amount of foreign material and contributes to silent operation.

THE LUBRICATING OIL COOLER

The cooling surface provided for lubricating oil depends on bearing velocity and unit pressures, the nature of housings, the temperatures prevailing in the engine room, the temperature of inlet water available and a great number of more or less indeterminate factors. A good general approximation for engines operating at the speeds considered desirable today and in tropical waters is a cooling surface expressed in square feet equal to $\frac{1}{8}$ the bhp rating of the engine.

Coolers usually are of the tube bundle type. The heads should be provided with zincs to prevent damage by electrolysis. Such coolers rarely have water flow area equal to that required for the engine jackets with a reasonable pressure drop (max 5 lb) so they are by-passed, a throttle being fitted to the detour pipe to force the diversion of enough water through the cooler for its purpose. The maximum lubricating oil pressure drop in passing through the cooler should not exceed 5 lb per sq in. The maximum oil temperature should not exceed 160° in spite of the high temperatures advised today.

THE LUBRICATING OIL STRAINER

It is usual to place a dual strainer in the discharge line from the lubricating oil pump to remove débris. These strainers are in no sense filters. Two units in one body are provided having a switch valve which cuts one out for cleaning and the other in, at will. The baskets may be made of wire cloth well supported with a heavy brass grid but more often are of perforated sheet metal. When clogged slightly the spaced weft and warp of the wire cloth tends to crowd together leaving large openings for the passage of adventitious material.

If wire cloth is used the mesh is usually 30 per inch or less. If the screen is of perforated sheet brass the holes are generally 0.025 in. in diameter, which is about the smallest commercial punching.

The pressure drop through a clean 30-mesh screen should not exceed ½ lb per sq in. Through perforated metal it may go to 1 lb or even 2 lb. In both cases this drop includes that through switch valve and strainer connections, all areas being at least

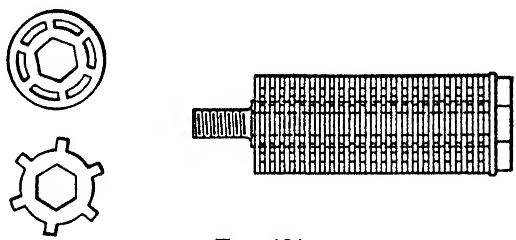


FIG. 121

equal to the pipe area. Edge strainers, Fig. 121, made of punched disks of two profiles clamped together are in general use at much higher pressures. The units are shaken apart slightly and washed in gasoline to clean.

THE CENTRIFUGE

The amount of "colloidal carbon" produced in the crankcase oil of a trunk piston diesel engine varies with the engine type, its condition, its fuel and its lubricant. Carbon removed varies from 1 lb per 3000 hp-hr to 1 lb per 10,000 hp-hr. The bowl capacity of the centrifuge determines the frequency of cleaning. The cost of the centrifuge increases with bowl capacity. This is an economic problem as well as one of convenience. Large centrifuges are advisable always. A small centrifuge is better than none. Any of the well-known standard makes will keep the colloidal carbon down to a concentration of less than 1 per cent. This result is shown by diluting the oil from the centrifuge with 4 parts of 90 per cent benzol and swinging for 15 minutes in a powerful laboratory centrifuge. The precipitate in the tube is considered as colloidal carbon although some metallic particles, asphalt and emulsion are removed.

ANTI-FRICTION BEARINGS FOR CRANKSHAFTS

Roller bearings are in quite common use for single-cylinder engine crankshafts. They permit easy hand starting of engines of 40 and more horsepower when equipped with the heavy wheels so useful in increasing the life of belting and pumping equipment. They reduce the beam length of the shaft by bringing reaction points closer to the crankpin, and thus reduce shaft deflection. As a consequence wheel shimmy due to lateral vibration of the flywheel masses with the crankshaft need not occur within the useful speed range.

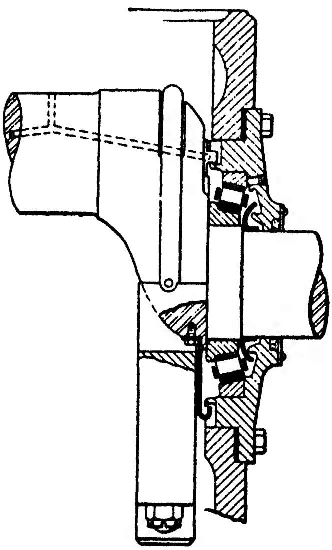


FIG. 122

The question has been raised as to whether single-throw crankshafts should be supported on single bearings as in Fig. 122 or double bearings like Fig. 123. It is not possible to find in any of the collections of beam formulas a case like Fig. 124, i.e., a simple beam supported at two points on each end and loaded at the center. In beam problems the supports R_1, R_2, R_3, R_4 in Fig. 124 are assumed to be inelastic. The slightest deflection of the beam under load or even that due to its own weight throws all the burden on R_2 and R_3 completely, relieving R_1 and R_4.

When the investigator prefers to consider the shaft as a restrained beam as in Fig. 125 it is easy to fall into the error of assuming R_1 and R_2 to be a couple supplying the resisting moment. Obviously the downward forces in Fig. 125 must equal the upward forces to assur^ a state of equilibrium or

$$W + R_1 + R_4 = R_2 + R_3$$

If this were a simple beam on two supports carrying the same load W the equation would be

$$W = R_2 + R_3$$

Clearly, the supports R_2 and R_3, Fig. 125, carry not only the imposed load, but the reactions R_1 and R_4 as well.

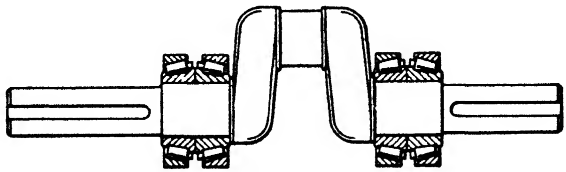


FIG. 123

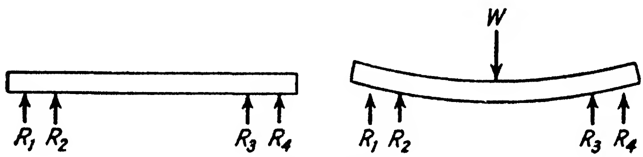


FIG. 124

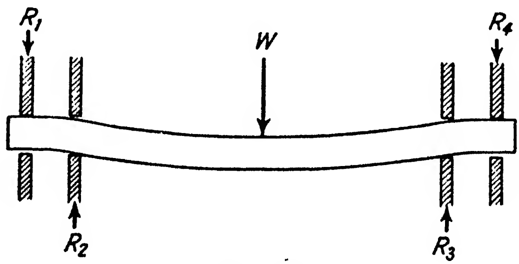


FIG. 125

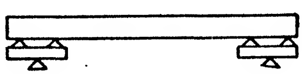


FIG. 126

The only correct way to apply multiple-row roller bearings is shown in Fig. 126, i.e., in a floating housing as in a car journal box. The housing accommodates itself on its pivot seat to the axle deflection. This mounting method is not very practicable for engine work.

It is sometimes assumed that coned roller bearings mounted as in Fig. 123 are possessed of the properties of a self-aligning bearing. This, too, is a dangerous fallacy. If the bearings are snug a very slight deflection results in restraint of the shaft. This is applied as in Fig. 127. If the bearing clearance is assumed

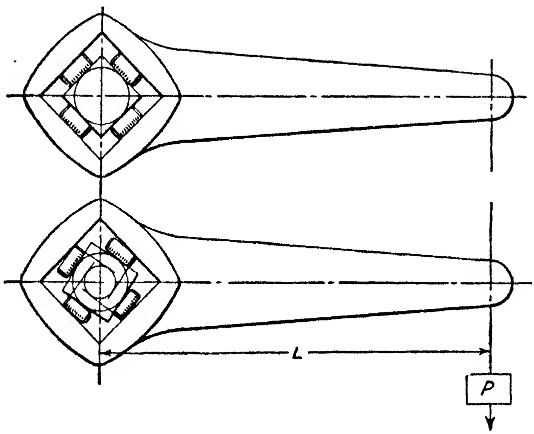


FIG. 127

to be infinitesimal the bursting force on the housing will be infinite, theoretically.

Crankpin lubrication may be accomplished as in Fig. 122 for roller bearing construction.

CHAPTER XI

THE INDICATOR DIAGRAM

For complete analysis of forces acting upon engine members the indicator card is necessary. With increasing speeds and increasing pressures the piston indicator, the only really convenient and easily calibrated pressure-stroke charting instrument yet invented, has proved inadequate. Its reciprocating drum motion

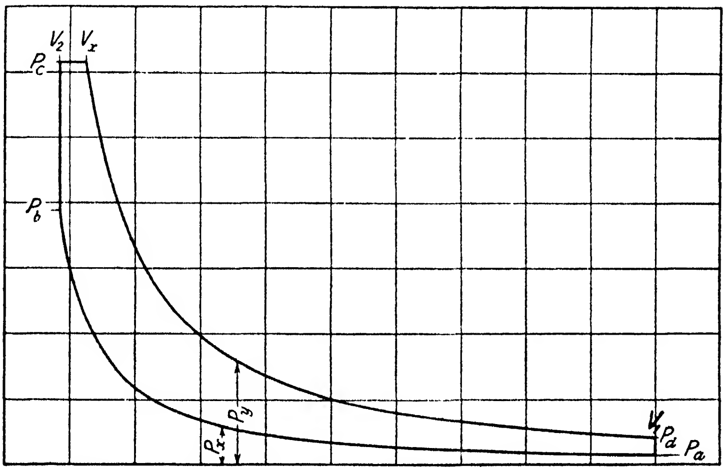


FIG. 128

is subject to errors due to flexibility and lost motion of the drive. The pencil motion records not only the cylinder pressures, but also its own inertia effects superimposed on the true pressure line. The scale of ordinates, usually from 400 to 750 lb per in., is subject to errors due to whip and play of the pencil motion. It is also difficult to measure the ordinates accurately.

On the whole, particularly for engines in prospect, it is better to work from constructed diagrams, Fig. 128. The foundation data for these cards should be acquired by careful test of a number of existing engines with the best instruments available.

Four cycle types show an average compression exponent equal to 1.35; a sea-level cylinder filling to 13.7 lb absolute and a maximum pressure at 750 to 900 ft piston velocity equal to 600 lb gage. The sea-level compression desired usually is 375 lb gage. The polytropic exponent of the expansion line is 1.30. The cycle is mixed; that is, constant volume combustion raises the cylinder pressure to 600 lb maximum, which can be arranged by the fuel injection timing. This is followed by fairly consistent constant-pressure combustion for a longer or shorter period depending on the mep being produced.

The compression volume, relative to the total cylinder volume (displacement plus "clearance"), is derived as follows:

V_1 = total volume P_a = initial pressure — absolute

V_2 = clearance volume P_b = compression pressure — absolute

$$\sqrt[1.35]{\frac{P_b}{P_a}} = \frac{V_1}{V_2}$$

Substituting

$$\sqrt[1.35]{\frac{389.7}{13.7}} = 11.94$$

When

$$V_1 = 1, \quad V_2 = \frac{1}{11.94} = 0.0838$$

Lest some experimenter find the practical outcome of this calculation discouraging it may be well to remark that it is quite common and sometimes of no great moment to find a total variation of compression pressure amounting to 25 lb per sq in. among the cylinders of an engine in good order. It is common practise to allow a tolerance of 12½ lb per sq in. When an ill-devised manifold contributes most to this variation, the cylinder indicating the lowest pressure suffers a volumetric loss of 6 to 7% over that indicating the highest. Diesel engines deserve as carefully studied manifolding as gasoline engines, although they seldom receive it.

Machine and gasket tolerances often are responsible for a total cumulative error of $\frac{1}{64}$ in. in the axial dimension of the combustion chamber. Long aluminum pistons extend greatly when hot and occasionally raise the expected compression surprisingly. Needless to say, clearance measurements should be made at working heat.

The mean effective pressure under the compression line is expressed as follows:

$$P_x = \frac{P_b V_2 - P_a V_1}{(n-1)(V_1 - V_2)}$$

Substituting

$$P_x = \frac{(389.7 \times 0.0838) - (13.7 \times 1)}{0.35(1 - 0.0838)} = 59.12 \text{ lb per sq in. absolute}$$

The mean effective pressure under the combustion and expansion line equals

$$P_y = \frac{614.7(V_x - V_2) + \left(\dfrac{614.7V_x - P_d V_1}{n-1}\right)}{V_1 - V_2}$$

V_x = volume at end of constant pressure phase

P_d = terminal pressure of expansion.

But

$$P_d V_1^{1.3} = P_c V_x^{1.3}$$

And

$$P_c = \text{constant} = 614.7 \text{ lb}$$

Transposing and introducing the finite value of P_c

$$P_d = \frac{614.7}{\left(\dfrac{V_1}{V_x}\right)^{1.3}}$$

Factoring and introducing constants

$$P_y = \frac{614.7\left[V_x - 0.0838 + \dfrac{V_x - \dfrac{1}{\left(\dfrac{1}{V_x}\right)^{1.3}}}{0.30}\right]}{0.9162}$$

Assuming values for V_x corresponding values of P_y may be calculated.

Tabulated, these are as follows:

TABLE XI

V_x	P_y
0.0838	98.2
0.10	122.0
0.13	163.8
0.15	189.7

The values of V_x and P_y are plotted as co-ordinates in Fig. 129.

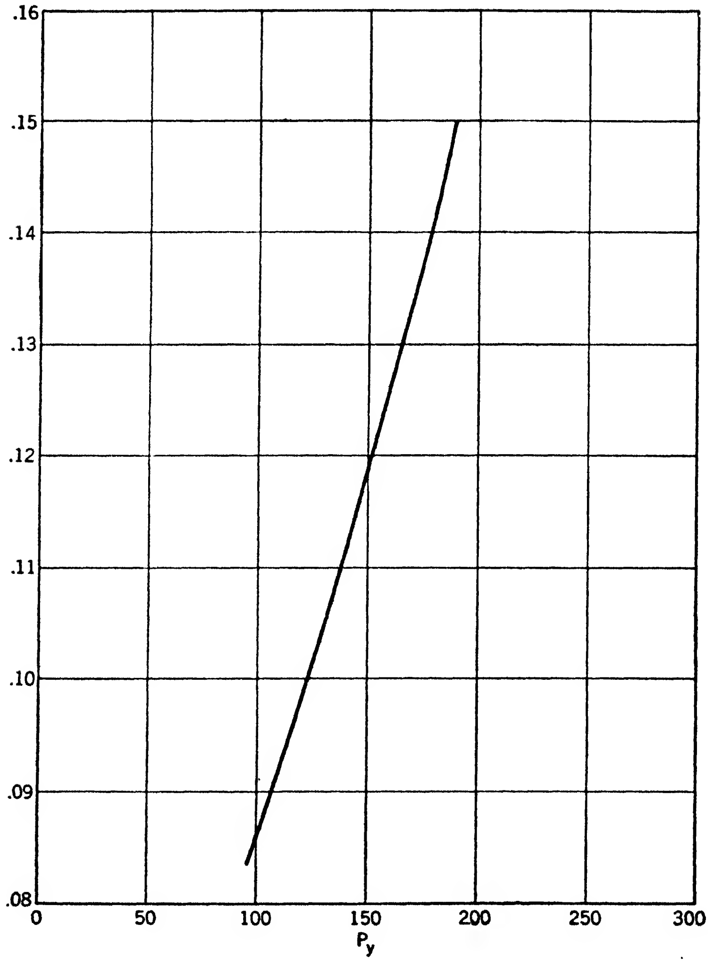


FIG. 129

A simple procedure is to plot the diagram first on logarithmic paper as in Fig. 130. The filling pressure 13.7 lb abs is known or assumed. The desired compression pressure is known as 389.7 abs. A straight line is drawn through the P and V intercepts.

As in Fig. 130, the slope of the line will be found to take the angle 53° 28′ from the horizontal, the tangent of which is

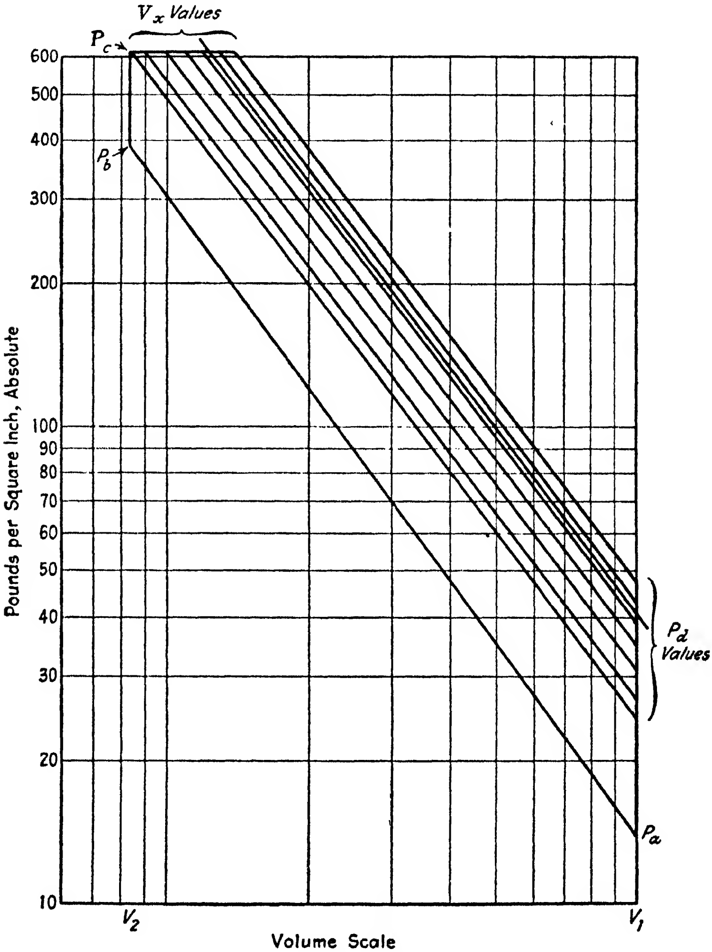


FIG. 130

1.35 or n. The expansion line is plotted by the same procedure, the line being drawn from the 614.7 level at its intercept with V_x and sloping downward, at an angle 52° 26′, the tangent of which is 1.30.

As angles are difficult of measurement on thin paper and with instruments of such a low order of accuracy as the draftsmen's tee square, parallel rule and protractor, it is well to locate the terminal pressure P_d of the expansion line by calculation.

$$P_d = \frac{614.7}{\left(\frac{1}{V_x}\right)^{1.3}}$$

From the logarithmic diagram it is a simple matter to transfer to a scale of ordinary numbers, avoiding a wearisome calculation. The diagram, Fig. 128, if carefully made on accurately scaled paper, will planimeter correctly to about plus or minus 2 per cent including the planimeter error and will agree quite closely with the best indicator cards obtainable. There is a small and negligible error in negative work under the atmospheric line, and the "release" or drop of pressure upon opening the exhaust valve is not in evidence. The release line is so variable in shape that it cannot be accounted for. A little "fairing" of the lines will yield a diagram as accurate as the means of measuring.

If the above procedure is applied to two-cycle engines P_a usually is taken at full atmospheric pressure when there is no supercharging. The stroke is considered as beginning when the piston end is line and line with the inner end of the exhaust port. All volumes are based on a stroke equal to the whole stroke less the "dead stroke" which latter is the length of the exhaust port.

CHAPTER XII

THE INERTIA OF RECIPROCATING PARTS AND BALANCING

SINGLE-CYLINDER ENGINES

The forces within the engine cylinder are resisted by the engine framing which is held in tension by the upward thrust of the cylinder head under gas pressure and the downward thrust of the piston against the crank.

The forces of acceleration and retardation and the centrifugal forces are called free forces because they are not self-contained within the engine mechanism and framing but are resisted only by the mass of the engine and its foundations unless other free forces are set in opposition to them.

All purely centrifugal forces should be balanced dynamically in any crankshaft design.

The crank and that part of the connecting rod which is described as rotating may be balanced in the plane of rotation by counterweighting. The counterweight, being fixed opposite to the crank, stresses only its own fastenings in establishing balance of rotating parts.

The piston and that part of the connecting rod which is assumed to reciprocate with it may be balanced approximately, at the dead centers, by a rotating weight secured to the crank. At the centers the radially directed centrifugal force of the counterweight opposes the force required to stop the reciprocating parts and start them in the opposite direction.

Near the half-stroke position the piston momentarily takes on the constant velocity of the crankpin and being neither accelerated nor retarded exerts no free force whatever on the engine frame. At this time any counterweighting in excess of that required to balance rotating parts exerts its full effect at right angles to the cylinder axis and becomes a free force. In short, anything added in opposition to the reciprocating weight is effective only in the direction of reciprocation and acts partially or wholly as free force at all other positions tending to produce a radial disturbance with a component in a direction normal to the cylinder axis. At the quarters or 90° positions of the crank the free force due to "over-counterweighting" is exactly equal to the reduction of free force at the dead centers.

COUPLES

Two or more cranks or two or more trains of reciprocating parts may be set in opposition to establish balance either partial or complete.

Two-cylinder engines with opposed cranks set up opposed free forces, but they act in different planes tending to first lower one end of the frame and raise the other and then to reverse this action during each revolution.

If two such cranks are joined as in four-cylinder four-cycle engines the couples neutralize each other and approximate balance is secured.

ANGULARITY

This balance would be perfect except for an irregularity in crank and connecting-rod motion. As the piston leaves the head center, crank and rod are arcing away from each other. As the piston approaches or leaves the outer center, crank and rod arc together. Consequently the piston's movements about the head center are faster than its movements about the outer center. The piston reaches mid-stroke when moving outward sooner than when moving inward, so diametrically opposed reciprocating parts are never quite perfectly balanced. This accounts for the distressing vibrations set up by four-cylinder four-cycle engines in many situations.

CALCULATION OF INERTIA

If it were not for the angularity of the connecting rod the calculation of inertia forces would be an extremely simple matter. For simple harmonic motion of reciprocating parts such as results from the Scotch yoke, Fig. 131, the accelerating and retarding forces at the dead centers are exactly the same as the centrifugal force of an equal weight rotating at the crankpin radius.

As simple harmonic motion is defined, the forces of acceleration and retardation are directly proportional to the displacement.

Thus the "first harmonic" of the reciprocating forces may be represented by the triangles of Frederick Slade, Fig. 132. The

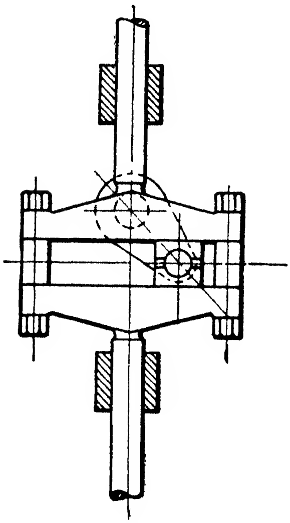


FIG. 131

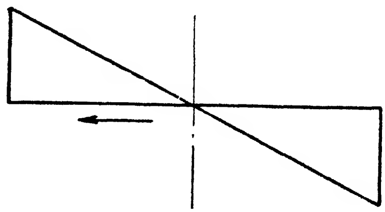


FIG. 132

ordinates at the dead center are

$$F_a = 0.00034\, WrN^2$$

in which

W = the weight of the reciprocating parts
r = crank radius in feet
N = rpm.

SECOND HARMONICS

Fourier elaborated the theory that any irregular curve repeating cyclically or periodically may be approximated by superimposing upon the first harmonic, possibly a sine curve of the same frequency as the curve to be investigated, other sine curves of greater frequencies and other amplitudes. Thus in Fig. 133 the triangle is approximated.

Analysis has shown that only two harmonics are of importance in this case. They are the first harmonic F_a having a period equal to that of crank revolution, and a second harmonic f_a

having an amplitude

$$f_a = \frac{r}{l_c} F_a$$

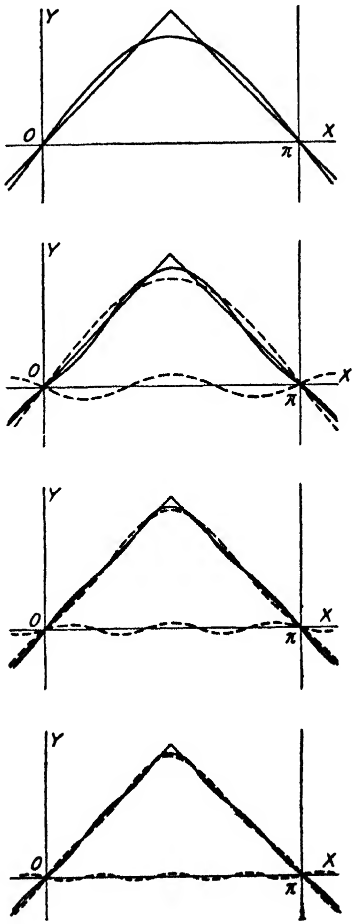


FIG. 133

in which

r = crank radius

l_c = connecting-rod length

and a frequency equal to twice that of crank revolution.

When plotted with crank angles as abscissas the points of the inertia curve are easily calculated.

Since the primary harmonic force is directly proportionate to displacement from the center of motion it is proportionate to the cosine of the crank angle, the second harmonics being proportionate to the cosine of twice the crank angle.

Then

$$F_{ax} = \cos\theta\,(0.00034\,WrN^2)$$

$$f_{ax} = \cos 2\theta\frac{r}{l_c}\,(0.00034\,WrN^2)$$

$$F_{ix} = F_{ax} + f_{ax} = 0.00034\,WrN^2\left(\cos\theta + \frac{r}{l_c}\cos 2\theta\right)$$

Values of the cosine term of the equation are given in Table XII.

A problem from practise follows:

First one and then the other end of the connecting rod is weighed as in Fig. 134. This results in the "divided rod" approximation. The weight of the large end is assumed to rotate. That of the small end is added to the other reciprocating weights such as the piston and crosshead if any.

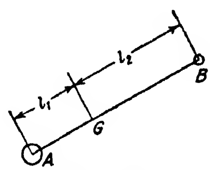


FIG. 134

If the rod is not yet in existence the division may be approximated by calculation. Find the center of gravity at G, Fig. 134. Then $w_1l_1 = w_2l_2$ where w_1 equals reciprocating weight and w_2 = rotating weight.

For most purposes it is convenient to use a unit value for W as this compares directly with the indicator card. Thus the total reciprocating weight divided by the piston area equals W_1.

Taking as an example a 12 in. by 18 in. engine operating at 257 rpm with a total reciprocating weight of 459 lb, the unit weight

$$W_1 = \frac{459}{113.10} = 4.06 \text{ lb per sq in.}$$

$$\frac{r}{l_c} = \frac{1}{4.5}$$

TABLE XII

VALUES OF ($\cos\theta + a_1 \cos 2\theta$)

θ	$\cos\theta + a_1 \cos 2\theta$	$\frac{1}{a_1}$ 3.0	$\frac{1}{a_1}$ 3.5	$\frac{1}{a_1}$ 4.0	$\frac{1}{a_1}$ 4.5	$\frac{1}{a_1}$ 5.0	$\frac{1}{a_1}$ 5.5	$\frac{1}{a_1}$ 6.0
0	$(1+a_1)$	1.333	1.286	1.250	1.222	1.200	1.182	1.167
30	$(0.866+0.5a_1)$	1.033	1.009	0.997	0.977	0.966	0.957	0.949
45	0.707	0.707	0.707	0.707	0.707	0.707	0.707	0.707
60	$(0.5-0.5a_1)$	0.333	0.357	0.375	0.389	0.400	0.409	0.414
90	$-a_1$	−0.333	−0.286	−0.250	−0.222	−0.200	−0.182	−0.167
120	$-(0.5+0.5a_1)$	−0.667	−0.643	−0.625	−0.611	−0.700	−0.591	−0.583
135	−0.707	−0.707	−0.707	−0.707	−0.707	−0.707	−0.707	−0.707
150	$-(0.866-0.5a_1)$	−0.700	−0.723	−0.745	−0.755	−0.766	−0.755	−0.783
180	$-(1-a_1)$	−0.667	−0.714	−0.750	−0.778	−0.800	−0.818	−0.833

NOTE: $\frac{1}{a_1} = \frac{1}{r/l}$.

The value of the term

$$0.00034\, W_1 r N^2 =$$

$$0.00034 \times 4.06 \times \tfrac{9}{12} \times 257^2 = 68.38 \text{ lb}$$

The rest of the calculation is tabulated (Table XIII).

TABLE XIII

θ	$\left(\cos\theta + \frac{r}{l_c}\cos 2\theta\right)$	$F_{ax} + f_{ax}$
0	1.222	83.56
30	0.977	66.82
45	0.707	48.34
60	0.389	26.60
90	−0.222	−15.18
120	−0.611	−41.78
135	−0.707	−48.34
150	−0.755	−51.63
180	−0.778	−53.20

NOTE: Angles θ are measured from head end dead center.

THREE-CYLINDER ENGINES

When three cranks are spaced 120° apart as they are for both two- and four-cycle engines the summation of free forces in a vertical direction is zero. A fragment of this summation is derived from Table XIII and shown in Table XIV.

TABLE XIV

No. 1 crank	Cos factor	No. 2 crank	Cos factor	No. 3 crank	Cos factor	Sum
0°	+1.222	120°	−0.611	120°	−0.611	0.000
30°	+0.977	150°	−0.755	90°	−0.222	0.000
60°	+0.389	180°	−0.788	60°	+0.389	0.000

NOTE: Angles are measured from head end dead center.

Here as in the two-cylinder engine a couple results since the forces are not in the same plane. This couple reaches a maximum value of moment (Fig. 135) equal to

$$\left(1 \pm \frac{r}{l_c}\right) 0.00034\, WrN^2 \times \frac{3}{2} l$$

in which l equals the cylinder spacing and W equals the total weight of one set of reciprocating parts.

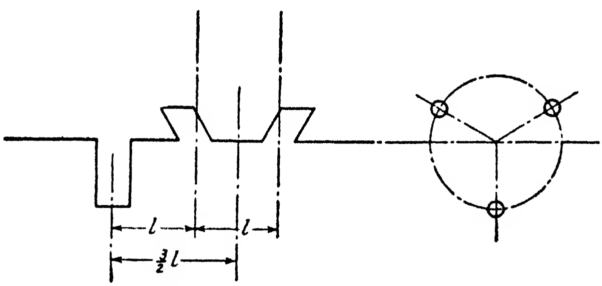


FIG. 135

It is presumed that the shaft has been counterweighted to balance the rotating masses at least.

Additional counterweighting may be added resulting in a diminished couple in the plane of the cylinder axes and a new one at right angles.

When two three-cylinder cranks of opposite twist are joined together with cranks 1—6, 2—5 and 3—4 together as in conventional four-cycle single-acting engines, Table XV, the couples resist each other through the engine framing and the free force is zero.

TABLE XV

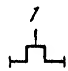
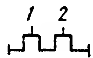
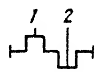
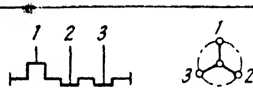
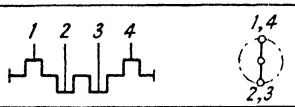
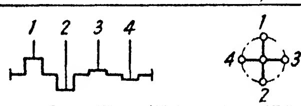
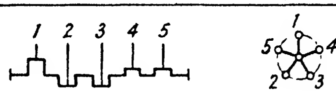
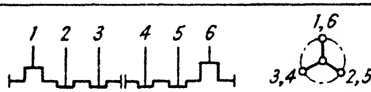
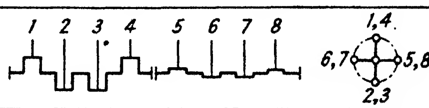
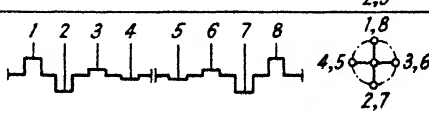
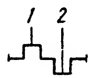
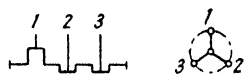
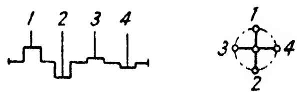
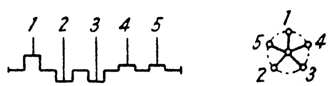
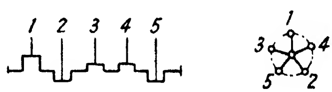
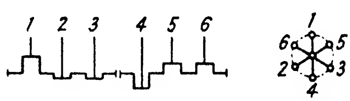
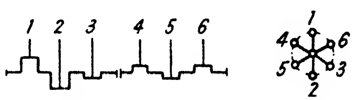
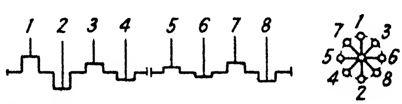
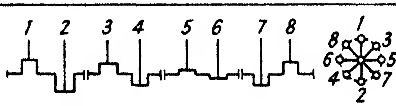
Single-Acting FOUR-Cycle Engines			
Number of Cylinders	Firing Order		Crank Arrangement
1	0	1	
2	360°	1, 2	
2	180°	1, 2	
3	120°	1, 2, 3	
4	180°	1, 2, 4, 3	
4	90°	1, 2, 4, 3	
5	72°	1, 3, 5, 4, 2	
6	120°	1, 2, 3, 6, 5, 4	
6	120°	1, 5, 3, 6, 2, 4	As Above
8	90°	1, 5, 2, 6, 4, 8, 3, 7	
8	90°	1, 6, 2, 4, 8, 3, 7, 5	

TABLE XVI

Single-Acting TWO-Cycle Engines			
Number of Cylinders	Firing Order		Crank Arrangement
1	0	1	
2	180°	1,2	
3	120°	1,2,3	
4	90°	1,4,2,3	
5	72°	1,5,2,3,4	
5	72°	1,3,5,2,4	
6	60°	1,6,2,4,3,5	
6	60°	1,4,5,2,3,6	
8	45°	1,7,5,4, 2,8,6,3	
8	45°	1,8,6,4, 2,7,5,3	

The two six-cylinder two-cycle arrangements in Table XVI also are free of couples.

FOUR-CYLINDER TWO-CYCLE AND EIGHT-CYLINDER FOUR-CYCLE ENGINES

The four-cylinder two-cycle crank arrangement, Table XVI, shows a vertical summation of free forces equal to zero. A couple is present with a maximum nearly equal to that of a three-cylinder engine of the same bore and stroke.

When two such crankshafts are placed end to end as in the eight-cylinder four-cycle arrangement, Table XV, the couples are opposed tending to stress the engine frame as a beam on two supports with two symmetrical loads. The resisting moment is set up by the greatest moment of inertia of the mid-transverse section of the frame. No free force is present.

OTHER ARRANGEMENTS

By summation of vertical components any other crank arrangement may be investigated. Couples are detected easily as in Fig. 135.

✓ RECIPROCATING WEIGHTS

It has been shown that couples and vertical free forces may be eliminated by suitable crank arrangements. The designer should never forget, however, that four-cycle engine crank and wrist-pins are subjected to shocks as the loads change from positive to negative so parts as light as is consistent with their required life are essential.

The mean terminal inertia ($0.00034\ W_1rN^2$) should not exceed 100 lb per sq in. of piston area if possible. When very high inertias are permitted in high-speed engines the connecting-rod bearings require frequent refitting as any excess of play causes destructive pounding.

Balanced engines do not show outward evidence of inertia loads, but the individual rods of a multicylinder engine are subjected to exactly the same loading as the single-cylinder engine rod acted upon by the same weight.

Heavy inertias are helpful in two-cycle single-acting diesel engines in reducing the intense unidirectional bearing loads.

This applies particularly to wristpins which are difficult to lubricate on account of their slight angular motion.

The three-cylinder four-cycle and the four-cylinder two-cycle engines are in a class alone in that they tend to pitch on the foundation rather than to stamp. The effect of this pitching

FIG. 136

which reverses every revolution is minimized by a long foundation having good footing at each end.

This results in a decided advantage often realized in ships over one-, two- or four-cylinder engines which act directly and equally on each foundation particle.

Second harmonics producing fluctuating forces having a frequency of twice engine speed are free in one-, two- and four-cylinder four-cycle engines. They greatly extend the range of vibration possibilities since the engine is not only able to tune in with all building members, ship members or fittings having a natural frequency of engine speed and half engine speed but also stiffer members having frequencies as high as twice engine speed.

Cork-seated foundations, Fig. 136, are effective in many cases in destroying vibration. They should always be used in commercial buildings as the periodic forces due to the counter turning effort cannot be eliminated by any ordinary means.

CHAPTER XIII

THE FLYWHEEL

Graphical methods may be used to develop the turning effort curve from which the quantity of energy to be absorbed and redistributed by the flywheel is determined.

In these operations the mean indicated pressure is assumed and an ideal indicator card, Fig. 128, is plotted (Chapter XI). This card is converted to an open diagram with crank angles as abscissas, Fig. 137, *a* and *c*, with the help of Table XVII.

TABLE XVII

VALUES OF $\frac{s}{r} = 1 - \cos\theta + \frac{1}{2} a_1 \sin^2\theta$

θ	$1-\cos\theta+\frac{1}{2}a_1\sin^2\theta$	$\frac{1}{a_1}$ 3	$\frac{1}{a_1}$ 3.5	$\frac{1}{a_1}$ 4.0	$\frac{1}{a_1}$ 4.5	$\frac{1}{a_1}$ 5.0	$\frac{1}{a_1}$ 5.5	$\frac{1}{a_1}$ 6.0
0	0	0	0	0	0	0	0	0
30	$0.134 + \frac{1}{8} a_1$	0.175	0.170	0.165	0.162	0.159	0.157	0.155
45	$0.293 + \frac{1}{4} a_1$	0.376	0.364	0.356	0.349	0.343	0.339	0.335
60	$0.50 + \frac{3}{8} a_1$	0.625	0.608	0.594	0.584	0.575	0.569	0.563
90	$1.0 + \frac{1}{2} a_1$	1.167	1.143	1.125	1.111	1.100	1.091	1.083
120	$1.5 + \frac{3}{8} a_1$	1.625	1.608	1.594	1.584	1.575	1.569	1.563
135	$1.707 + \frac{1}{4} a_1$	1.790	1.778	1.770	1.763	1.757	1.753	1.749
150	$1.866 + \frac{1}{8} a_1$	1.908	1.902	1.897	1.894	1.891	1.889	1.887
180	2.0	2.00	2.00	2.00	2.00	2.00	2.00	2.00

Inertia values are calculated as in Chapter XII. They are shown at *b*, Fig. 137, continuing through all four strokes. On the power stroke the curve *d* represents the algebraic sum of curves *a* and *b*. On the compression stroke the algebraic sum of curves *b* and *c* results in curve *e*.

The continued curve *d* for the power stroke, *b* for suction and exhaust strokes and *e* for the compression stroke represents the unit pressure in a direction parallel to the cylinder axis for one cycle of a single-cylinder four-stroke engine.

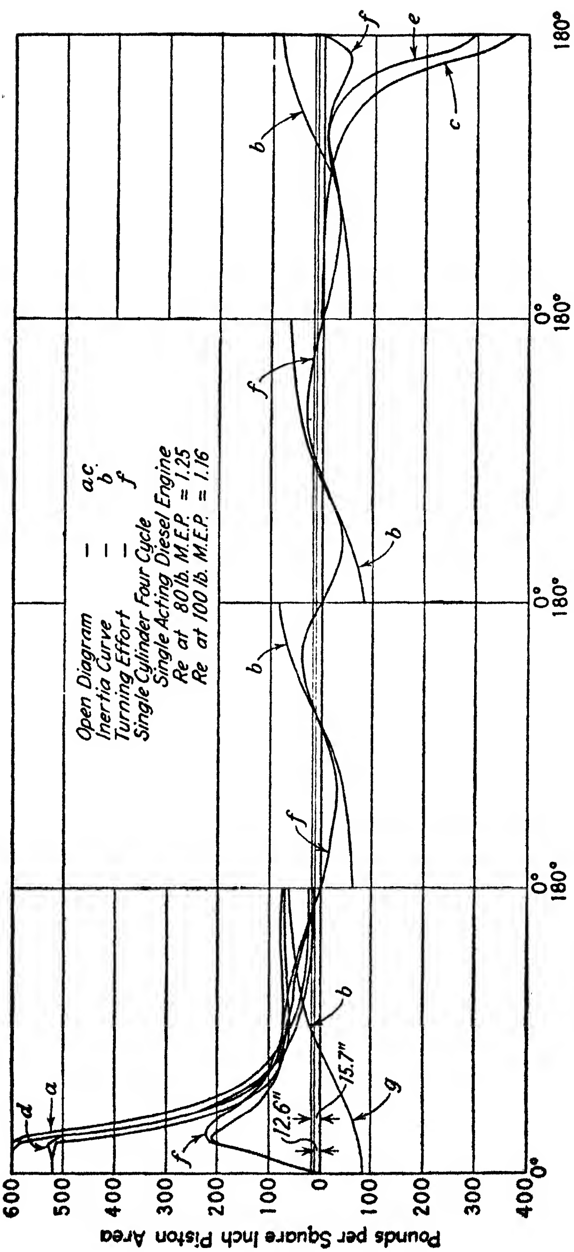

Open Diagram — a.c.
Inertia Curve — b
Turning Effort — f
Single Cylinder Four Cycle
Single Acting Diesel Engine
Re at 80 lb. M.E.P. = 1.25
Re at 100 lb. M.E.P. = 1.16
Pounds per Square Inch Piston Area
12.6"
15.7"

FIG. 137

Although graphical methods are much in use for converting the horizontal or vertical pressure as the case may be to tangential effort, it is best to calculate these factors.

Engine series are designed with a constant value for $x = \frac{l_c}{r}$ (Fig. 138), and tables of tangential components convenient and of permanent value may be calculated for drafting-room use.

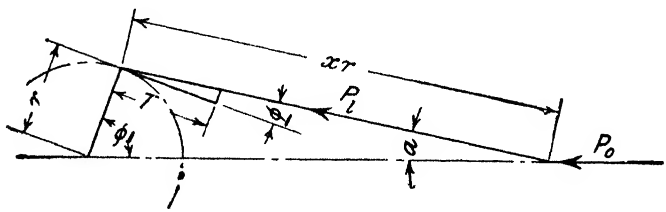


FIG. 138

In most instances it is far more convenient to calculate the tangential components for a number of crank angles than to construct diagrams for each position.

Let

P_0 = the effort on the piston due to working fluid and to the inertia of the moving parts.

P_l = the component of the effort acting along the connecting rod.

$$P_0 = P_l \cos a \quad \text{and} \quad \sin a = \frac{\sin \theta}{x}$$

from which a can be obtained, since θ and x are given.

The tangential component

$$T = P_l \cos \phi$$

and $\quad \phi = 90 - (\theta + a)$

whence $\quad T = \dfrac{P_0}{\cos a} \cos \{90 - (\theta + a)\} = \dfrac{P_0 \sin (\theta + a)}{\cos a}$

The curve f results from d, b, b, and e.

The mean effort equals

$$\frac{MEP}{2\pi}$$

for a single-cylinder four-cycle engine, and

$$n \frac{MEP}{2\pi}$$

for multiple-cylinder engines of any type in which n equals the number of power impulses per revolution.

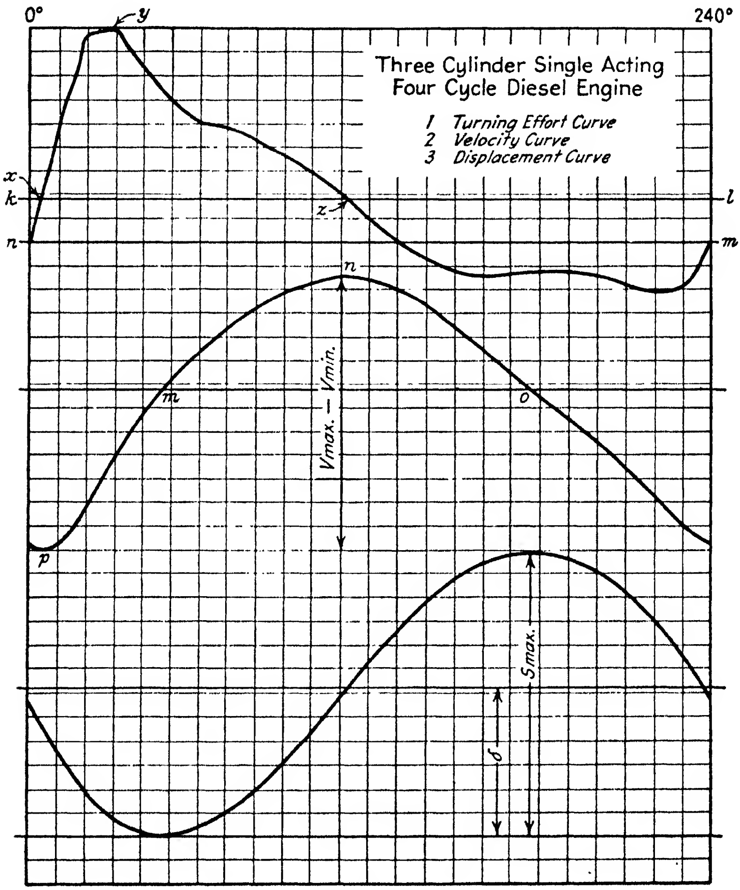


FIG. 139

After the line of mean effort is drawn in, it is advisable to check the work. The sum of the planimetered areas above the line and under the curve *f* should equal those below the line and above the curve *f*. The total excess of energy above the mean must exactly equal the deficiency below the mean in order that the speed of rotation may persist.

There should be no great discrepancy in the completed graph although a slight fairing by raising or lowering of the turning effort curve perhaps the line's width is permissible or even to be expected.

The turning effort curves for multicylinder engines are developed by superimposing an equivalent number of single-cylinder curves with impulses spaced at firing intervals. The algebraic sum of the ordinates of the superimposed curves results in a new curve for each multicylinder arrangement.

Figs. 139 to 141 show, with descriptive legends, a number of useful curves.

Some measure must be established for the excess (= deficiency) of energy to be absorbed by the flywheel. This usually is the ratio

$$R_e = \frac{E_e}{E_m}$$

in which E_e equals the greatest excess of energy per power impulse and E_m equals the mean energy input per impulse. E_m is equivalent to the work shown on the indicator card.

$$R_e = \frac{E_e}{E_m} = \frac{\text{area } xyz}{\text{area } klmn}$$

for the special case of Fig. 139.

TABLE XVIII

TURNING EFFORT CHARACTERISTICS

VERTICAL SINGLE-ACTING FOUR-CYCLE DIESEL ENGINES

Cylinders	Crank angles, °	R_e at 80 lb mip	R_e at 100 lb mip	R_v at 100 lb mip	R_d at 100 lb mip
1	0	1.25	1.16	....	....
2	0	1.00	0.93	....	....
2	180	1.50	1.41	....	....
3	120	0.94	0.85	0.14	0.53
4	180	0.21	0.18	0.13	0.54
6	120	0.30	0.27	0.15	0.52
8	90	0.34	0.30	....	....

Table XVIII gives characteristic values of R_e for four-cycle single-acting solid injection diesel engines.

Table XIX from Dr. Chas. E. Lucke's "Gas Engine Design" brings out a very interesting fact.

TABLE XIX

I. Maximum inertia pressure = 25 lb per sq in.

Number of Cylinders	1	2	3	4	5	6	8
Ratio R_e	1.04	0.85	0.66	0.25	0.36	0.21	0.15

II. Maximum inertia pressure = 50 lb per sq in.

Number of Cylinders	1	2	3	4	5	6	8
Ratio R_e	1.02	0.82	0.65	0.21	0.30	0.18	0.13

III. Maximum inertia pressure = 100 lb per sq in.

Number of Cylinders	1	2	3	4	5	6	8
Ratio R_e	1.01	0.79	0.67	0.64	0.33	0.18	0.14

Inertia forces while they modify all tangential effort curves are of special moment in the case of four-cylinder four-cycle engines. Here two pistons travel simultaneously in one direction while the other two travel in the opposite direction. Their inertias always are in the same sense so that a single inertia curve of four times the value of that for one cylinder could be substituted except for the angularity of the connecting rod which introduces opposed second harmonics.

This great combined inertia force subtracts from the peak effort of each cylinder and contributes a positive impulse when the expansive force is low. The result, with normal inertia values, is a surprisingly uniform turning effort with two low peaks per stroke, Fig. 140, and an R_e value often as good as that for a six-cylinder engine.

In Table XIX calculated by Dr. Lucke for gas engines it will be noted that the R_e value varies greatly with the unit inertia forces (i.e., weight of reciprocating parts or speed of the same) in four-cylinder designs. Three- and six-cylinder engines are not so much affected.

The development of the rational formula for flywheel design is a classic example of the application of elementary physics and algebra to a major design problem.*

* "The Modern Gas Engine and the Gas Producer," by Levin.

Let K equal the degree of irregularity defined by the equation

$$K = \frac{V_1 - V_2}{V} \tag{1}$$

V_1 = maximum rim velocity, feet per second.
V_2 = minimum rim velocity, feet per second.
V = mean rim velocity, feet per second.

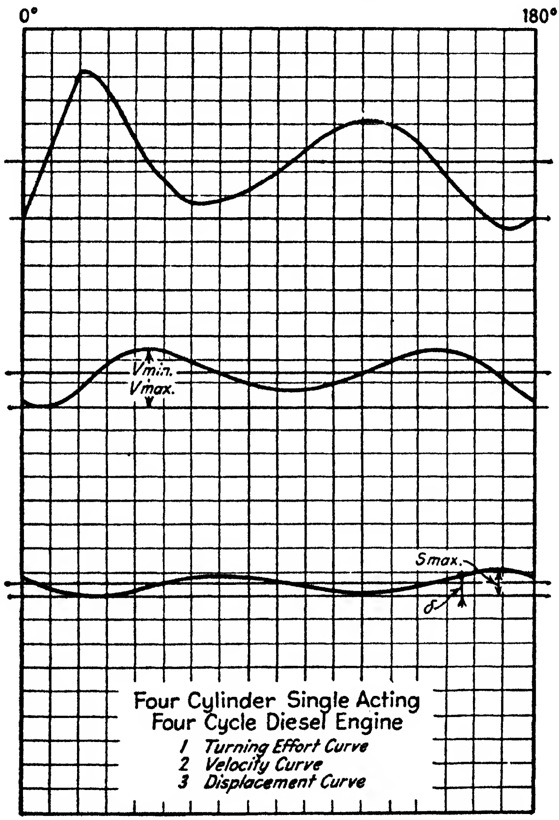


FIG. 140

$$E_e = E_1 - E_2 = \frac{WV_1^2}{2g} - \frac{WV_2^2}{2g} = \frac{W}{2g}(V_1^2 - V_2^2) \tag{2}$$

E_e = energy input or output of flywheel in foot-pounds above or below the mean per cycle.
E_1 = energy of wheel rim at V_1.

E_2 = energy of wheel rim at V_2.
W = rim weight, in pounds.
V = rim velocity at radius of gyration, in feet per second.
g = 32.16 ft per sec^2.

(3) $$E_e = \frac{W}{2g}(V_1 - V_2)(V_1 + V_2)$$

Transposing (1)

(4) $$KV = V_1 - V_2$$

Substituting (4) in (3)

(5) $$E_e = \frac{W}{2g}KV(V_1 + V_2)$$

it is self-evident that

(6) $$V_1 + V_2 = 2V$$

Substituting (6) in (5)

(7) $$E_e = \frac{W}{2g}(KV)(2V) = \frac{W}{g}KV^2$$

The mean energy per cycle $= E_m$

(8) $$E_m = P_0La$$

P_0 = mep of indicator diagram.
L = length of working stroke, in feet.
a = area of piston.

From the turning effort diagram

(9) $$R_e = \frac{E_e}{E_m}$$

(10) $$E_e = R_eE_m$$

Substituting from (8) and (7):

(11) $$R_eP_0La = E_e = \frac{W}{g}KV^2$$

(12) $$W = g\frac{R_eP_0La}{KV^2}$$

Electrical engineers have specified that the maximum pole displacement due to cyclical velocity fluctuations of engine-driven generator units shall not exceed 3° phase on either side of the position of uniform rotation.

The ordinates of the turning effort curve represent force, and since acceleration is directly proportional to force, they may be said to represent acceleration.

The abscissas represent crank angles, and assuming the velocity constant for this purpose they may represent time to the proper scale.

By integrating the turning effort curve as an acceleration time curve we produce the velocity-time curve. By again integrating the velocity-time curve we produce the displacement curve. Figs. 139, 140, 141 show examples of this work.

These graphical integrations are carried out by reckoning first the individual areas *abc* and *cbde*, etc., on the turning effort curve Fig. 141, for example, continuing right across the figure. The areas should be totaled up and each sum should equal the aggregate area of the whole figure *adf* enclosed between turning effort curve and mean effort line to which it belongs.

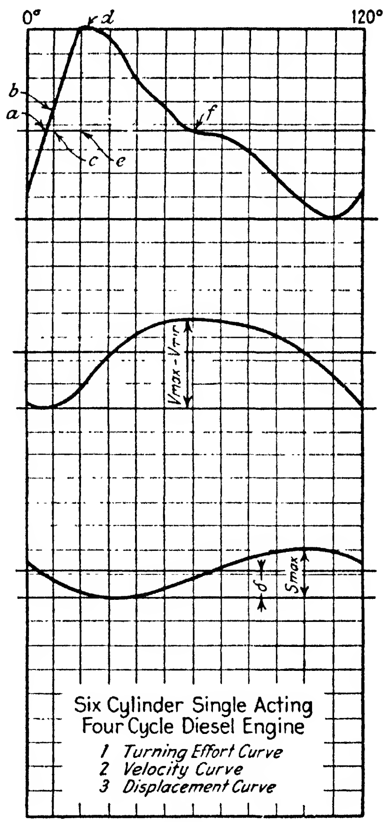


FIG. 141

After this, each increment of area representing force multiplied by time should be added to the preceding total as in Table XX. These increasing or diminishing sums represent ordinates on the velocity curve. If the work is done correctly the ordinate at the beginning will equal that at the end of the cycle.

The process is repeated on the velocity curve to derive the displacement curve.

TABLE XX

Fig.	Area, sq in.	Velocity ordinates
1	0.008	0.008
2	0 203	0.211
3	0.329	0.540
4	0 234	0.774
5	0 120	0.894
6	0 032	0 926
7	−0.001	0.925
8	−0.012	0.913
9	−0 042	0.871
10	−0.105	0.766
11	−0.187	0.579
12	−0.264	0.315
13	−0.238	0.077
14	−0.077	0.000

Next it is necessary to find the scale of the velocity and displacement curves.

The highest ordinate of the velocity curve, Fig. 139, represents

$$V_{max} - V_{min} = \text{feet per second}$$

The length of the velocity curve for each cycle represents the time

$$\frac{60}{Nn} \text{ seconds}$$

in which N equals revolutions per minute and n equals the number of power impulses per revolution.

A mean line is established above the base of the velocity curve at such height that the areas enclosed between the velocity curve and mean line are equal above and below the line. The line of zero displacement is established in exactly the same way.

Areas *mno* above the mean line or *opm* below, Fig. 139, indicate the product of velocity and time. The relation of either of these areas to the great area

$$(V_{max} - V_{min}) \frac{60}{Nn} \text{ is the ratio } R,$$

The velocity curve is constructed solely to determine this relation since

$$S_d = R_v(V_{\max} - V_{\min})\,\frac{60}{Nn}\text{ feet}$$

$$S_d = \text{the maximum pole displacement}$$

Values of R_v for a few engine types are given in Table XVIII.

The upper and lower loops of the velocity curve are not symmetrical, therefore S_d may not be equally divided on each side of the mean.

The ratio R_d of the greatest deviation δ of a pole from the position of mean speed to the total displacement S_d is given for a few cases in Table XVIII.

Then

$$\delta = R_d \times R_v\,(V_{\max} - V_{\min})\,\frac{60}{Nn}\text{ feet}$$

but

$$V_{\max} - V_{\min} = KV$$

so

$$\delta = \frac{60\,R_dR_vKV}{Nn}\text{ feet}$$

When y is the angle of deviation of a point and r_1 = radius of point in feet in the revolving system

$$\delta = \frac{2\,\pi r_1 y}{360}$$

If e is the fluctuation in degrees phase and the standard 60-cycle system is used the duration of each electrical cycle is

$$\frac{360\,N}{60 \times 60} = \frac{N}{10}\text{ degrees of crankshaft travel}$$

The displacement of three electrical degrees d represents

$$\frac{3}{360} \times \frac{N}{10} = \frac{N}{1200}\text{ degrees of crankshaft travel}$$

in which N = revolutions per minute, and the allowable linear pole displacement

$$\delta = \frac{\frac{N}{1200}}{360} \times 2\,\pi r_1\text{ feet} = \frac{2\,\pi r_1 N}{1200 \times 360}$$

Equating

$$\frac{60\ R_d R_v K V}{Nn} = \frac{2\ \pi r_1 N}{1200 \times 360}$$

Substituting

$$V = \frac{2\ \pi r_1 N}{60}$$

and solving

$$K = \frac{Nn}{R_d R_v\ 432{,}000}$$

Electric lighting usually calls for a minimum coefficient of fluctuation of 1/150. A value of 1/250 is desirable. If the average value of R_d for three-, four-, six- and eight-cylinder four-cycle single-acting engines is assumed to be 0.52 and the average value for R_v equals 0.14, Table XXI indicates that it is not necessary to calculate the pole displacement for multicylinder engines except for unusual cases.

TABLE XXI

VALUES OF $K = \dfrac{N_n}{R_d R_v\ 432{,}000}$

No. cylinders	K at 300 rpm	K at 720 rpm	K at 1200 rpm
3	$\frac{1}{80}$	$\frac{1}{33}$	$\frac{1}{20}$
4	$\frac{1}{60}$	$\frac{1}{25}$	$\frac{1}{15}$
6	$\frac{1}{40}$	$\frac{1}{17}$	$\frac{1}{10}$

Pole displacement sets up directly proportionate cross currents. These cross currents create a restoring force so it is quite possible for two rotating crankshaft and armature systems to swing harmonically in resonance if the natural frequencies of the systems accord with their forced frequencies due to the nature of the engine turning effort or some simple multiple of it.

Formulas and factors are issued by generator manufacturers to predetermine suitable WR^2 values for the rotating systems to avoid resonance. These formulas have changed repeatedly, so they will not be given here. It is advisable, however, to submit to the electrical machinery builder a minimum and maximum desirable flywheel weight. Usually resonance can be avoided by choosing a weight between the two.

Engine builders frequently are asked to provide flywheels such that a momentary overload may be carried with a specified speed drop by virtue of the kinetic energy stored in the wheel alone. If sufficient time is allowed for recovery before the next peak is due this feature often is useful in preventing over-regulation with accompanying high cylinder pressure and detonation. However, with any reasonable flywheel weight the kinetic energy is available but momentarily.

For instance, let it be desired to increase the normal full-load horsepower of a diesel engine 25 per cent for 1 sec at peak load of a power shovel cycle by extraction of energy from the flywheel. A drop of speed not exceeding 6 per cent during this period will be allowed.

One horsepower equals 550 ft-lb sec, 25 per cent of which equals 137.5 ft-lb sec. Thus, each cycle, 137.5 ft-lb. of energy are to be taken from a flywheel running with a maximum velocity of 6000 ft per min or 100 ft per sec for each horsepower of engine capacity and in 1 sec of time.

E_x = energy extracted each shovel cycle.
E_1 = energy in wheel at 100 ft per sec at V_1 velocity.
E_2 = energy in wheel at 94 ft per sec at V_2 velocity.
W = weight of wheel rim required for each horsepower of engine rating.
g = gravity = 32.16 ft per sec².

$$E_x = E_1 - E_2 = \frac{WV_1^2}{2g} - \frac{WV_2^2}{2g}$$

$$E_x = 137.5 = \frac{W \times 100^2}{64.32} - \frac{W \times 94^2}{64.32}$$

$$= 137.5 = \frac{(100^2 - 94^2)\,W}{64.32} = \frac{1164\,W}{64.32}$$

$$W = \frac{64.32 \times 137.5}{1164} = 7.6 \text{ lb per rated hp of engine}$$

With a flywheel of this weight and velocity the possible kinetic torque is as follows for shock loads:

Duration of output, sec	Speed drop, %	Torque increase, %
1	6	25
$\frac{3}{4}$	6	$33\frac{1}{3}$
$\frac{1}{2}$	6	50
$\frac{1}{4}$	6	100

This wheel is unusually heavy, giving a value of K equal to about 1/400 for six- or eight-cylinder four-cycle engines.

CHAPTER XIV

THE CRANKSHAFT

Practically all attempts to use cast material for solid or continuous crankshafts have resulted in disaster. Cast steel regardless of its analysis or physical properties is an unworked material no more fit for such service than the rolling mill ingot. Present day engines having many cylinders and operating at high speed require the toughest material available.

Small shafts are drop-forged. Large one-piece crankshafts are best made from forging ingots which are distinguished from rolling ingots in that they are carefully annealed in the process of cooling when they are intended for shipment. If forgings are to be made at the steel works they can, of course, be produced from the hot ingot direct from the soaking pit. The ingot usually is cogged down under a hydraulic press, sometimes under a hammer, to the section of the crankarms. This should effect a reduction of area to $\frac{1}{4}$ the original section if ingots of such size are available. Cheaper shafts are made from hammer blooms which have been rolled to the cross-section of the crankarms. Although this is a convenient way to purchase material it is not approved for important work. If shafts are so made the forge should have certain knowledge of the amount of reduction of cross-sectional area that was effected by rolling at the mill. The size of the ingot should be specified as well as the size of the bloom.

At the forge the end bearings and flywheel fit are first reduced after which the gaps for the throws are laid out, sawed, drilled and broken out. Little or no forging is done on the intermediate bearings. The shaft is next sent to a lathe where these bearings are rounded up leaving plenty of stock for finish. The throws are then twisted into proper relation, the intermediate bearings being heated and twisted one at a time. The twisting may be done by clamping one set of arms under press or hammer and applying torque to the next with a great spanner hooked to a hoist or crane or it may be done in a special back-geared motor-driven machine built especially for the purpose. In the first case the section to be twisted is heated in the furnace; in the latter a ring of oil burners under oil blast is set up on the machine. In either case the correct heat is determined by an optical pyrometer, and this heat must reach well into the arms or the shaft will show helical ghost lines when finished owing to axial shear under torsion. Sometimes such lines are cause for rejection of a shaft. More lenient inspection, perhaps warranted if shaft stresses are low, rejects shafts showing ghost lines only if, at each revolution, the lathe chip breaks at the lines, indicating actual slip of the "fiber" of the shaft.

After twisting, the shaft is rough-turned all over and heat-treated. Heat treatment should consist of a thorough anneal in the modern sense, that is, of "air quenching" from about 1550°F or just above the exact critical temperature and drawing at from 1000° to 1200°F.

As heavy shafts are designed for stiffness and are very lightly stressed, alloy steel rarely is used. Clean, homogeneous material is far more important than special composition. Hardness of journals is desirable but it is so opposed to non-fatigue qualities that it is secondary.

Built-up shafts are used on large engines. The arms and crank-pin are often in one piece with the journals shrunk in. This plan is followed instead of building up arms and pin because the relation of crank radius to pin diameter is such that it is not possible to leave a safe section of metal standing between the bores for pin and journal as in steam engine practise. The journals are swelled where they enter the cranks to compensate for the weakening due to the keyseat. The cranks may be steel castings or steel forgings and usually are made to Lloyds' or American Bureau specifications both as to material and minimum dimensions.

DIE-FORGED SHAFTS

The increasing size of forge shop presses has recently made it possible to procure die-pressed shafts of large dimensions. Fig. 142 shows an etched section of a common shaft and Fig. 143 a like section of a die forging. The improvement in grain section is manifest.

The operations of die-pressing heavy shafts, a throw at a time, are shown in Fig. 144.

COUNTERWEIGHTING

Counterweighting of six- or eight-cylinder shafts at any speed used for stationary or marine oil engines today is rarely neces-

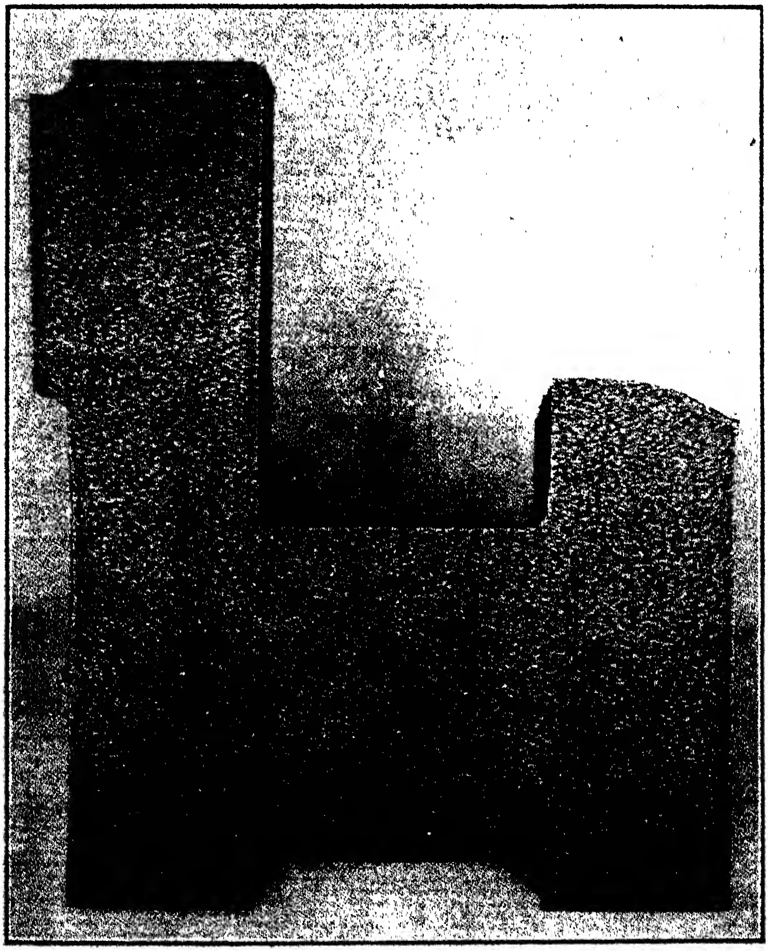


Fig. 142

sary for bearing relief. It contributes nothing to the general balance of the engine and very seriously lowers the critical speeds

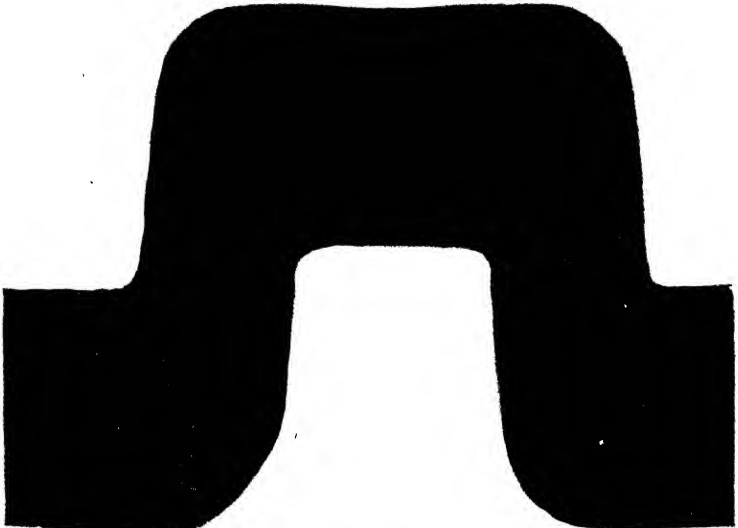


FIG. 143

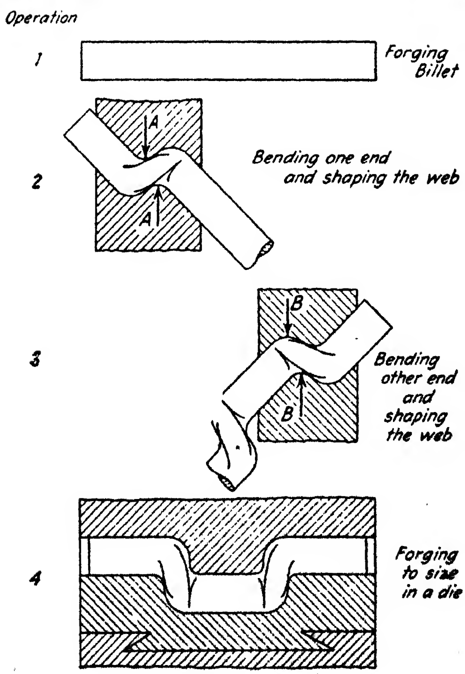


FIG. 144

of torsional vibration. It should never be permitted except in the special cases where it is required in unfortunate design for lowering a critical from the operating range.

TORSIONAL VIBRATION OF SHAFTS

The crankshaft like all weighted elastic bodies has a natural frequency of vibration. That is, if twisted and released it will oscillate at a certain rate until friction, hysteresis and other damping forces bring it to rest. If, however, the repeated harmonic impulses to which the shaft is subjected are in step with the natural torsional oscillations of the shaft the vibrations may be sustained and magnified to a dangerous degree.

With steadily advancing speeds, shafts designed to Lloyds' or American Bureau formulas are to be regarded as of minimum dimensions. It is advisable to examine every shaft to make certain that it fulfills the survey requirements and then to make a further dynamic analysis which may reveal the necessity of stouter proportions.

Elaborate stress calculations such as Ennslin* undertook some thirty years ago have been superseded by researches into the mass elastic properties of the shaft system. Where combined bending and twisting stress once reached values as high as 12,000 lb and even 15,000 lb per sq in. in modern six- and eight-cylinder shafts designed for torsional stiffness they fall far lower and may even be neglected. Dynamic stresses or more particularly the avoidance of dangerous critical speeds are the ruling consideration, in designing engines having three or more power impulses per revolution.

OPERATION OF ENGINE WITH CRITICALS

It was once a part of the operator's duty to recognize critical speeds if warned of them by the manufacturer. In stationary plant, turbo or diesel engine, a centrifugal pump or turbo blower was run quickly through the critical ranges to normal speed. Designers sought only that the operating speed of the equipment should be safely removed from the vibration zones. This is a passing phase in the design of all machinery. Modern designs

* "Mehrfach gelagerten Kurbelwellen," by Ennslin.

are guaranteed free of all criticals of orders superior to control by natural damping.

A full instrument equipment for the observation and recording of torsional vibrations is desirable but not always necessary. In most instances dangerous cases are all too evident. No other form of vibration, and there are many, equals it in disturbance.

When large engines operate in the range of a major critical, the noise and vibration of the whole structure are quite incredible. If the camshaft is driven from the free end of crankshaft, timing gears are excessively noisy and may be very short-lived. General engine vibration is always a little more prominent at the free end although it may be felt all over the engine and foundation. At the flywheel end the best design and fitting may fail to keep couplings firm. As soon as motion within the coupling is possible heating sets up, warming the shaft, with surprising quickness. At the node on the crankshaft, bearings may heat and in severe cases burn out.

RECLAIMING OLD DESIGNS

Turning effort curves show that the maximum twisting effort is very little increased if six or eight cylinders are used instead of four. On this basis some old models designed and satisfactory for three or four cylinders were extended to provide lines of six- and even eight-cylinder engines. Serious criticals developed. The bearings usually had been made small in diameter for low velocity and long to reduce unit pressure with the very logical idea that friction work, being the product of velocity and frictional resistance, thus would be reduced with a corresponding reduction of heat to be dissipated. This idea was quite rational when the splash or drip systems of lubrication were in common use. Modern engines are fully enclosed, and the lubricating oil is pumped in quantity through all major bearings. Its heat is extracted in a cooler. The oil is therefore both lubricant and cooling medium. While short large-diameter journals may tend toward lower mechanical efficiency forced lubrication is no doubt an offset. At any rate the spindle idea is no longer applicable to multicylinder engines. Fortunately, many old shafts may be redesigned to fit the original bed plate. Some increase of diameter may be required, but much can be done by taking inches off the length of intermediate bearings and adding to the arm width.

Great gain, sometimes to be measured in feet, can be made by shortening the end bearing and the flywheel extension placing the wheel as close to the crank as possible.

NEW DESIGN

Before a new shaft is dimensioned a thorough understanding of what is expected should be established among all interested parties.

First of all the maximum number of cylinders should be decided upon, bearing in mind that a serious cost handicap may be placed upon engines of three and four cylinders if fitted with shafts designed for eight-cylinder work. Again if a lighter and cheaper four-cylinder shaft is designed the extra cost of equipment to produce two shaft models for one series of engines may offset any saving.

The maximum speed expected should be set. A decision should be made as to whether the engine will be offered for marine work only, for stationary work only or for both services from which the heaviest flywheel ever required may be calculated and its influence on critical speeds considered.

Finally a prediction should be made as to the performance of the shaft in various cylinder combinations, under all the possible commercial conditions which it may be called upon to satisfy and frankly placed before the management.

Most severe cases of torsional vibration in moderate-speed engines are avoidable without abnormal crankshaft proportions. The utmost care should be used to keep the shaft line up between engine flywheel and generator or other driven machine rotor as stiff and as short as possible so that the two masses may act as one. It is good practise to fit into the generator armature or rotor the largest shaft that the manufacturer's design permits. If the flywheel is carried on a stub flanged to the crankshaft this too should be very short and stiff. It costs little to make such stubs 25 to 30 per cent larger than the end journal.

ORDERS OF VIBRATION

The word "order" indicates the number of natural vibrations of the crankshaft in the time of one revolution. A "third order critical" implies that the engine is operating at such speed that three natural vibrations occur per crankshaft revolution which is also the exact number of forced vibrations for a six-cylinder four-cycle single-acting engine. One-node vibrations usually are all that need be considered for the crankshaft alone.

The amplitudes possible under the various orders of vibration can be revealed only by harmonic analysis, but a good rule is to avoid entirely a shaft frequency such that the number of free oscillations equals the number of power impulses, also $1\frac{1}{2}$ times or twice the number of impulses. It is well to beware of a frequency equal to $2\frac{1}{2}$ times the number of impulses. If the shaft frequency is over 3 times that of the impulses there is no likelihood of trouble.

VIBRATION FORMULA

The derivations of the vibration equations as usually presented are quite sophisticated. The following treatment is based on laws so often in use and so easily demonstrated with elementary physical laboratory apparatus that they are axiomatic.

A simple exposition of harmonic vibration may be made if it be remembered that harmonic motion is one component of circular motion. This is demonstrated by setting in motion two pendulums of the same length. If one is allowed to swing in a circle and the other to oscillate both will complete a cycle in the same time.

Let f = force required for unit linear displacement.

x = displacement.

fx = force at actual displacement.

$\frac{fx}{2}$ = mean force to produce displacement.

$\frac{fx}{2} \times x = \frac{fx^2}{2}$ = mean energy or work.

Since the potential energy of displacement is turned into an equal amount of kinetic energy in the restoring movement,

$$\frac{fx^2}{2} = \frac{1}{2}mv^2$$

m = the mass involved

or
$$fx^2 = mv^2$$

Also
$$t = \frac{2\,\pi x}{v}$$

$$v = \frac{2\,\pi x}{t}$$

$$fx^2 = m\,\frac{4\,\pi^2 x^2}{t^2}$$

t = time of one oscillation or revolution.
v = velocity on the circular orbit.

$$t^2 = \frac{4\,\pi^2 m}{f}$$

$$t = 2\,\pi\sqrt{\frac{m}{f}}$$

From this the equation for torsional vibration may be deduced. The angle of torsion

$$\theta = \frac{32\,lfr}{\pi G d^4}$$

θ is expressed in radians so that linear displacement on the circumference

$$x_1 = \theta r$$

and
$$\theta = \frac{x_1}{r}$$

Thus

$$x_1 = \left(\frac{32\,lfr}{\pi G d^4}\right) r = \frac{32\,lfr^2}{\pi G d^4}$$

When $x_1 = 1$

$$f = \frac{\pi G d^4}{32\,lr^2}$$

In the fundamental equation

$$t = 2\,\pi\sqrt{\frac{m}{f}}$$

The polar moment of inertia J of the masses involved, when reduced to the crank radius, equals mr^2 and

$$m = \frac{J}{r^2}$$

So we may write

$$t = \frac{2\,\pi\sqrt{\frac{J}{r^2}}}{\sqrt{\frac{\pi G d^4}{32\,l r^2}}} = 2\,\pi\sqrt{\frac{32\,Jl}{\pi G d^4}}$$

The units used in the above are

l = inches from node to center of mass on shaft.
$J = mr^2 = \frac{W}{g} r^2$ pound inches squared.
r = inches.
G = 11,000,000.
d = inches, shaft diameter.
t = seconds, period of oscillation.
g = 386 in. per sec^2.

RECIPROCATING MASSES

A number of assumptions have been put forward for the inclusion of the mass of reciprocating parts into the summation. These compromises make the best of a bad situation as the relative position and the relative effect of the reciprocating masses at any phase of a shaft vibration cannot be dealt with accurately by any mathematical means yet developed. The commonly accepted solution* is to reduce the rotating and reciprocating mass to a value at the crankpin equal to

$$J_2 = \left\{m_1 + \tfrac{1}{2}\,m_2\left(1 + \frac{r^2}{4\,l_c^2}\right)\right\}r^2$$

where r = crank radius.
l_c = connecting-rod length.
m_1 = rotating masses of rod.
m_2 = reciprocating mass of rod and piston.

THE EQUIVALENT LENGTH

It is also necessary to reduce the length of the crankshaft to an equivalent length of straight shaft having the same torsional properties. This is sometimes done by actually securing the

* "Vibration Problems in Engineering," by Timoshenko, page 158.

flywheel against rotation and loading a rigid lever attached to the shaft at the other end, the angular deflection of the lever being measured. Such practise is a valuable check on calculations but something else is needed for engines in the design stage.

It has been shown mathematically that the deflections of center cranks differ from those of end cranks. It has been demonstrated both mathematically and by actual test that the torsional properties of a cranked shaft vary with the bearing clearance. These are refinements necessary to the mathematical investigation of existing shafts, but when new shafts are being designed with criticals of the dangerous orders safely removed from the operating range a good approximation sustained by practise is most useful.

The following checks fairly well when combined with the other assumptions given in this text.

Let

$C_1 = \dfrac{\pi d_1^4 G}{32}$ = torsional rigidity of journal of diameter $= d_1$.

$C_2 = \dfrac{\pi d_2^4 G}{32}$ = torsional rigidity of pin of diameter $= d_2$.

$B = \dfrac{hc^3 E}{12}$ = flexural rigidity of crankarm of width $= h$, and depth $= c$.

$G = 11{,}000{,}000.$

$E = 30{,}000{,}000.$

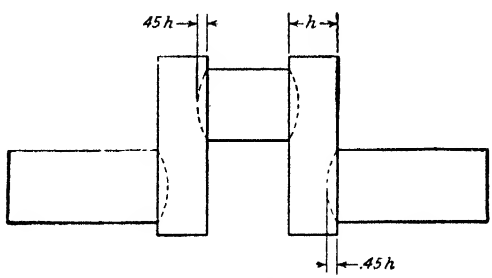


FIG. 145

The torsional strain on pin and journal extends into the arms as indicated in Fig. 145. To account for this, consider crankpin length $a_1 = a + 0.9\,h$, and journal length $2\,b_1 = 2\,b + 0.9\,h$.

When

$$a = \text{length crankpin}$$
$$b = \tfrac{1}{2} \text{ length main journal.}$$

The angle of twist under a torque moment M_t will then be

$$\theta = \frac{2\,b_1M_t}{C_1} + \frac{a_1M_t}{C_2} + \frac{2\,rM_t}{B}$$

$2\,r$ accounts for the length of both arms, r being the crank radius.

If l is the length of an equivalent straight shaft

$$\theta = \frac{M_tl}{C}$$

$$\frac{M_tl}{C} = \frac{2\,b_1M_t}{C_1} + \frac{a_1M_t}{C_2} + \frac{2\,rM_t}{B}$$

C_1 and C_2 are equal in most modern shafts. C may be assumed equal to C_1 and C_2.

Cancelling

$$l = 2\,b_1 + a_1 + \frac{2\,rC}{B}$$

LOCATING THE NODE IN A TWO MASS SYSTEM

Still another assumption is required for calculation of the frequency of the system. The J_1 value for the flywheel results from the coefficient of fluctuation applied to the flywheel formula. The J_2 value for crank and reciprocating parts of multicylinder engines is merely the sum of the J value for each set. In other words, the masses of the crankshaft and the reciprocating parts are considered as concentrated at the center of gravity of the crank system.

As all masses on the side of the node away from the flywheel are considered as one mass the vibrating length of the shaft l_2 is measured from the middle of the center bearing to the node.

Then

$$J_1l_1 = J_2l_2 \quad \text{when} \quad l_1 + l_2 = l$$

POLAR MOMENT OF INERTIA OF MASSES

Great care should be used in evaluating the J values. The center of gravity should not be used as the radius of gyration of any of the masses involved.

The following theorem is useful in deriving the moments of inertia of the crank members.*

> The moment of inertia of an area with respect to an axis perpendicular to its plane is called the polar moment of inertia of the area. The moments of inertia with respect to each of two axes at right angles and lying in the plane of the area are sometimes called rectangular moments of inertia to distinguish them from the polar moment of inertia. THEOREM: The polar moment of inertia of an area is equal to the sum of the rectangular moments of inertia with respect to any two axes at right angles lying in the plane and passing through the same point as the axis for the polar moment.

The flywheel sections should be treated as carefully for accuracy of the J value. For the elimination of the uncertainty of spoke deflections and vibrations, the disk-type wheel is very desirable.

AN EXAMPLE IN TORSIONAL VIBRATION

This system was used in a six-cylinder four-cycle single-acting engine of 16-in. bore and 20-in. stroke. The shaft was counterweighted, for some unknown reason.

The polar moments of inertia of the rotating parts were approximated by dividing arms and the like into small masses and taking the sum of all the small J values. This is a necessary procedure when arms and weights are of such form that they cannot be divided into fewer geometrical elements.

The total moment of inertia of the crankarm and fantail areas is 116,432 sq in. in.^{2} The thickness of one double arm crank and counterweight is 11 in. so that the polar moment of area of the rotating volume is 1,280,752 cu in. in.^{2}, and by weight at 0.28 lb per cu in.

$$0.28 \times 1{,}280{,}752 \text{ cu in. in.}^2 = 358{,}611 \text{ lb in.}^2$$

The polar moment of inertia of the cross-section area of crankpin is

$$I_p = \frac{\pi d^4}{64} + \left[\frac{\pi d^4}{64} + \left(\frac{\pi d^2}{4} \times x^2\right)\right]$$

in which $x = r = 10$ = the crank radius, and $d = 10$ in. = the crankpin diameter.

* "Practical Calculus," by Palmer, page 343.

Then

$$
\begin{aligned}
I_p &= \frac{\pi d^4}{32} + \left(\frac{\pi d^2}{4} \times x^2\right) \\
&= \frac{3.1416}{32} \times 10^4 + \left(\frac{3.1416}{4} \times 10^2 \times 10^2\right) \\
&= 981 + 7854 = 8835 \text{ sq in. in.}^2
\end{aligned}
$$

The volumetric polar moment of inertia is

$$10 \times 8835 = 88{,}350 \text{ cu in. in.}^2$$

The weight polar moment of inertia is

$$88.350 \times 0.28 = 24{,}738 \text{ lb in.}^2$$

To this must be added the moment of inertia of the rotating weight of the connecting rod or $272 \times 10^2 = 27{,}200$ lb in.^{2}

Also the reduced reciprocating weight, thus

$$\tfrac{1}{2} W\left(1 + \frac{r^2}{4\, l_c^2}\right) r^2 = W_r^2$$

The total $\sum W_r^2$ is then

W_r^2 of 2 crank arms and counterweights	=	358,611 lb in.^{2}
" 1 crankpin	=	24,738 "
" 1 connecting rod end (rotating weight)	=	27,200 "
" set reciprocating parts	=	59,451 "
		470,000 "

To reduce this to mass moment of inertia we divide by

$$12 \times g = 386 \text{ in. per sec}^2$$

Thus $J = 470{,}000 \div 386 = 1218$ lb in.^{2} in the form for application to the vibration formulas.

For the six crank systems $J = 6 \times 1218 = 7308$ lb in.^{2}

The term "moment of inertia" with its various implications is not warranted to clarify any discussion. The chief object of introducing this calculation is to record the units used as the work progresses. The symbol J has not been given a new subscript for each phase of its development as that would seem only to confuse the text by requiring frequent back reference.

The usual advice is to ignore the shaft journals, which may well be done as the tendency of their approximation is to give a rather lower frequency than that realized in practise. We will include them here, however. The position of the node is not yet ascertained. It may be assumed that it will fall aft of the after center bearing so only 5 of these will be included or 60 in. of 10-in. shaft.

For this section $I_p = \frac{\pi}{32} d^4 = 981$ sq in. in.^{2}

For the shaft $J = 981 \times 60 = 58{,}860$ cu in. in.^{2}

In weight units $= 58{,}860 \times 0.28 = 15{,}557$ lb in.^{2}

In mass units $= \frac{15{,}557}{386} = 40$ lb in.^{2}

Thus the total J value of the crank system

$$= 7308 + 40 = 7348 \text{ lb in.}^2$$

That of the flywheel $= 21{,}700$ lb in.^{2}

That of the system $= 29{,}048$ lb in.^{2}

Next in order is the equivalent length of shaft.

For any single crank

$$l_e = 2\,b_1 + a_1 + \frac{2\,rC}{B}$$

$$= [12 + (0.9 \times 5.5)] + [10 + (0.9 \times 5.5)] + \frac{2\,rC}{B}$$

$$= 31.9 + \frac{2 \times 10 \times 3.1416 \times 10^4 \times 12{,}000{,}000}{32}$$

$$\times \frac{12}{5.5 \times 14^3 \times 30{,}000{,}000} = 38.14$$

The length from the center of the system to the end of flywheel keyseat is then

$$(3 \times 38.14) + 8 = 122.42 \text{ in.}$$

The length from the center of the system to the node is

$$\frac{21{,}700}{29{,}048} \times 122.42 = 91.44 \text{ in.}$$

$$t = 2\,\pi \sqrt{\frac{32\,Jl_e}{\pi G d^4}}$$

$$= 6.2832\sqrt{\frac{32 \times 7348 \times 91.44}{3.1416 \times 12{,}000{,}000 \times 10^4}} = 0.047 \text{ sec}$$

$$\frac{1}{t} = 21.3 \text{ vibrations per second}$$

$$60\,t = 1278 \text{ vibrations per minute}$$

The sixth order critical speed will then be

$$\frac{1278}{6} = 213 \text{ rpm}$$

As a matter of fact, this engine developed its sixth order critical at a speed about 3 per cent higher than calculated. Removal of the counterweights made it possible to operate with complete satisfaction at the desired speed of 257 rpm.

Though sometimes it is claimed that criticals can be predicted within a revolution or two this is true only when experimental data on the shaft are available.

Many approximations enter into the calculation of the equivalent length, therefore such accuracy if obtained is accidental.

APPENDIX

REDUCING CALCULATIONS IN DESIGNING CRANK-SHAFT MEMBERS*

Reciprocating engine crankshaft calculations present some interesting mathematical properties when analyzed. For any series of engines assume that:

1. The crankshaft and pin diameters bear a simple and constant ratio to the cylinder diameter. For example, the crankshaft may be 6/10 or 5/8 or 7/10 the cylinder diameter.

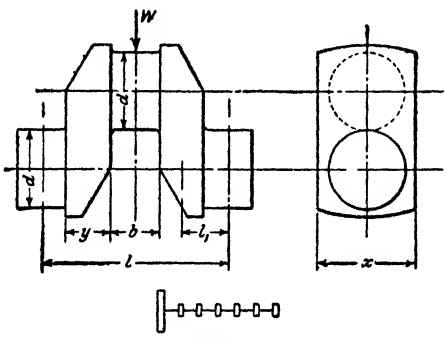


FIG. 1

2. The cylinder centers or main bearing centers also bear a simple and constant ratio to the cylinder diameter. As an instance, the main bearing centers may be separated 1.5 or 2 times the cylinder bore.
3. The crankarm sections are as in 1 and 2.
4. The stroke bore ratio is constant for all sizes.
5. All unit bearing pressures are alike for the same location.
6. All engines have the same number of cylinders.

It remains now to demonstrate that these are rational and not "rule o' thumb" proportions.

Basic formulas in the following may be found in any book on machine design or in any engineers' handbook. They are so long established that they need no exposition. The symbols, so far as possible, are those commonly used.

* By permission of *Machine Design*, in which magazine this article appeared, December, 1930.

CRANKSHAFT AS A BEAM

The first and simplest assumption in designing a crankshaft is that each section is a beam with the piston pressure load imposed at its center and finding support at the middle of each crankshaft bearing, or "main bearing."

The bending moment M_b of a beam of this class is $\frac{1}{4}\ Wl$, in which l is the distance between supports and W is the impressed load. Also, $M_b = S \times I/c$, in which S is the maximum fiber stress and I/c the section modulus.

I/c for a round section such as the crankpin equals $\pi/32\ d^3$ or $0.1\ d^3$ approximately, d being the crankpin diameter. Thus

$$\frac{1}{4}\ Wl = 0.1\ d^3S \quad \text{or} \quad S = \frac{\frac{1}{4}\ Wl}{0.1\ d^3}$$

Assuming that a series of crankshafts is to be designed in which d/D and l/D are constant, D being the cylinder diameter,

$$W = \frac{D^2\pi}{4} \times 600$$

or the area of the piston multiplied by the maximum cylinder pressure.

$$l = K_lD \quad \text{and} \quad d = K_dD$$

in which K_l and K_d are constants chosen by the designer after investigation of one size of unit.

Then substituting values in the basic equations,

$$S = \frac{\frac{1}{4} \times \frac{D^2\pi}{4} \times 600 \times K_lD}{0.1(K_dD)^3}$$

$$= \frac{600 \times \pi \times 10 \times K_l}{4 \times 4 \times K_d^3}$$

or

$$S = 375\frac{\pi K_l}{K_d^3}$$

Thus it is evident that for constant ratios d/D and l/D, S also is a constant and when the ratio l/D is kept constant it is rational to make the crankpin diameter a fixed percentage of the cylinder diameter.

CRANKSHAFT IN TORSION

The diesel engine with its sustained constant pressure phase brought in the necessity of considering the twisting stresses.

For a multicylinder solid injection engine the maximum tangential pressure is about 275 lb per sq in. of piston area. This results in a twisting moment

$$M_t = \frac{D^2\pi}{4} \times 275 \times \frac{s}{2}$$

where $s/2 = r =$ the crank radius.

If the stróke-bore ratio is made constant for a proposed series of engines as $s/D = 1.25$, we may say

$$M_t = \frac{D^2\pi}{4} \times 275 \times \frac{1.25\ D}{2}$$

The section modulus for torsion $I_p/a = 0.2\ d^3$, a being the cross-sectional area of crankpin and I_p the polar movement of inertia; or if the shaft diameter is the same as the crankpin diameter,

$$\frac{I_p}{a} = 0.2(K_dD)^3$$

and

$$S = \frac{M_t}{\dfrac{I_p}{a}} = \frac{\dfrac{D^2\pi}{4} \times 275 \times \dfrac{1.25\ D}{2}}{0.2\ K_d^3D^2}$$

$$= \frac{\dfrac{\pi}{4} \times 275 \times \dfrac{1.25}{2}}{0.2\ K_d^3}$$

or

$$S = \frac{675}{K_d^3}$$

Thus it also is quite rational to vary the shaft diameter directly with the bore. This conclusion will not be changed by investigation of the effects of combined stresses.

In like manner it may be shown that the bearing lengths for shafts of like design are directly proportional to the cylinder diameter.

If a unit pressure of 1500 lb per sq in. is desired in designing the crankpin its projected area will be $600/1500 = 0.4$ that of the piston area. Its length b then will be

$$b = \frac{0.4 \times \frac{\pi}{4} D^2}{K_dD} = \frac{0.314\ D}{K_d}$$

Many designers make the intermediate crankshaft bearings the same length as the crankpins.

The crankarms next may be analyzed for bending. Arm section modulus against bending the flat way will be

$$\frac{I_{xy}}{c} = \frac{1}{6}xy^2$$

in which x = thickness of slab, y = width of arm.

The moment arm l_1 in bending is measured from the center of the crankshaft bearing to the neutral axis of the arm. Since the crankpin and center bearing lengths are directly proportional to the cylinder bore as well as the total length l between each pair of center bearings, it follows the room remaining for y also must be proportional to D and may be expressed:

$$y = K_yD \quad \text{and} \quad l_1 = K_{l_1}D$$

We may assume that $x = K_xD$

The bending moment

$$M_{b_1} = \frac{W}{2} l_1 = \frac{\frac{\pi}{4} D^2 \times 600 \times K_{l_1} D}{2}$$

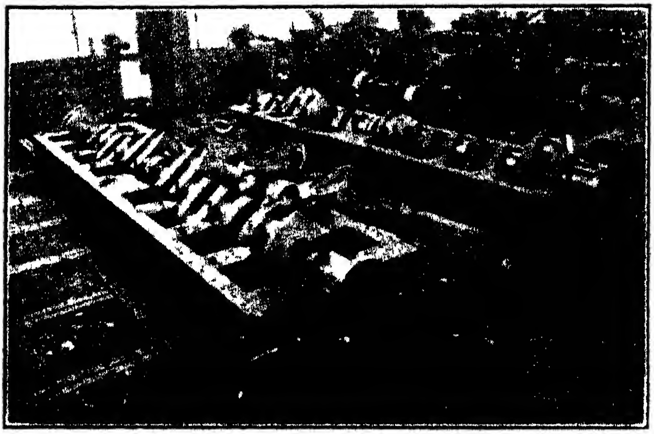


FIG. 2

since each arm receives half the piston load in cantilever fashion.

$$\frac{I_{xy}}{c} = \frac{K_x D \times (K_y D)^2}{6}$$

$$S_1 = \frac{M_{b_1}}{\frac{I_{xy}}{c}} = \frac{\frac{\pi}{4} D^2 \times 600 \times K_{l_1} D}{2} \times \frac{6}{K_x D \times K_y^2 D^2}$$

$$S_1 = \frac{450\,\pi K_{l_1}}{K_x K_y^2}$$

The terms in the equation all being constants, the transverse static stresses will be constant in the arms of any series of shafts in which the arm slab dimensions are fixed simple ratios of the cylinder bore.

THE RATIONAL FLYWHEEL FORMULA

In torsional vibration the shaft is a torsion spring loaded with a series of masses, usually flywheel, cranks, reciprocating parts, etc. The rational flywheel formula is

$$w = \frac{g R_e P_8 A}{f V^2}$$

in which w = the weight in pounds reduced to the radius of gyration of the rim.

R_e = the ratio of excess energy to the mean energy as shown by the turning effort curve. It is constant for each cylinder arrangement in any series of engines if the inertia values are the same, which is a possible achievement.

P = the mep, developed in one power cylinder.

s = the stroke in feet and with a fixed stroke bore ratio a straight-line function of D, the bore of the cylinder.

Thus $s = K_sD$, A = the piston area or $\pi/4\ D^2$, f = the coefficient of irregularity, usually a constant for any given class of service.

V = the flywheel rim velocity in feet per second at the radius of gyration of the rim and usually is constant for any series of engines as it is economical to use the greatest velocity consistent with the wheel construction. Marine engines are an exception to this on account of space, but even then the wheel diameter often is made a fixed ratio to either bore or stroke. $g = 32.16$ ft per sec².

With values chosen for these factors we may then derive a single constant embracing all of them and write $w = K_wD^3$, since all the terms are constant except A which varies as D^2 and S which varies as D.

Let J_w = the mass polar moment of inertia of the flywheel in pounds feet squared. Then:

$$J_w = wr^2 = K_wD^3 \times \left(\frac{K_sD}{2}\right)^2$$

or

$$J_w = K_jD^5$$

In the same way crankshaft masses have been shown to vary as the cube of D since they are directly proportional in three dimensions to D. So also are the reciprocating parts of the rod and the piston and rotating part of rod roughly proportional in weight to D^3 for the same reason.

We then may write for the summation of the wr^2 values on the crankshaft system, $J_C = K_cD^5$.

Then in the expression for the period of free vibration of a system of two masses on a shaft with a node between, assuming the crank system considered as one equivalent mass and the flywheel another

$$T = \frac{2}{r_s^2}\sqrt{\frac{2\pi o}{G}}\sqrt{\frac{J_cJ_w}{J_c + J_w}}$$

We find that r_s^2 = shaft radius squared, may be written K_rD^2.

Also o = the distance between oscillating masses, may be written K_oD since the cylinder centers are proportional to D and the flywheel offset well may be the same.

The modulus of elasticity G is a constant for any one material so the whole expression may be written for any set of assumed conditions,

$$T = \frac{2}{K_rD^2}\sqrt{\frac{2\pi K_oD}{G}}\sqrt{\frac{K_cD^5K_wD^5}{K_cD^5 + K_wD^5}}$$

For the sums, products, roots, etc., of constants we may substitute another constant K_e. Then,

$$T = \frac{K_e}{D^2}\sqrt{D}\sqrt{\frac{D^{10}}{D^5}} = \frac{K_eD^{\frac{1}{2}}D^{2\frac{1}{2}}}{D^2}$$

$$= \frac{K\,D^3}{D^2} = K_eD$$

That is, T is proportional to D or the frequency of vibration of the shaft with its loading varies inversely with either bore or stroke.

To sum up, if one has a series of engines to design with a chosen stroke bore ratio common to all and with a common ratio of cylinder spacing to cylinder diameter, the shaft and pin diameters properly are a fixed percentage of the bore and the critical speeds of any one order will vary inversely as the bore or stroke.

Some questions may arise regarding matters that have been reserved to avoid complicating the treatment. They are anticipated in the following.

The crankshaft length is not considered equal to its dimensional length in these calculations. An equivalent length, usually greater, is substituted allowing for the greater torsional flexibility of cranked shaft. The ratio of actual to equivalent length remains equal for a series of shafts of the same straight line proportions if the usual semi-empirical formulas may be trusted at all. In carrying out the method in practice no grief has resulted.

LOW FREQUENCIES ARE OBTAINED

The method of treating the cranks and reciprocating masses as one equivalent mass results in frequencies slightly too low. If the aim is to avoid all destructive vibration in the range from zero to full speed of the engine, a critical at a speed slightly higher than calculated obviously cannot be harmful.

There are no practical difficulties in the way of straight-line proportions for shafts. It tends to order in the design process and a pleasing appearance of proportionality in the finished series of engines.

This analysis indicates the possibility of testing shaft models and complete torsional system models of more complicated form to determine critical speeds, equivalent lengths and the like prior to the construction of the massive full-size system.

INDEX