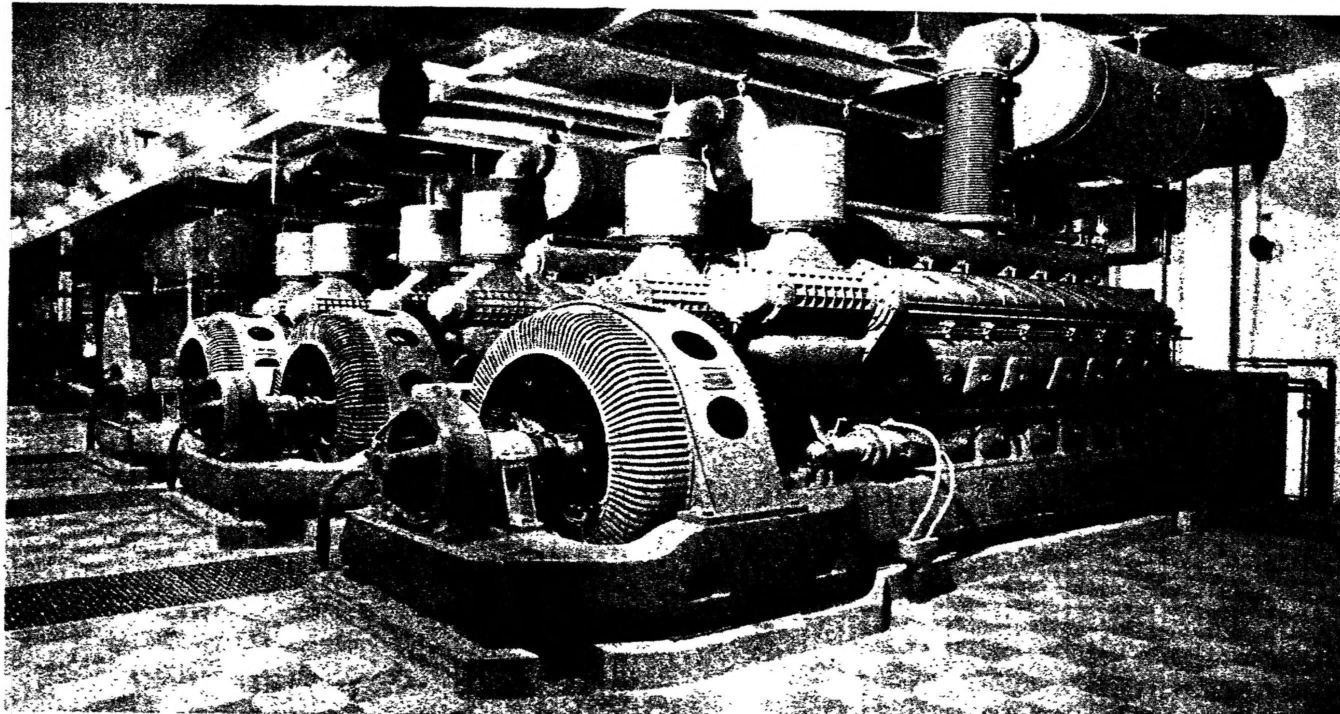
AMERICAN DIESEL ENGINES



(*Frontispiece.*)

Power plant of Alfred I. duPont Building, Miami, Fla., a new office building. Dieselized and fully air conditioned. The plant consists of three General Motors sixteen-cylinder, two-cycle Diesel engines, rated at 1,050 hp. at 600 rpm., and one General Motors eight-cylinder, 225-hp., four-cycle Diesel engine for the Sunday and night load.

This is the first large office building in the United States to depend entirely on Diesel engines for *all* its power and light requirements.

AMERICAN DIESEL ENGINES

BY

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PREFACE TO THE SECOND EDITION

In 1930, the author dared to prepare a volume dealing with the then current American designs of Diesels, their operation and maintenance. The effort met a welcome from Diesel engineers, quite unexpected in its extent.

Since then there has been a tremendous development in Diesel design. This, together with the success of the first edition of *American Diesel Engines*, has encouraged the author to revise the volume completely, bringing it up to date.

It is the author's faith that the reader will find in the present edition much of real value.

L. H. MORRISON.

PORT WASHINGTON, N. Y.,
July, 1939.

PREFACE TO THE FIRST EDITION

Several years ago the author gathered material which was published under the title "Diesel Engines." Engineering is always in a state of progress, so that it became necessary to revise this volume.

Early in the revision the author was convinced that the developments in Diesel design and application had been so broad that a mere revision was not sufficient, and that the subject could only be covered properly through a complete rewriting of that book. To hold the material within reasonable dimensions for a single volume, foreign engines could not be included, save as the designs have been adopted by American builders; consequently, to have the title indicate more clearly the field covered, this present volume is entitled "American Diesel Engines."

The early history of Diesel development, as well as some phases of the economics of oil-engine power plants, has been included, on the assumption that the business man and factory manager are interested in cheap power sources.

The author desires to thank various engine builders for the generosity with which they have supplied information not otherwise available. He also desires to acknowledge his indebtedness to his wife, Emma Snow Morrison, for her assistance in reading and arranging the manuscript.

L. H. MORRISON.

NEW YORK, N. Y.,
December, 1930.

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AMERICAN DIESEL ENGINES

CHAPTER I

THE DIESEL ENGINE

General.—Owing to the widespread publicity given to the Diesel-powered, high-speed, streamlined trains, every engineer knows that the Diesel engine falls under the classification *internal-combustion engines.* He also knows that the basic difference between the Diesel and the gasoline engine is the system of fuel handling and ignition—that in the Diesel, *air only* is compressed, to a temperature of above 1000°F., and that fuel is injected into this hot air and is ignited automatically, without the presence of a spark plug. Many structural differences exist, but those mentioned above are the basic ones.

The Carnot Cycle.—If one were to essay the establishment of the primary principle of heat engineering that Rudolph Diesel attempted to follow in his invention, he would finally arrive at the "perfect heat engine," the proposition advanced by Sadé Carnot, a French scientist, early in the nineteenth century. His suggested cycle of events for the perfect heat engine was given the name "Carnot cycle."

Common knowledge was that in water power the greatest efficiency was obtained by having all the water enter the water wheel at the maximum head of the water source and discharge at the minimum head of the tailrace, without either loss or addition of other water in the process.

Applying this self-evident water-power truth to all engines whose action was obtained from the addition of heat to the working medium, air, steam or other material, the declaration of Carnot, set down in everyday language, was to the effect that the perfect engine was that which took in all its heat at the maximum available temperature; expanded the heated medium without loss to the walls or atmosphere; rejected the heat remaining after expansion, at the lowest possible temperature (usually the temperature of the atmosphere); and, finally, recompressed the medium (or an equal weight of other medium of the same temperature and pressure), without external loss, back to the original high temperature.

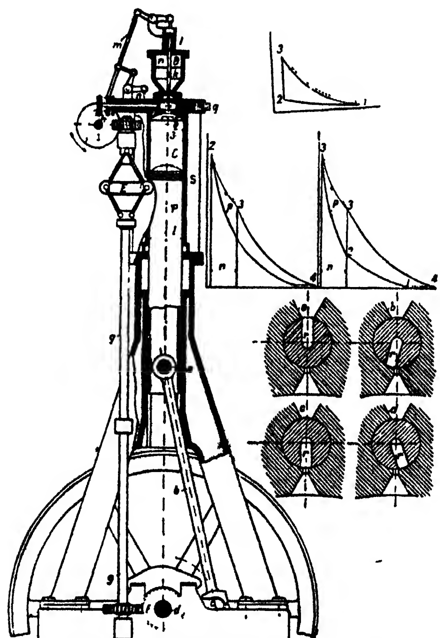


FIG. 1.—Dr. Diesel's first patent, a coal-burning engine.

No other engine operating between the two temperature levels could be more efficient.

Beau de Rochas.—Beau de Rochas provided the mechanical arrangement for all internal-combustion engines of today. This scientist outlined in 1862 the cycle of events that a reasonably efficient engine should have. He realized the significance of Carnot's idea of taking in heat at the maximum possible temperature after compression to this maximum temperature, but was practical enough to appreciate that, if the air-fuel mixture was to be compressed, the final compression temperature and pressure must be low enough to prevent preignition. His cycle, which received the name of "Otto cycle," owing to the fact that Otto & Langen built the first on this design, was as follows:

1. *Suction stroke*—draw in air and fuel.
2. *Compression stroke*—compression of mixture, followed by ignition at dead center.
3. *Power stroke*—expansion of the burned gases.
4. *Exhaust stroke*—piston sweeps gases from the cylinder.

All present-day gas and gasoline engines employ this Beau de Rochas cycle, although the Scotsman, Sir Dugald Clerk, was able to carry out the same cycle in two strokes, thereby inventing the two-cycle engine. The compression pressure has always been

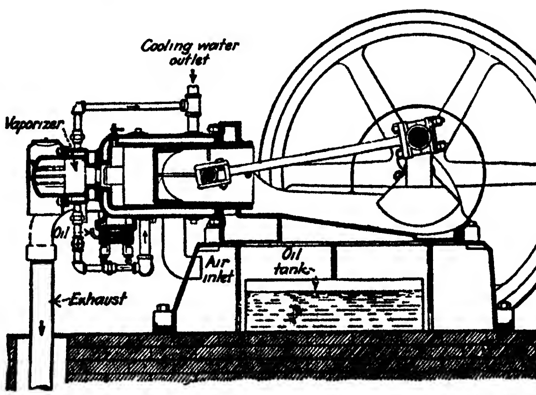


FIG 2—Akroyd Stuart oil engine.

limited to about 60 lb. in the slow-speed gas or gasoline units, for a higher pressure produces preignition.

Akroyd Stuart Oil Engine.—In 1888 Charles Akroyd Stuart patented an engine in which air only was drawn into the cylinder. A chamber was attached to and was connected with the cylinder head, as shown in Fig. 2.

On the suction stroke of the engine a pump sprayed oil into the combustion chamber, which was kept hot by a torch. The hot surface vaporized the oil which was held in the chamber by the restriction of a narrow throat.

On the compression stroke the cylinder charge of air was raised in pressure to about 60 lb., and a temperature rise accompanied this pressure rise.

During this stroke the air was gradually forced into the combustion chamber, until finally compression raised the temperature of air and fuel to the point where the mixture ignited spontaneously.

The burning gases flowed back through the throat, to exert pressure on the piston during the power stroke.

This was a low-pressure engine, to avoid preignition; the efficiency was correspondingly low. It was built in large quantities by Ruston & Hornsby under the name Hornsby-Akroyd and was introduced into America in 1895.

Stuart also took out a patent on an engine in which the oil was injected during the compression stroke and the compression was to be high enough to insure self-ignition. This predated Diesel's patents.

The Diesel Engine.—Having in mind the Carnot cycle, Diesel conceived the idea of compressing the air charge, without loss of heat to the cylinder wall, to a pressure such that the resulting temperature would be the equal to the temperature of combustion. The fuel was then to be introduced after the piston started on the power stroke and at a rate such that the energy released by combustion just equaled the energy used to move the piston; that is, combustion was to be *isothermal*. The temperature in the cylinder would then remain constant at, say, 3500°F., until the introduction of fuel ceased. The hot gases were to continue to move the piston, dropping in pressure and temperature during this expansion, without loss of heat to the cylinder walls, until the gases reached atmospheric pressure and temperature.

At the end of this power stroke the burned gases were to be forced from the cylinder, and a fresh charge of air drawn in.

During the early part of the compression stroke, the heat due to the compression was to be removed, so that the temperature of the charge remained the same as that of the atmosphere, giving *isothermal* compression. At some point in this stroke, the cooling was to be discontinued, and the air compressed without heat loss until it reached, say, 3500°F., corresponding to the temperature of combustion; this was *adiabatic* compression.

This cycle was the Carnot cycle, introducing all the heat at the highest temperature, expansion without loss to the outside (adiabatic), heat removal at constant temperature (during the early part of the compression stroke), and adiabatic compression.

This was the conception of the first engine patented by Diesel. In it he betrayed the fact that he was eminently a theorist rather than a practical engineer. His plan of cooling the air charge during the initial stages of compression by injecting a stream of water was impossible in a commercial engine. His plan to follow this "cooled" compression by compression without loss of heat to the cylinder walls to over 2,500 lb. per square inch, in order to reach a final temperature equal to the temperature of combustion, involved more than metallurgy had to offer. Furthermore, the power that might be developed from the combustion of the fuel charge would have been employed almost entirely in overcoming friction.

Diesel's original plan of burning coal dust was, of course, impractical and gave way to fuel oil injected into the cylinder by a blast of highly compressed air.

Not surprisingly, the first commercial Diesel did not follow the Carnot cycle but was a constant-pressure engine. In other words, the air charge was compressed to a pressure sufficient to raise its temperature to, say, 1,000°F. The cylinder was water-jacketed to keep the metal reasonably cool, and fuel was introduced at a rate such that while it burned behind the moving piston, the pressure rose no higher than the final compression pressure.

It is obvious that Diesel did not invent the compression-ignition oil engine, but he is to be credited with the commercial development of the engine, which, after all, is of equal importance with, if not of greater importance than, the original invention.

Weiss Hot-bulb Engine.—Carl W. Weiss invented a hot-tube-ignitor, two-cycle, crankcase, low-compression oil engine (Fig. 3). From this engine developed a variety of hot-tube and hot-bulb, low-pressure engines; the precombustion engine; and, finally, in the opinion of many, the modern two-cycle, high-pressure, mechanical-injection Diesel.

Rudolf Diesel.—Rudolf Diesel was born in Paris, France, Mar. 18, 1858. His father was a Bavarian leather goods manufacturer of some means, who many years previous to his son's birth had migrated to France, where the family gained some financial security.

The intense bitterness of the Franco-Prussian War forced the family to leave France. A new home and business were established in London, and Rudolf was sent to relatives at Augsburg, Germany, where he attended the Augsburg Technical School.

Later he matriculated at the Munich Technical High School Here he came under the influence of Linde and Schrotter Linde

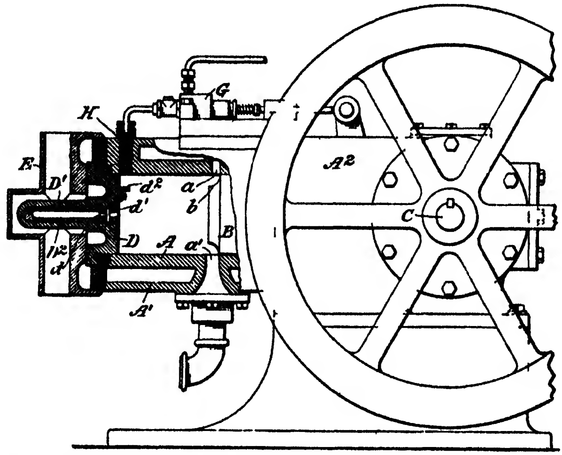


FIG 3 —Carl W Weiss introduced the hot-tube engine in 1893

was developing the Linde refrigerating machine, and for a time after his graduation Diesel was Linde's assistant

FIG 4 —Rudolf Diesel

In 1879 Diesel was an employee of Sulzer Brothers, Winterthur, Switzerland, one of the prominent machine builders of Europe

With the practical experience gained here, he entered the employ of Baron Hirsch, who had acquired Linde's French patents and had formed a company for their development.

In technical school Diesel had specialized in the study of thermodynamics and was much impressed with the Carnot theory of a perfect engine. Continued study of the subject led to the evolution of a theoretical high-efficiency, internal-combustion engine, which he expounded in a startling volume entitled "Theory and Design of a Rational Heat Engine," published in 1893.

Shortly before this, he obtained a German patent (67207), dated Feb. 28, 1892, in which he set forth claims for both simple and compound Diesel engines, the suggested fuel being coal dust. His patent application carried an admission that prior inventions of others included compression ignition His invention dealt with an engine embodying the principle of the Carnot cycle. Later patents included the use of petroleum oil and the employment of high-pressure air to inject the oil charge into the cylinder. His book was much criticized, but Diesel's strong personality prompted Maschienenfabrik Augsburg Nurnberg and Fried. Krupp, two of Germany's largest machinery builders, to cooperate in the development of an actual working engine.

Licenses were sold to firms in various European countries and, as will be mentioned on other pages, in America. Although several were profitable, Diesel embarked on various ventures in which he experienced severe financial losses. In fact, he was often hard-pressed for funds.

Diesel visited America in 1912 on business affairs; in September, 1913, on a voyage from Germany to England he disappeared from the ship. Various theories for his death have been advanced, but all are based on the most meager evidence. His death came at the moment when the engine was starting its marvelous career. In 1938 over 1,600,000 hp. was built in America alone, and there is hardly a country or an industry where Diesel engines are not found.

Carl W. Weiss.—Carl W. Weiss, inventor of the two-cycle, hot-bulb oil engine, was born in Sophienthal, Germany, and was educated in a private institution founded and maintained by Princess Mary, sister of Emperor William of Germany. He left Germany for the United States in 1876 to visit the Centennial

Exposition in Philadelphia. He decided to stay permanently in the United States and continued his engineering education by attending a series of lectures by Professors Plymton, Stone, and Van Der Vals.

Weiss possesses a highly ingenious brain and invented several "caloric," or hot-air, engines before 1890, in which year he developed a cash register, the patents of which were finally purchased by the Dayton Cash Register Company. About this

FIG. 5.—Carl W. Weiss, inventor of the hot-bulb oil engine

time he turned his attention to gas and oil engines and gas turbines.

The first Weiss oil engine, the original two-cycle, hot-tube design, was built in 1893 and completed for production in 1894. Weiss and August Mietz, owner of a foundry on Mott Street in New York City, formed the Mietz & Weiss Engine Company. In this plant the Weiss engine was built by the hundreds until 1915. In this year Mietz died, and disagreement with the executrix about royalty on the Weiss engine patents caused Weiss to leave the company, which soon gave up the building of engines.

Weiss continued his engine development and during the World War built several 400-hp. engines in Lansing, Mich. Later he became interested in the development of transmission devices and formed the Carl W. Weiss Engineering Company. He is still actively engaged in engineering work (1939)

Herbert Akroyd Stuart.—Herbert Akroyd Stuart, born in Yorkshire in 1864, was a pioneer in the development of the oil engine.

He was educated at Newbury Grammar School and at the City and Guilds of London Technical College, Finsbury. He received his early practical training in the engineering works of his father, Charles Stuart, at Fenny Stratford and on the death of his father

Fig. 6—Herbert Akroyd Stuart.

took over the management of the works. He died on Feb. 19, 1937, at his residence in Western Australia, where he had been experimenting with producer-gas engines.

The inventor of the low-compression oil engine began experimental work on the oil engine in the year 1886 and, after four years of labor and heavy expenditure, had two automatic, or compression-ignition, heavy-oil engines built and working in 1890, in both of which were first applied principles now accepted as fundamental in modern oil-engine construction. It is generally agreed that his work did not receive due acknowledgment or recognition in his lifetime. Certainly the Diesel engine of 1939 is as easily traced to Stuart as to Diesel.

The Diesel Principle.—As has been outlined, the chief point that distinguishes the Diesel from other forms of the internal-combustion engine is the compression of a charge of air until the resulting high temperature is sufficient to ignite the fuel without the aid of any external object. In most Diesel engines built prior to 1920 the fuel was forced into the cylinder by a blast of high-pressure air coming from an air compressor. As will be discussed later, most present-day Diesels depend upon the pressure exerted by a pump to force the oil into the cylinder and to atomize it.

At the present time a number of oil engines manufactured both here and in Europe, although using a fairly high compression sufficient to secure ignition of the fuel, do not operate on the Diesel cycle but on what is a combination of combustion at constant volume (the Otto cycle) and at constant pressure (the Diesel cycle). For this reason, they have been called "dual-cycle" engines on account of the constant-volume and constant-pressure combustion.

The features that set the true Diesel apart from other oil engines are (1) compression sufficient to produce autoignition of the fuel; (2) injection of the fuel near the end of compression.

Diesel Cycle of Events.—Figures 7 *A* to *D* show diagrammatically the valve action and the conditions existing in the engine cylinder when the piston is at different points in the engine cycle. The pressure in the cylinder at any point in the piston stroke is indicated in Fig. 7 *F*, whereas Fig. 7 *E* shows that part of the crank circle swept by the crank during any of the several events in the engine cycle.

Figure 7 *A* covers the suction, or admission, stroke of the piston. The admission, or air-suction, valve at *J* opens at the point *a* just before the piston reaches upper dead center. This valve remains open from the point *a* to the point *b*, during which time the piston moves downward, drawing in a cylinder charge of air, reverses, and starts to move upward. This does not mean that the piston pushes part of the air charge back out through the admission valve. In operation the inrush of the air through the air valve continues even after the piston starts upward. The momentum of the air causes it to "pile up" in the cylinder; in this way the amount of air entering the cylinder is actually greater than if the valve had closed at bottom dead center. This admission stroke is shown in Figs. 7 *E* and *F*; in the latter the indicator diagram shows this line as being slightly below the atmospheric pressure line xy. In Fig. 7 *B* the admission valve *J* closes at *b*, and the pure-air charge is compressed by the piston up to the point *g*, which is top dead center. This process is covered by the compression line *bc* on the indicator diagram in Fig. 7 *F*. The clearance volume is very small, and the maximum

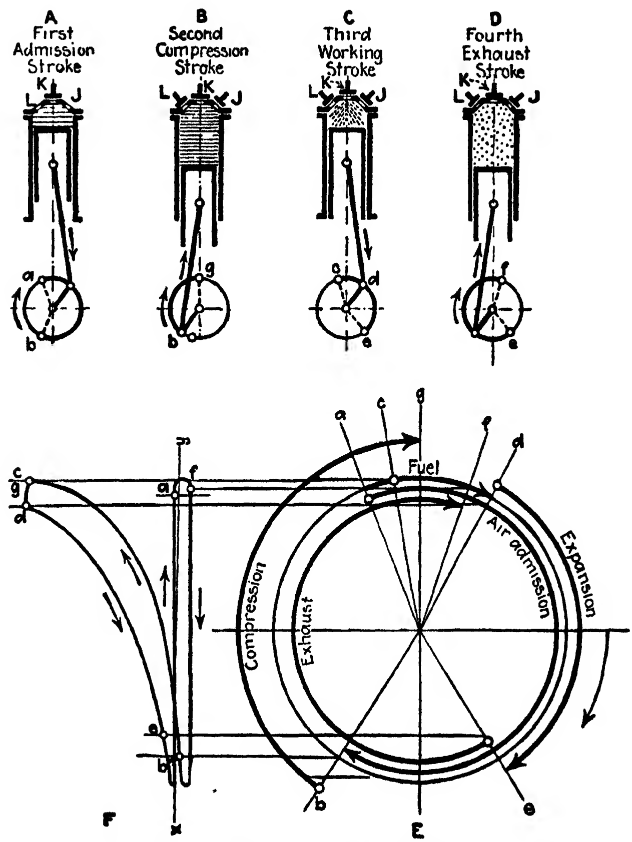


FIG. 7.—Cylinder events of a four-cycle engine.

or final compression pressure rises to some 450 to 550 lb. per square inch. The work done on the air charge in compression causes the temperature to rise to a final temperature of 900 to 1400°F., the air being in a "red-hot" condition.

At the point *c* in Fig. 7 *C* the injection valve *K* opens, and a charge of fuel is blown into the cylinder by means of a blast of high-pressure air. This is the action in an "air-injection" engine; in airless- or mechanical-injection Diesels the pressure exerted on the oil by the injection pump sprays the oil into the engine cylinder. The fuel-injection valve, or "spray valve," as it is often called, is designed to cause the rate of flow through the valve to be regulated so that the entire oil charge is not injected instantaneously but takes place while the engine crank turns through a considerable angle. In Fig. 7 *C* the injection of fuel starts when the crank is at *c* and ends when the crank is at *d*. In Fig. 7 *F* the line *cd* represents the fuel admission and combustion period, and the desired condition is attained when the line *cd* is practically horizontal, showing that the rate of heat addition is such that there is no increase in the cylinder pressure. In all mechanical-injection Diesels there is a rise in pressure, the amount depending upon the engine speed, etc.

The injection and the combustion of the fuel ceasing at the point *d*, the piston continues to the end of its stroke under the influence of the expanding gases. Before the completion of the stroke, the exhaust valve *L* opens when the crank is at *e*. This allows the gases to rush out through the exhaust passage. The exhaust valve remains open until the piston again ascends to the top of the cylinder, expelling all the exhaust gases. This part of the cycle is shown in Figs. 7 *E* and *D* as continuing from *e* to *f*. In Fig. 7 *F* this forms the exhaust *ef*. Before the exhaust valve *L* closes, the admission valve opens at *a*, allowing a fresh air charge to be inducted into the cylinder during the stroke shown in Fig. 7 *A*.

These events complete the cycle of the four-stroke-cycle, air-injection Diesel. From a practical viewpoint the differences between the Diesel and the gas engine are that in the Diesel nothing but pure air is compressed in the cylinder and that the fuel is forced into the cylinder slowly, at about top dead center, causing the combustion to be gradual; in the gas engine both the gaseous fuel and the air are compressed, and the combustion takes more of the form of an explosion.

Summarizing, the Diesel four-stroke cycle consists of: (1) a suction stroke during which a charge of pure air is drawn into the cylinder; (2) a compression stroke wherein this air is compressed by the piston to a maximum pressure; (3) a period of injection and combustion of a charge of fuel and the expansion of the gases, this being the working stroke of the piston; and (4) an exhaust stroke by means of which the burnt gases are expelled from the cylinder.

The Suction Stroke.—The air charge should be drawn into the cylinder at atmospheric pressure (zero gage, or 14.7 lb. absolute). Since the only reason air or any gas flows from one point to another is a difference in pressure, it follows that the pressure in the cylinder must be somewhat below that of the atmosphere if the air charge is to flow in and fill the cylinder. If there is a direct passage into the cylinder, this difference in pressure need be but a few ounces. Where a long air-suction line is used, such as installations with a small pipe leading from outside the building, the drop may easily reach 2 lb. This would give a pressure at the beginning of compression of 12.7 lb. in place of 14.7 lb.

The Compression Stroke.—The purpose of the compression in any internal-combustion engine is to raise the pressure and the temperature of the air charge or, in case of a gas or gasoline engine, of the mixture of gas and air. It has been found that the efficiency depends upon the compression ratio and upon the temperature at the beginning and end of the compression stroke.

The high-compression pressure in the Diesel engine leads to (1) the attainment of a satisfactory efficiency; (2) the securing of a temperature sufficient to produce self-ignition of the fuel and air charge.

Experience has demonstrated that the increase in thermal efficiency by increase of compression pressure above 500 lb. per square inch is too small to overcome the disadvantages from constructional and operating viewpoints. It is customary to carry a maximum compression of 480 to 520 lb., although in certain high-speed and two-stroke-cycle engines pressures up to 600 lb. are not unusual. Since the compression temperature must be above the ignition point of the fuel, there is a minimum pressure possible to employ in the true Diesel.

The temperature of the air charge at the end of the compression stroke must be high enough to ignite the fuel. If no heat were lost to the cylinder walls—that is, if compression were adiabatic—the relation of the pressures and temperatures at the beginning and the end of the compression stroke would be expressed by the equation

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$$

in which

T_1 = temperature at the beginning of compression, degrees absolute (460 + Fahrenheit reading).

T_2 = final compression temperature.

P_2 = final compression pressure, pounds absolute.

P_1 = initial pressure, pounds absolute.

n = 1.408.

This relation is shown in Fig. 8 by the curve marked "adiabatic." There is a loss of heat during compression and expan-

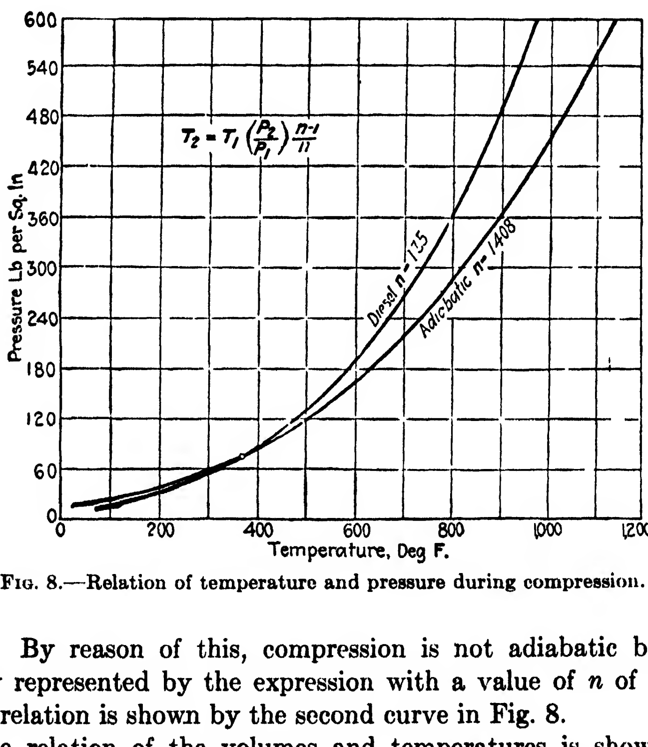


FIG. 8.—Relation of temperature and pressure during compression.

sion. By reason of this, compression is not adiabatic but is fairly represented by the expression with a value of n of 1.35. This relation is shown by the second curve in Fig. 8.

The relation of the volumes and temperatures is shown in Fig. 9, where V_1 is the volume before compression and V_2 is clearance volume, whereas T_1 is the temperature previous to compression and T_2 the temperature after compression.

The Combustion Period.—The fuel valve closes after the piston has traveled about 10 per cent of its stroke, the crank moving about 40 deg. of its cycle. The fuel and air mixture theoretically should be completely burned by the end of the fuel injection, but in the actual engine the combustion continues far down the expansion line.

Although the Diesel is spoken of as a constant-pressure engine —that is, the fuel is burned at such a rate that the pressure within the cylinder does not increase—it is seldom that an indicator diagram from an engine in everyday service shows a horizontal combustion line. A rising pressure is customary and has been found to give a higher efficiency. With the ideal horizontal combustion line, although the pressure is constant,

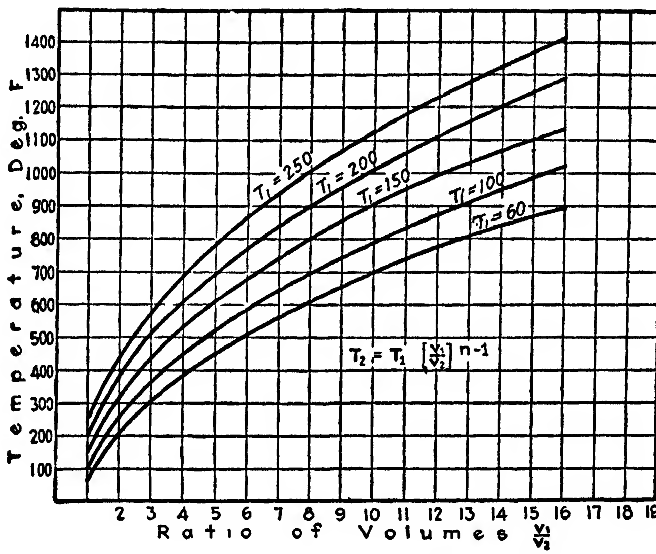


FIG. 9.—Volume-temperature relation of air.

the temperature rises from a compression temperature of, say, 1200 to some 2700 to 3500°F.

At the beginning of this fuel-combustion period, the opening of the fuel valve is followed by a slight time lag before the fuel enters the cylinder and ignites.. This lag is due to the inertia of the fuel resting in the fuel valve, to the resistance of the atomizer disks, and to the slowness of ignition after the oil is in the cylinder. This delay in ignition depends upon the compression pressure and temperature and upon the pressure of the air spray as well as upon the characteristics of the fuel oil.

The Expansion Stroke.—After the fuel valve closes, theoretically combustion should be at an end. The burned gases now expand, forcing the piston outward and so increasing in volume and decreasing in pressure.

The Exhaust Period.—In the actual engine the exhaust valve opens before the end of the stroke, approximately 50 deg. ahead of crank dead center, or about 10 per cent of the stroke. The energy lost by early release is negligible; and if the release is delayed until dead center, the drop in pressure will be carried on during part of the exhaust stroke, making the negative work of the exhaust stroke greater.

The temperature of the gases, when first issuing through the exhaust valve, is around 1400°F. This temperature does not exist beyond the valve, since the pressure drop is accompanied by a temperature drop. The exhaust temperature will range from 500 to 900°F.

The return stroke of the piston forces the gases still in the cylinder out through the exhaust line, the gas temperature falling considerably during this stroke. There is, however, the clearance volume, which is filled with burned gases when the piston reaches the end of its stroke. To remove this volume of inert gas it is customary to allow the exhaust valve to remain open somewhat after the piston completes this stroke, reverses, and starts on the suction stroke. The admission valve is open ahead of dead center, and the momentum of the column of exhaust gases causes the flow to continue even after the reversal of the piston, creating a partial vacuum into which the fresh air rushes. Some gas, however, is left.

Schematic Layout of the Diesel Engine.—Figure 10 embodies the schematic arrangement of the essential mechanism of a Diesel engine. In this particular instance the engine is of the horizontal type operating on the four-stroke-cycle principle. The engine crank is represented with its center at *O*, and the crank revolves clockwise. To the crankshaft is geared a layshaft *A*, which revolves at half engine speed. On this layshaft are mounted the cams used to actuate the exhaust, admission, and fuel-injection valves, which are operated in the sequence outlined in Fig. 7. The fuel pump is driven off the layshaft *A*, the amount of fuel charge being under control of the governor. The fuel is deposited in the fuel valve *C*, out of which it is forced by the air charge at the proper moment. The air blast is supplied by the air compressor *D*, which is driven by a crank on the end of the engine shaft.

In this diagram the air line is not supplied with a receiver or air bottle, and the air is delivered directly from the compressor to the fuel valve.

The parts enumerated are the essential details of the Diesel engine. However the arrangements may differ, it is necessary that the unit include an air compressor, fuel pump, governor, fuel cam, and injection, or fuel, valve, in addition to those parts generally found on an internal-combustion engine.

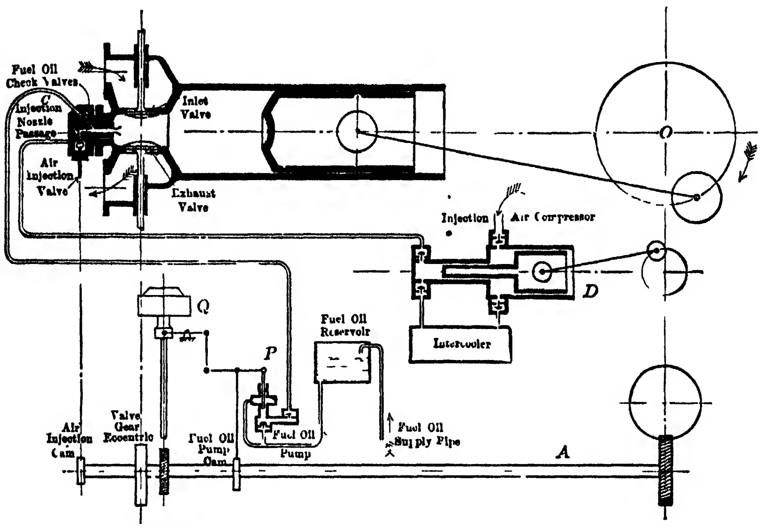


FIG. 10.—Schematic arrangement of the air-injection Diesel mechanism.

Cylinder Events of a Two-stroke-cycle Diesel.—Figure 11 represents the working diagram of a two-stroke cycle. In Fig. 11 *F* the air charge is compressed from *b* to *c*. The fuel is injected when the piston reaches top dead center, and the piston, forced downward on the working stroke, makes the lines *cd* and *de* (Fig. 11 *C*); at *e* (Fig. 11 *D*) the piston uncovers passages or ports in the side of the cylinder through which the exhaust gases pass. As the piston moves downward to the point *a*, scavenging valves in the cylinder head open, and a charge of pure air, which has been compressed in a large air pump or compressor to about 4 lb., blows into the cylinder, clearing it of all exhaust gases. At *b*, or slightly before this point, the scavenging valves close; and as the piston moves upward, it seals the exhaust ports at the point *b*. Continued upward motion compresses the air charge until upper dead center is reached, whereupon the cycle is repeated. In some engines the scavenging valves in the head are replaced by a row of air ports which are uncovered by the piston

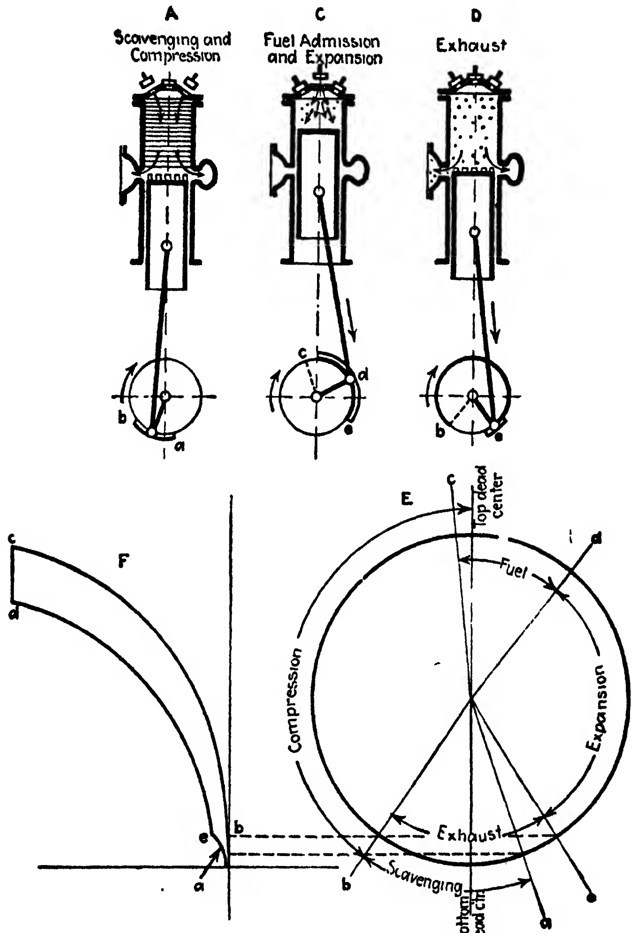


FIG. 11.—Cylinder events of a two-stroke-cycle Diesel.

at the end of the expansion stroke. The air flowing in through these ports clears the cylinder of the exhaust gases. The principle, however, is the same.

Mechanical-injection Diesels.—In most modern designs of oil engines high-pressure air is *not* used to blow the oil into the cylinder; instead, a direct-action oil pump forces the fuel through an orifice in the form of a fine spray. The breaking up of the fuel is sufficient to give excellent combustion.

Such engines may be separated into three general classes: (1) *direct-injection,* those injecting and burning the fuel within the engine cylinder; (2) *turbulence chamber,* those burning the fuel in a combustion cavity separated from the working cylinder and connected with it through a suitable opening; (3) *precombustion,* those designs, mostly of the two-stroke cycle, wherein the fuel is sprayed into a small cavity within the cylinder head, where part

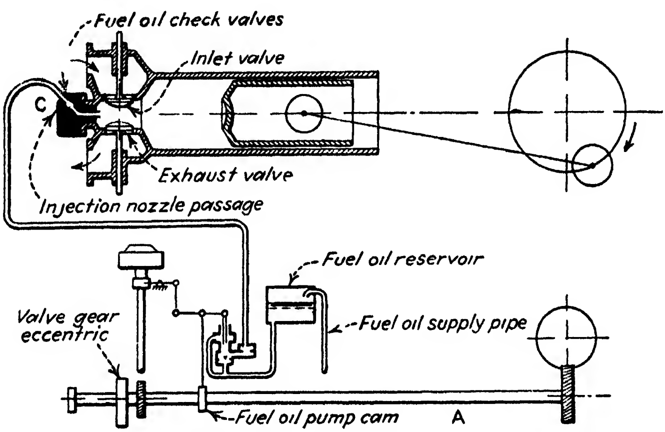


FIG. 12 — Schematic arrangement of a compressorless, or mechanical-injection Diesel

or all is vaporized and then permitted to flow into the working cylinder where combustion is completed; and (4) *energy-chamber,* design, in which part of the fuel is burned in an auxiliary chamber and the blast of these burning gases blows out into the main chamber, to mix the unburned fuel and air. These several departures from the original design will be dealt with in the appropriate chapters. It is well to mention that the mechanical-injection engine predominates in capacities below 1,000 hp. and has been built in sizes up to 11,000 hp. Its popularity is due principally to the elimination of the injection-air compressor, which reduces the manufacturing costs and is believed, by many, to minimize the operating difficulties.

The schematic arrangement in Fig. 12 reveals the simplicity of the fuel-injection system of the mechanical-injection engine compared to the air-injection system. It will be observed that the air compressor, air-injection cam, and air piping, although present in Fig. 10, are absent in Fig. 12.

The reader should understand that Fig. 12 does not illustrate any particular airless, or mechanical-injection, design; it is presented merely to call attention to the simplicity. An actual fuel nozzle of this type of Diesel does not follow the primitive design shown in Fig. 12. In later chapters the various mechanical-injection systems will be explored.

The cycle of events of such an engine is represented quite fairly by the air-injection Diesel diagrams of Figs. 7 and 11, except that in almost all mechanical-injection Diesels combustion of fuel is not at constant pressure. Instead, there is a rise in pressure above the final compression pressure; the total, or peak, combustion pressure will range from 600 lb. in a slow-speed Diesel to 1,200 lb. in a high-speed unit such as the General Motors two-cycle automotive Diesel.

CHAPTER II

DEVELOPMENT OF THE DIESEL

General.—Even though it be almost repetitive of statements on previous pages, occasion is taken here to point out that high-compression, self-ignition engines, now being built in the United States, to which the generic name of "Diesel engine" is applied, are the outcome of three parallel developments. One group of American Diesels are derived from purchases of the patent rights and licenses of Diesel himself, the earliest American rights having been obtained in 1896; another started with the Weiss low-compression, hot-bulb engine built in the United States in 1893 and gradually developed here to present designs through successive increase in compression and improvements in combustion systems. A third group may trace its ancestry back to the Hornsby-Akroyd low-pressure, four-cycle oil engine.

The engine developed by Diesel in collaboration with Krupp and M.A.N. of Germany, whose financial resources and shop facilities were able to evolve a workable engine from a most tenuous design, was first placed into actual operation in America. True, both Krupp and M.A.N. displayed Diesels at the Munich 1898 Exposition, but these were not intended for commercial usage.

Diesel Reaches America.—It was left to Adolphus Busch, head of the Anheuser-Busch Brewery Company, St. Louis, Mo., to give commercial significance to this new engine. Busch, at the suggestion of Baron von Krupp, purchased for $250,000 the American rights to the Diesel patents of 1895, which embodied air injection of fuel and constant-pressure combustion. These patents were assumed to run until 1912 but actually, as the result of some confusion in the application, expired in 1906. This, however, was not discovered until later and was held a secret among the company officials. The humor lies in the patience with which many manufacturers awaited the arrival of 1912 before engaging in Diesel building, when any good patent attorney would have discovered the facts without great effort.

The first American Diesel, a 60-hp. four-cycle unit (Fig. 13), was practically a copy of the first Krupp engine; and following European practice, which was to some extent based on marine steam-engine design, this Busch engine had an open A-frame.

Since the designer had had no great amount of field experience by which operating difficulties could be ironed out, it was not

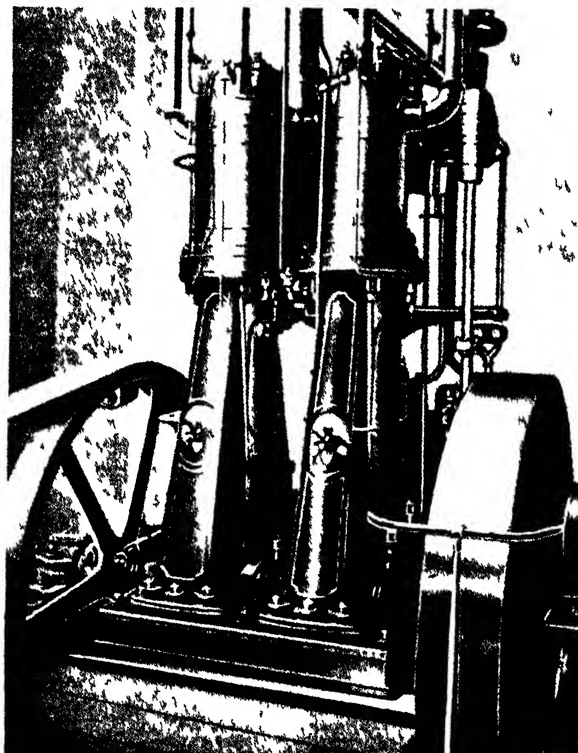


FIG. 13.—The first Diesel to operate in America.

surprising that it did not prove entirely satisfactory. A. J. Frith, chief engineer of the Diesel Motor Company of America, was a theorist with an academic approach to engine design, so J. D. McPherson, an old marine engineer, was employed to carry on the practical design and finally developed a new engine frame, valve gear, and spray valve, shown in Fig. 14.

First American Design.—The new frame has the distinction of being the first gastight-box, or enclosed, frame employed on any Diesel. The valve gear was unique, in that the cylinder head was of the type later called in automotive circles "F-head," with the intake valve immediately above the exhaust valve. The spray valve was placed in a horizontal position, as the illustration shows.

Mention must be made of the splash-oiling system. The crankcase was filled with water up to a level at which the big

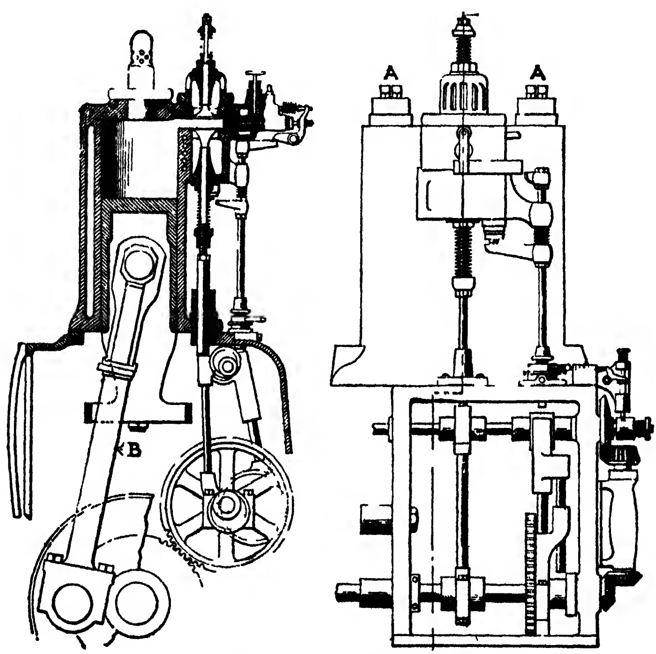


FIG. 14.—First American design of Diesel

ends of the connecting rods struck the surface; to this water was then added about 2 in. of lubricating oil. Lubrication of all bearings (crankshaft, crankpin, and wristpin) was obtained in this way. The water, which quickly formed an emulsion with the oil, acted as a cooling agent and by its evaporation kept the bearings and other parts at the temperature set up by the water evaporation under the pressure within the crankcase.

The first box-frame design had two single-acting, single-stage compressors, $1\frac{1}{2}$-in. bore and 20-in. stroke, cast with the single engine cylinder. The compressor pistons were connected to lugs on the lower end of the engine piston, as shown in Fig. 14. This piston had a diameter of 11 in. and a 20-in. stroke. Running at 200 r.p.m., this single-cylinder Diesel was rated at 20 hp., which works out to a brake mean effective pressure of 41 lb.

Two engines after this design were built at the old Hewes and Phillips Iron Works, Newark, N. J., and installed in the plant of the Long Arm System Company, Cleveland. They gave a world of trouble. The single-stage compressor, delivering injection air at 1,000-lb. pressure, raised the air temperature so high that lubricating oil exploded in the compressor cylinder, and the resulting pressure on the compressor connecting rod broke the lugs on the working piston. The fuel-spray valve had no atomizer disks; the air and fuel merely passed through a series of ⅛-in. holes; thus the fuel consumption was high. As a matter of fact, this general design was employed on many of the later engines.

Norman McCarty (died 1938), sales manager of the Diesel Motor Company of America, the firm created by Busch, suggested that the single-stage compressors be abandoned and a three-stage independently driven compressor used. The change made a surprising improvement.

Shortly after the change was made, Joseph H. Hoadley of the American and British Manufacturing Company, Providence, R. I., suggested to Busch a new plan. The engine was to be redesigned, retaining the box frame and the valve arrangement but employing cams in place of eccentrics for the valve gear, and a suction-valve-control fuel pump. This new engine was the Type A which, with minor improvements, was built until 1912. A cross section of this engine appears in Fig. 15.

Without going too extensively into details, this engine was sold in large numbers by the American Diesel Engine Company, formed by Busch. The first of these Diesels—two three-cylinder, 10 by 15-in. units—were installed at the Jewett City, Conn., municipal light plant.

After the exhibition of three 16 by 24-in., 3-cylinder machines at the St. Louis, Mo., World's Fair in 1904, orders were numerous. One of these 1904 Diesels is still in operation in the plant of News-Register, Wheeling, W. Va. In New England many were installed in cordage plants, textile mills, light plants, etc. The largest installation, at a phosphate mine in Florida, consisted of 16 sets, each made up of two 225-hp. engines.

To relieve the overtaxed plant at Providence, the Power and Mining Machinery Company, later a part of Worthington Pump & Machinery Corporation, contracted to build the overflow.

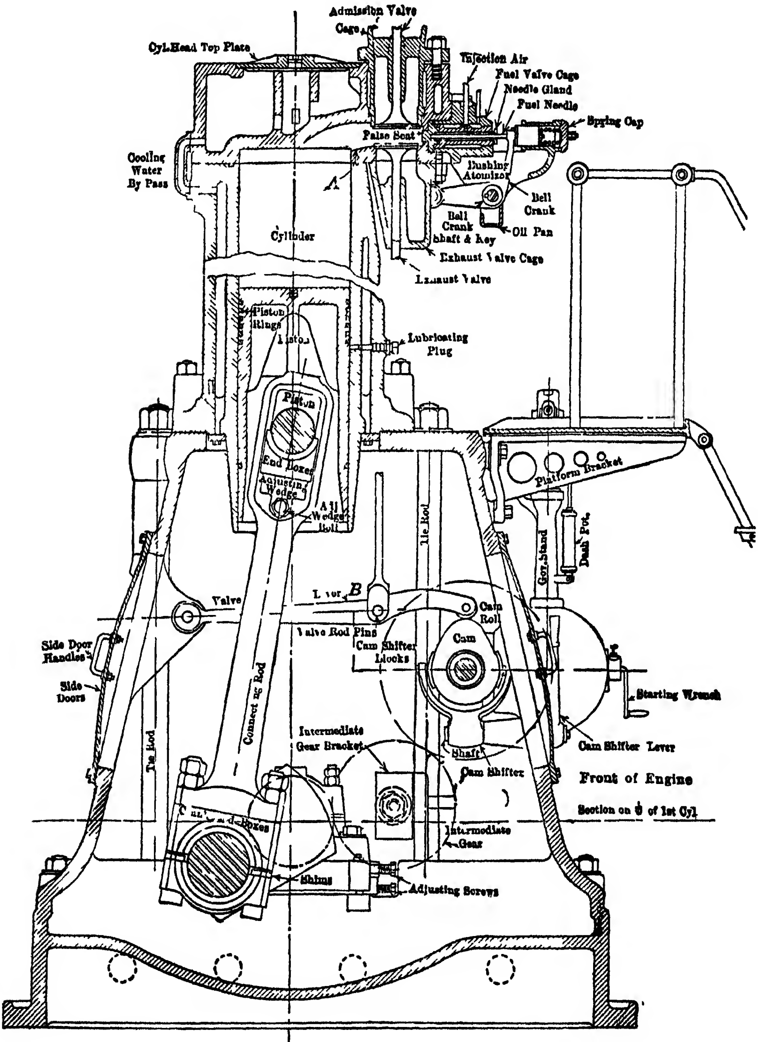


FIG. 15.—Cross section of the Type A Diesel.

Surprisingly, many of these Type A engines, in spite of the numerous faults of design, are still in service in all parts of the world.

The necessity for a better design and the danger from coming competition prompted Busch to open negotiations with Sulzer Brothers of Winterthur, Switzerland. The outcome was the formation of the Busch-Sulzer Brothers Diesel Engine Company and the adoption of the Sulzer two- and four-cycle designs, which formed the basis of the company's product until the Hesselman mechanical-injection license was secured in the 1930's.

Early Builders.—In spite of the apparent monopoly held by the Busch interests, a second American firm, the New London Ship and Engine Company (now the Electric Boat Company), started to build air-injection Diesels in 1908. This builder, employing a design developed by Vickers of England, constructed four engines for the U. S. Submarines E1 and E2. These vertical Diesels operated on the four-stroke cycle and, with four cylinders of 13¾-in. bore and 13½-in. stroke, developed 290 b.hp. at 400 r.p.m. They represented America's first venture in medium rotative speeds.

Paralleling this development of the air-injection, high-compression engine based on Dr. Diesel's license came the work of the oldest American builder of oil engines, the De La Vergne Machine Company, at present a subsidiary of Baldwin Locomotive Works. Although the Hornsby-Akroyd oil engine, which this firm started building in 1893 under English license, was an engine of the low-compression (50 lb. per square inch), surface-ignition type, the De La Vergne company evolved from it through a number of gradual steps a modern type of high-compression, solid-injection, cold-starting engine now known as a "Diesel" engine, yet in which the work of Diesel himself had no part.

Hornsby-Akroyd Engine.—The Hornsby-Akroyd engine (Fig. 16) antedates the Diesel as a self-ignition engine. The patent (1888) upon which the engine was based embodied the use of a combustion chamber connecting to the working cylinder by a throat. Fuel was injected during the suction stroke; and since the compression pressure was low, the combustion chamber was heated initially by a torch in order to supply heat for vaporization of the fuel. On the compression stroke the air was forced into the combustion chamber when the air-fuel mixture was ignited by the hot surface.

Despite its poor fuel economy and its inability to consume heavy crude oils, it was the first oil engine used for oil-pipe-line pumping. Arthur H. Goldingham, with the De La Vergne Machine Company at that time but now living in England, was responsible for the first installation, which was made for the Standard Oil Company of New Jersey, at Fawn Grove, Pa., in 1902. By 1908 some 50 of these engines in cylinder sizes up to 27 by 33 in. had been put in service on various oil pipe lines. The present widespread use of Diesel engines in pipe-line service is a result of this beginning.

Inexplicably, the patent obtained by Stuart in 1892 which contemplated fuel injection at the end of compression, the latter

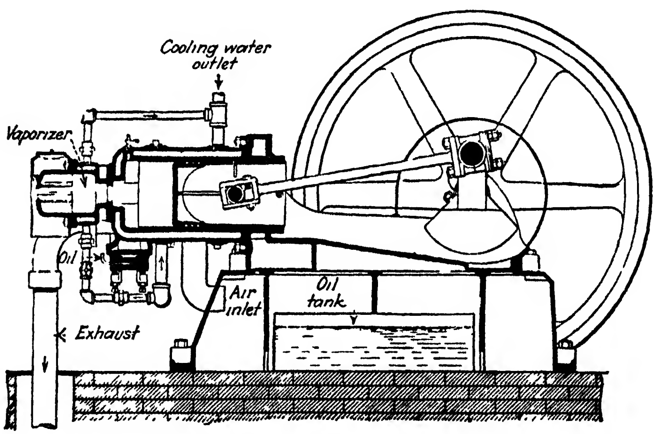


FIG. 16.—Cross section of Hornsby-Akroyd oil engine.

being high enough to insure autoignition, was never applied by either the English or the American builder.

Realizing the desirability of an engine with better fuel economy and particularly one that could use the heavier grades of crude oil, the De La Vergne Machine Company brought out in 1908 a design of the hot-bulb type but with higher compression (280 lb.) and air injection. Alexandro Franchetti, now residing in Italy, was the designer, and the engine was known as the Type FH. The combination of air atomization and hot bulb facilitated the clean burning of heavy, asphaltic fuels and permitted the use of a lower compression than with the usual air-injection engine. Maximum pressures were about 450 lb. This type of engine was built in large numbers for many years, but by 1917 the advent of the Price solid-injection system (to be discussed later), coupled with competitive improvements in high-compression air-injection engines, caused it to be superseded.

Side by side with the development work of the Diesel and Akroyd-Stuart license, the third progenitor of the present-day Diesel appeared; this was the two-cycle, hot-bulb, low-pressure oil engine.

Weiss Hot-bulb Engine.—The first of this basic type was the Mietz & Weiss, the invention of Carl W. Weiss, under a patent application dated Oct. 7, 1892.

The engine incorporated certain features of the two-cycle gas engine, with the crankcase precompressing the air charge. The distinguishing things about this, the first of the hot-bulb oil engines, were the ignition method and the fuel injection into the cylinder. The ignition was, in a measure, an adaptation of the hot tube, long used in gas engines; this was a long tube resting in an extension of the cylinder head.

Fuel was to be injected at the end of the power stroke, vaporized when it struck a ledge, and mixed with the air from the crankcase entering through a scavenging port. Ignition occurred at the end of the compression stroke when the advancing piston forced part of the mixture into the tube.

The first of these units was completed in 1894, and larger engines were built in which the ignitor tube was replaced by a hot bulb. Weiss antedated the precombustion designs which flourished after 1915. In 1904 Mietz & Weiss sold its first vertical engine, a three-cylinder, 150-hp. unit, to Russia.

A most novel device was featured in the Weiss engine built after 1907 (Fig. 17). It had been necessary to inject a small quantity of water into the cylinder in order to control the temperature and thereby prevent preignition durirg compression. Weiss patented a device whereby the cylinder jacket was not completely filled with water, a float in a side chamber controlling the level. Steam formed in the jacket, above the water, passed into a dome and was then admitted into the air-transfer passage between the crankcase and cylinder where it mixed with the air entering the cylinder.

As pointed out in the patent, the use of water injection impaired cylinder lubrication, trouble that did not occur with dry steam. In spite of this statement, water-injection semi-Diesels were built as late as 1930, if not today, and builders still wonder why pistons and cylinder wore rapidly.

Many firms embarked on the manufacture of variations of the hot-bulb oil engines, using such ignition devices as a hot tube, hot bolt, and hot cup.

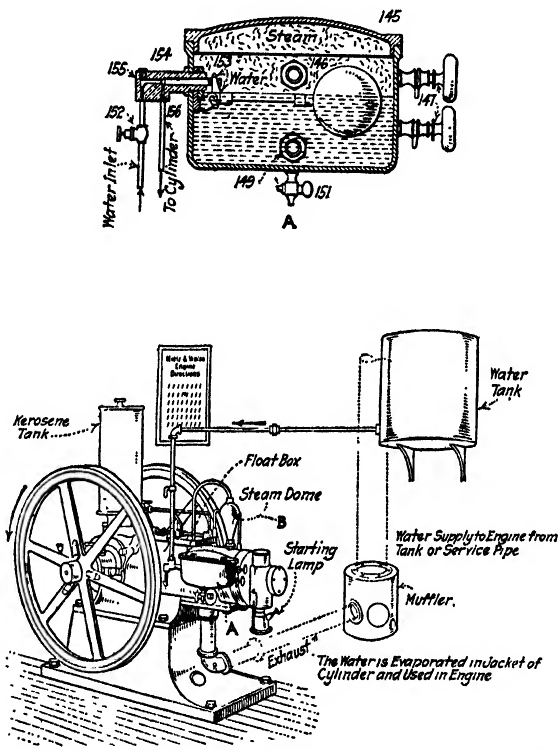


FIG. 17.—The Mietz and Weiss two-cycle hot-bulb oil engine.

Venn-Severin.—Among these early builders was Venn-Severin Machine Company. This firm's first attempt, in 1906, was a converted Comet two-cycle gasoline engine, to which Venn and Severin added a hot ball, a fuel pump, and a water jet. Shortly afterward the firm designed a two-cycle, crankcase-compression, water-injection, hot-bulb engine. This design was improved upon from year to year until, by 1922, a medium-pressure engine without water injection, which will be discussed later, was produced. Still other early builders of low-pressure engines, which later were to be given the names "surface ignition" and "semi-Diesel," were Power Manufacturing Company and Muncie Oil Engine Company (now Ball-Muncie Engine Company).

The Power engine had a horizontal frame, with the crank end of the cylinder serving as the air compressor. The ignition device was a bolt projecting into the cylinder clearance space. To start the engine the bolt, which was held into its recess by a locking plate, was removed and heated by a torch. The chief difficulty

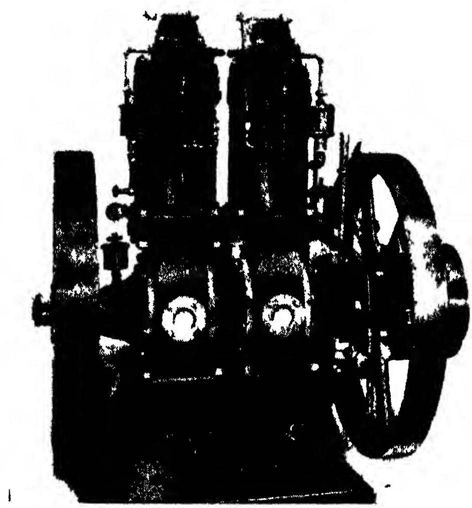


Fig. 18.—Early Venn-Severin two-cycle engine.

was to replace the bolt quickly enough to have it still hot when the engine was turned over. Later designs included an improved ignition and a governor-controlled injection-water pump in 1920 and, finally, medium, autoignition compression in 1930.

Air-injection Diesels of 1910–1920.—As might be expected, since the Diesel engines of 1910–1920 were based on the M.A.N. and Krupp early development, the various American manufacturers who started Diesel construction in the 1910–1920 era employed air injection. The mechanical-injection engines, such as the Akroyd and the Weiss types, were still merely oil engines and had not attained the self-ignition stage.

Busch-Sulzer Brothers Diesel engine company continued to build engines but now embodying the Sulzer Brothers design features.

Four-cycle engines up to 500 hp. per unit found an expanding market. Toward the end of this period the company began to market two-cycle engines ranging upward from a four-cylinder, 750-hp. unit. The two-cycle and the larger four-cycle Diesels had the pistons provided with water cooling. Lubrication of bearings was from a pressure system, the first application of this system.

McCarty's Diesels.—Norman McCarty, who as sales manager of the early Busch company had been of influence when design changes were made, was convinced as 1910 approached that the market demanded Diesels larger than the 225-hp. Type A. Knowing that the Diesel patents had actually expired in 1906, he entered into an arrangement with Atlas Engine Works, Indianapolis, Ind., for it to build a large unit after his designs. In spite of the fact that this firm, for years a builder of slide-valve steam engines, was in the hands of a creditors' committee, construction of the engine was started.

The cylinder bore was 21 in., and with a stroke of 30 in. and at 180 r.p.m., the engine was rated at 175 hp. per cylinder. A-frames, which included the cylinder jackets, were used. Separate cylinder liners were features of the design; this application was the first time that removable liners appeared in America. Pistons were water-cooled through swinging tubes. Naturally, air injection of fuel was followed, with air coming from an independently driven compressor. The first of these engines was installed at the Casper, Wyo., light plant, and another went to the Hawaiian Islands. The latter, a six-cylinder unit, was rated at 1,000 hp. and represented at that time the largest Diesel built in America.

The manufacturer finally went through the throes of receivership, out of which emerged a new corporation, the Lyons-Atlas Company, whose life, however, covered only a brief span. The company also built an experimental double-acting, four-cycle Diesel (Fig. 19).

With the year 1911 a tremendous change in the industry took place. Diesel's patents, which actually were ended in 1906, were no longer a deterrent to those who were attracted by the possibilities inherent in the manufacture of oil engines. Many either obtained licenses from European firms that had patents on engine details or boldly developed their own designs.

Worthington Designs.—Among the first of these was the Snow horizontal, four-cycle, air-injection Diesel developed by Worthington Pump & Machinery Corporation.

This design (Fig. 20) was made available in one-, two-, and three-cylinder units. The injection-air compressor was driven

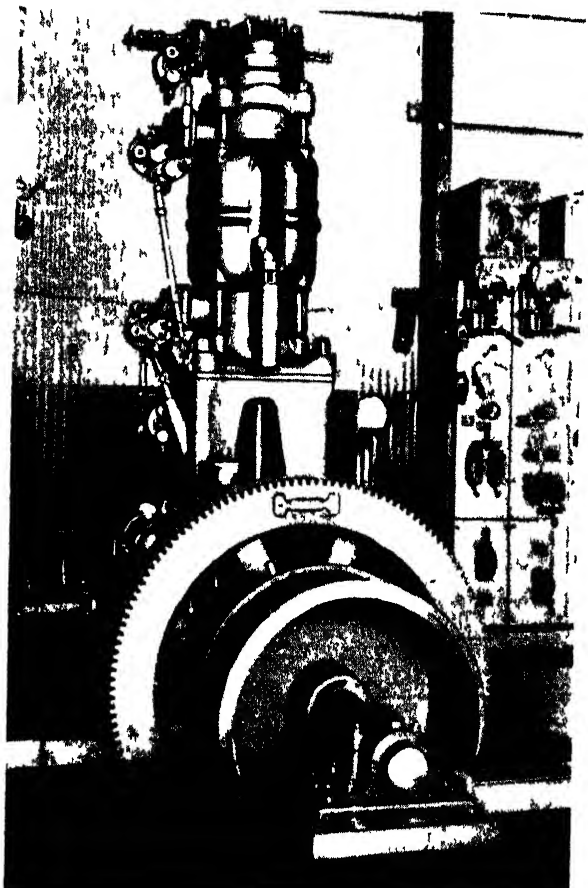


FIG 19.—The McCarty double-acting Diesel of 1915.

from the crankshaft by a drag crank. Valves were placed horizontally in the cylinder head, with the spray valve between, and were operated by a short camshaft, driven, in turn, by a layshaft. The latter also drove the governor and fuel pump. The engine (Fig. 20) had a bore of 19½ in. and a stroke of 33 in.; at 165 r.p.m. it developed 150 b.hp. Many of these engines were installed on oil-pipe lines.

Worthington also built a few two-cycle, horizontal, air-injection Diesels having the same general lines as the four-cycle. This design was hardly a commercial success.

Blanchard's Design.—A design which, although it had but a short life, possessed a number of unusual features, was developed on the drafting board of C. H. Blanchard. Built first by the Standard Fuel Oil Engine Company, it was in succession built by Hadfield-Penford Steel Company, American Clay Making Machinery Company, and, for a time, Chuse Engine

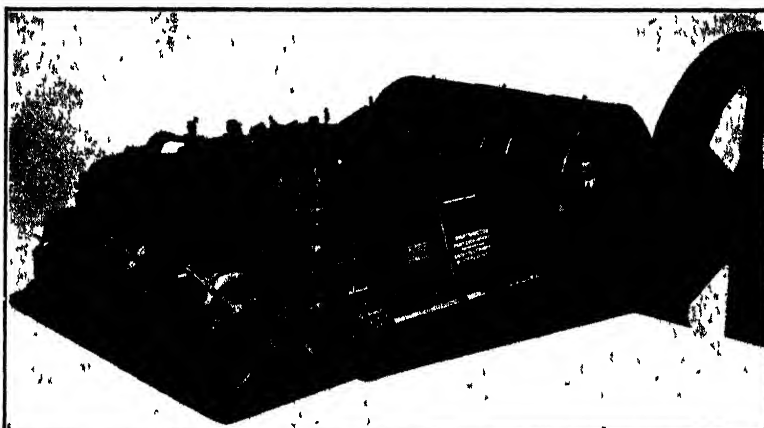


FIG. 20.—The Snow horizontal air-injection four-cycle Diesel of 1912.

Company. This novel two-cycle, air-injection engine employed a stepped piston, which afforded a space for air compression. A Rites inertia governor controlled both the fuel and the injection air supplied to the spray valve.

Nordberg Diesel.— The first American company to build large two-cycle, air-injection Diesels was Nordberg Manufacturing Company. Phelps Dodge Corporation owned mining interests in Arizona and Old Mexico where power was needed in large quantities. Absence of water for condensing purposes and expensive fuel made steam plants impossible. As a solution the mining company in 1912 imported a 1,250-hp. Carels two-cycle Diesel from Belgium. Nordberg obtained the American rights and engaged in building large Diesels only. The feature of this engine was the employment of four valves in the cylinder head to admit scavenging air. The air was supplied by a reciprocating blower driven from the crankshaft. Exhaust took place through a belt of ports. A unit is shown in Fig. 21.

Many of these 1,250-hp. engines were sold, and for years, up to 1920, no other Diesel builder ventured into the "over 1,000 hp." classification. The Nordberg Diesels in the Phelps-Dodge plants demonstrated conclusively that when the fuel oil was properly

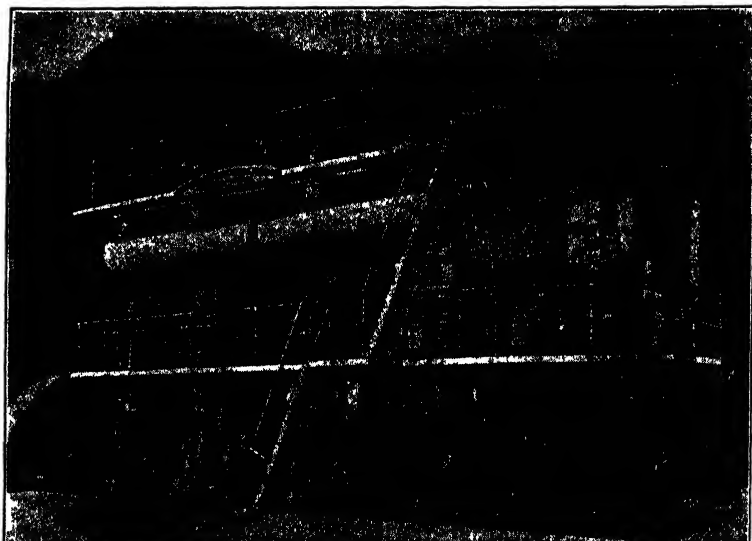


Fig. 21.—Nordberg type 2,000-hp. engine.

handled and the engine correctly designed, an oil of the lowest and cheapest grade could be burned successfully. Later Nordberg Diesels embodied departures from the original Carels A-frame design.

Electric Boat Company.—In 1914, New London Ship & Engine Company built a 2,500-hp., six-cylinder, two-cycle Diesel of 25.157-in. bore and 39.37-in. stroke, for the U. S. Navy, for installation in the *Maumee*. This represented the largest effort in commercial Diesel construction either here or abroad. The company also built in 1912, a 300-hp., 300-r.p.m., two-cycle engine of 9 7/16-in. bore and 10 1/4-in. stroke after M.A.N. design. The first of these units was installed in Standard Oil Company's Barge 62. It also built a line of M.A.N. Diesels, Model 120V4FS, starting in 1913. This was a four-cylinder, four-cycle unit of 9-in. bore and 12½-in. stroke, developing 120 hp. at 350 r.p.m. (Fig. 21 *A*.) This company is now the Electric Boat Company.

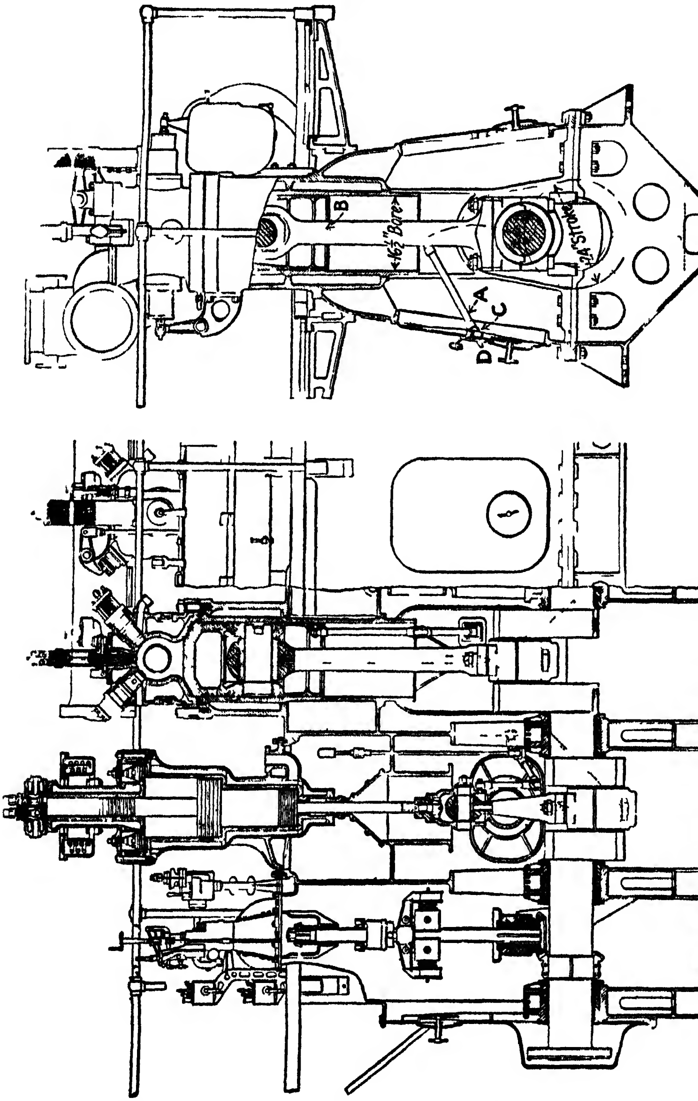


Fig. 21 *A*.—Cross section of New London 600-hp., four-cycle, air-injection engine

Allis-Chalmers Diesel.—Allis-Chalmers Manufacturing Company, with considerable experience in the building of large gas engines, embarked on a Diesel development in 1912. The engine was a horizontal, four-cycle unit, with cylinder numbers ranging from six to one (Fig 22) As many as three cylinders were embodied in one frame, whereas a greater number entailed the employment of two units with the flywheel and generator swung between

The design included several unusual features One of these was the use of eccentrics in place of cams for valve actuation The required quick valve opening was obtained by rocking-type levers, or rocker arms The fuel valve was of the "open" type,

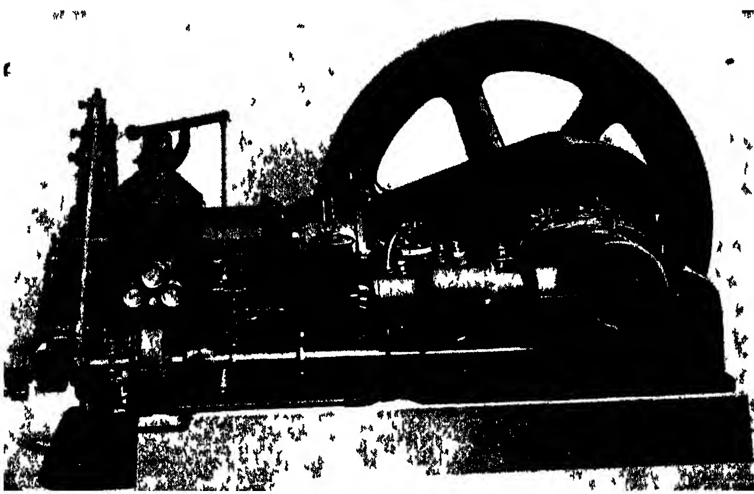


FIG 22 — Allis-Chalmers horizontal-frame Diesel

in that the oil was deposited in the tip of the spray valve which was in open communication with the engine cylinder The needle valve thus controlled only the injection air flow. The schematic outline of a Diesel's mechanism in Fig 10 is that of the Allis-Chalmers design.

With the intake and exhaust valves placed vertically, the cylinder head was provided with a recess forming the combustion chamber and space for the valves, following what was known as the "Nurnberg" designs A feature worthy of mention was the cast-iron (or, rather, semisteel), crankshaft Many early Diesels suffered broken shafts, but Allis-Chalmers engineers realized that to avoid torsional difficulties, shafts must be designed for stiffness, which inherently gave a shaft amply strong Consequently the cast shaft was eminently correct. This was in many respects a forerunner of the present tendency to use cast-iron shafts for high-speed engines.

A considerable number of Allis-Chalmers Diesels were installed in all parts of the country. The most important of these were in the Yarhola pipe-line pumping stations.

The company discontinued their manufacture after a few years but now builds a spark-ignition, pump-injection, low-pressure tractor oil engine.

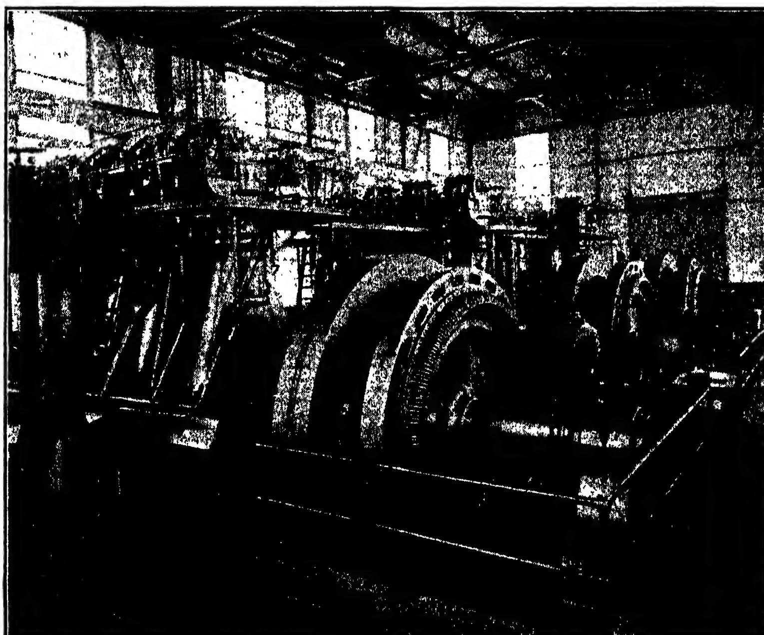


FIG. 23.—500-hp. McIntosh & Seymour (now American Locomotive Co.) Diesels, Texas Light and Power Company, Paris, Tex.

McIntosh & Seymour.—A builder who shortly after 1910 embarked on Diesel construction with a European design was McIntosh & Seymour, now a division of the American Locomotive Company. This firm had built steam engines since 1886 but decided in 1910 that a more efficient prime mover was required; after study, the Diesel seemed to offer advantages.

It was thought best, in order to obtain a better development and quicker commercial results, to start with a Diesel engine that had already been well developed in Europe, and this company joined with the Aktiebolaget Atlas Diesel of Sweden for this work.

For the first to be built, there was selected a 500-hp., four-cylinder, four-cycle unit which had given good results abroad, so the first Diesel engines turned out by this company exactly reproduced this successful European model. The first three Diesels so built were put in service for the Texas Power and Light Company, in Paris, Tex., in 1915 (Fig. 23). Incidentally, after carrying the load of the plant for 15 years the engines were moved to Odessa, Tex., where they are in service today.

For use in this country, a slightly different type appeared desirable, one with a box frame and employing pressure lubrication in place of ring oiling. This was to take the place of the European A-frame engine with which McIntosh & Seymour had started.

However, these first A-frame engines proved so satisfactory that about seventy of them were built before McIntosh & Seymour stopped turning out this type.

Opportunities in the Diesel field attracted other firms which had been building steam engines.

Fulton Diesel.—Fulton Iron Works, for example, had a large factory devoted to the manufacture of sugar-mill machinery, including the Corliss steam engine.

The company obtained a license from the Italian Tosi Company and built its first Diesel in 1914. This design (Fig. 24) included individual A-frames, a feature that was retained until 1926. The fuel pump was a Fulton development, after the design of the late George Pogue. The pump plungers had a variable stroke; a latch engaged the plunger, to pull it on its suction stroke, and at the proper time a governor-controlled cam disengaged the latch, thereby determining the length of the suction stroke and the amount of oil drawn into the pump casing. As in the case of most of the Diesels of 1910–1920 vintage, the bearings of the Fulton were oiled by sight-feed gravity lubrication. The piston pins, however, were supplied oil through tubes that met the ends of fixed tubes filled with oil under a slight pressure. The contact opened a check valve, and oil entered the feed line to the pin.

Atlas-Imperial.—In 1914 Atlas-Imperial Gas Engine Company (now the Atlas Imperial Diesel Engine Company) built an air-injection Diesel. This was a six-cylinder, four-cycle unit of 11½-in. bore and 14-in. stroke. Originally sold to an Arizona mine, the World War stopped delivery, and it was later installed in the ferryboat *Vashon Island.*

This Atlas-Imperial engine had a box frame, with long push rods operated from a camshaft in the frame and connected to the valve rocker arms. Two three-stage direct-connected air compressors supplied injection air.

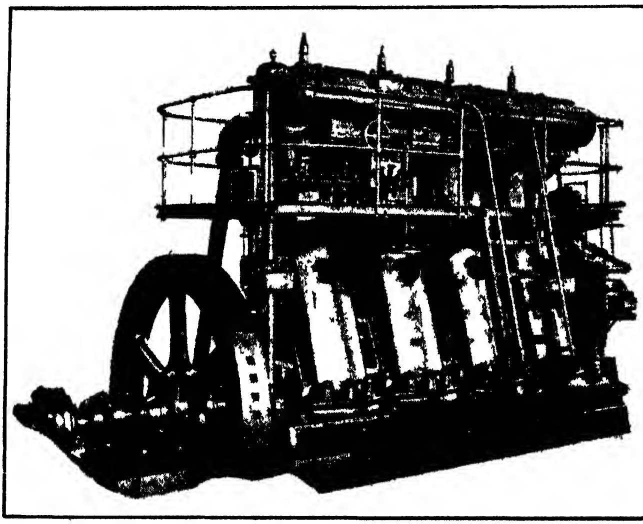


Fig. 24.—Fulton-Tosi Diesel.

Hvid Engines.—Midwest Engine Company, originally Lyons-Atlas Company, built a few vertical engines after McCarty's design, using a Hvid ignition cup. These, although not air-injection engines, injected the oil spray into the working cylinder by means of ignition of a primary fuel charge in a receptacle in the cylinder head (Fig. 25). Other firms essayed to use the Hvid patent because of its apparent simplicity. Experience proved definitely, however, that the system was not suitable for multi-cylinder engines.

Besides Midwest, firms building Hvid engines included St. Mary's Oil Engine Company and Burnoil Engine Company (later taken over by Dodge Manufacturing Company).

Winton.—Because of the general use of Winton Diesels in marine and railroad circles and the absorption of the company by General Motors, the early engines built by Winton Engine Company are interesting.

The first engines built by Alexander Winton were carburetor-type distillate units for the Russian government. When the United States entered the World War, Winton began to build six- and eight-cylinder air-injection engines, mostly for marine

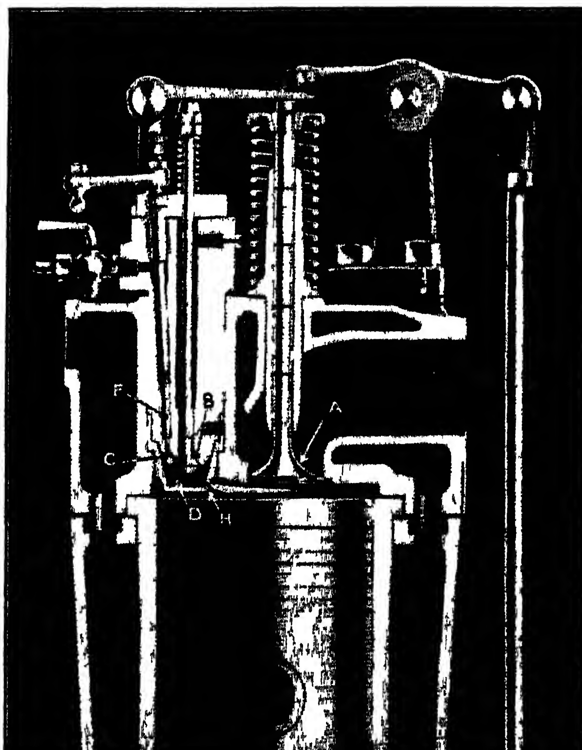


FIG. 25.—Lyons-Atlas Hvid engine.

application. The first units (Fig. 26) had a bore of 13 in. and an 18-in. stroke and weighed 195 lb. per horsepower. The fuel pump was controlled by the governor through throttling the suction; this later was abandoned. The design, although improved from year to year, retained its general lines until Winton adopted solid injection.

Mechanical-injection Diesels of 1910–1920.—The high-pressure, mechanical-injection or, as it is frequently termed, "airless-injection," Diesel may be regarded as coming into existence early in the 1910-1920 decade.

Price Design.—In 1914 W. T. Price, then chief engineer of the De La Vergne Company, conceived the idea of an engine having

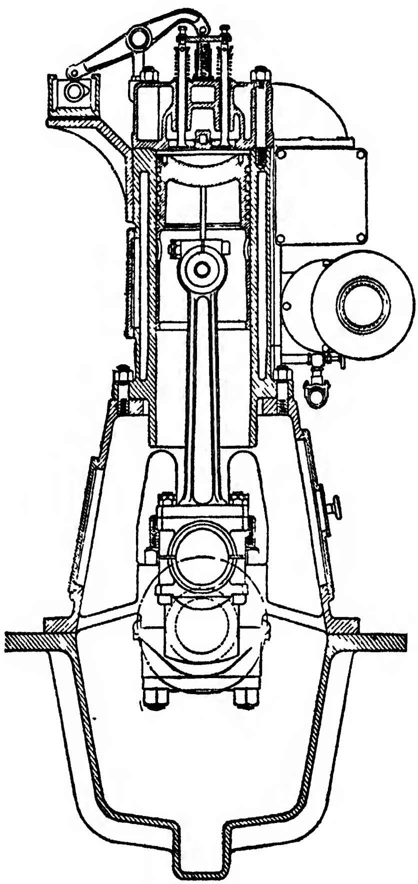


FIG. 26.—Winton (General Motors) early engine frame.

the same efficiency as the Type FH, already mentioned, using, for the sake of simplicity, solid injection as in the old Hornsby-Akroyd engine but dispensing with the hot bulb. Having occasion to notice the extremely fine water mist produced by opposing spray nozzles in an air conditioner, he set about to apply opposed fuel sprays to the oil engine. To prevent the oil from touching water-cooled walls, the combustion cavity was made up of two conical volumes, whose surfaces fairly conformed to the envelope of the oil sprays.

The combustion chamber (Fig. 27) was separated from the cylinder by a throat, and the turbulence produced an excellent mixture of fuel and air. Only a little clearance was allowed between piston crown and cylinder head, so that practically all of the air was driven into the combustion chamber, and all of the burning took place there. The first engine designed by Price

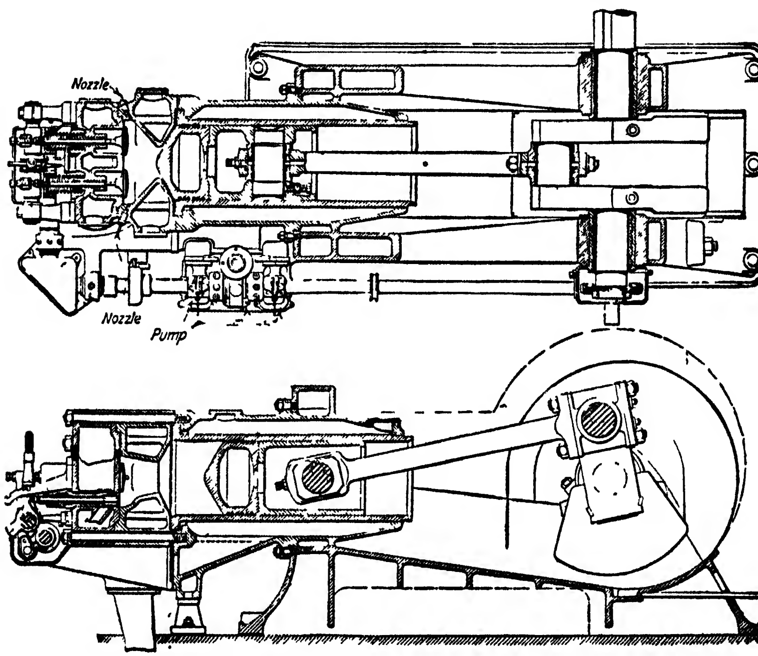


FIG. 27.—Cross section of De La Vergne S. I. engine.

showed the surprisingly good fuel economy of less than 0.4 lb. per brake horsepower-hour.

The Price engine as originally built had a compression of but 220 lb., and the chilling effect of the water-cooled walls prevented cold starting. Consequently a punk stick or cartridge was used. Later the pressure was raised to 300 lb., thereby giving self-ignition in starting. Combustion is at both constant volume and constant pressure by reason of the long injection period.

Price formed the Price Engine Company, which later sold the patents to the Rathburn-Jones Company, now a part of the

Ingersoll-Rand Company. The De La Vergne Engine Company holds shop rights, and the two companies have developed the engine along lines that differ only in mechanical details.

The Price engine is distinguished as being the first American design to be adopted by a European engine manufacturer; it is built by Carels of Belgium. De La Vergne Engine Company (a division of Baldwin-Southwark Corporation) builds the engine up to 1,600 hp. in eight 22 by 30-in. cylinders and as small as 65 hp. per cylinder.

Ingersoll-Rand.—Ingersoll-Rand builds the Price engine as small as 40 hp. in horizontal design and up to 1,000 hp. in vertical design. Investigation indicated that for high speeds the Price system was not suitable, as there was too great a pressure lag due to the separate combustion chamber; consequently, Ingersoll-Rand practically abandoned the Price design.

As with all engines having separate combustion chambers, this engine does not exhibit so high a mean effective pressure as does an air-injection Diesel. If over 75 lb. brake m.e.p. is obtained, combustion becomes poor, and the exhaust extremely smoky. To offset this, the mechanical efficiency is high, so that the fuel consumption is as low as with most air-injection engines.

Atlas-Imperial.—As has been pointed out, Atlas-Imperial first engaged in building air-injection engines. It was soon apparent, so the officials believed, that the demand would be for small engines which precluded air injection. This led to the development of a four-stroke solid-injection engine in which a mechanically operated needle valve was employed. Although McVickers had built an engine of this general type in 1915, the introduction of the Atlas-Imperial engine in 1919 marked the appearance of the first American engine of the "common rail" type. Individual cylinders with pushrod-operated valves characterized this engine, as will be discussed in later chapters. Although engines up to 300 hp. have been built, the company has centered upon small units, from 50 to 100 hp. running at from 400 to 600 r.p.m.

Fairbanks, Morse.—In the decade 1910-1920, the low-compression oil engine, familiarly known as the "hot-bulb" engine, came into general popularity and speedily surpassed the air-injection type in sales. The sales superiority of the hot-bulb oil engine in this period was due primarily to the entrance of Fairbanks, Morse & Company into the oil-engine field.

Fairbanks, Morse & Company started out in the internal-combustion engine field with the production of the Charter gas engine before the birth of the Diesel engine. This engine was followed by various modifications of spark-ignition and distillate engines. In 1912 its first horizontal, hot-bulb, or semi-Diesel, two-cycle-type engine was put on the market. This was of the

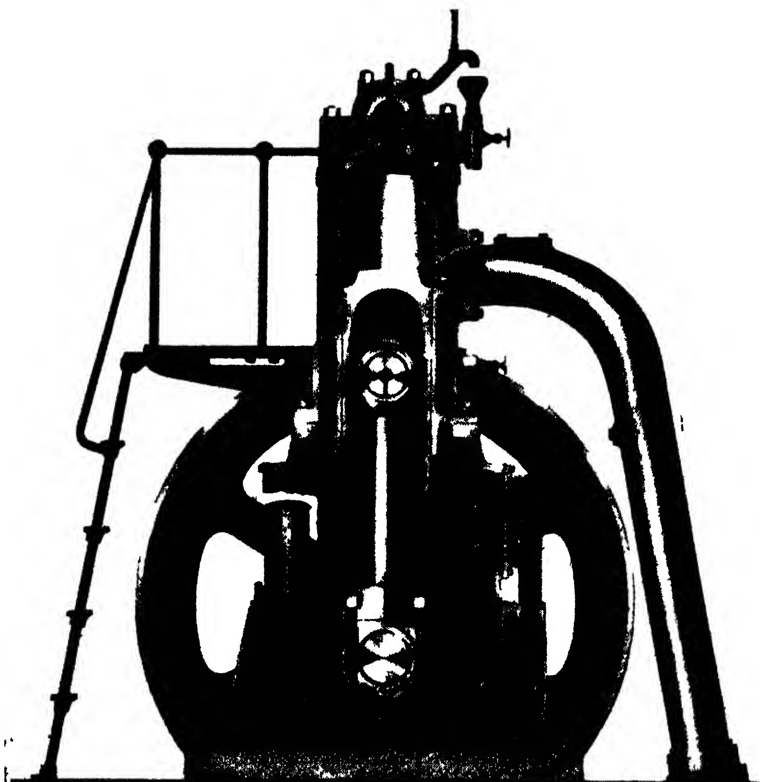


FIG. 28.—Fairbanks, Morse & Co. precombustion Diesel.

so-called "hothead" type, incorporating a small glow plug which was heated with a torch for the initial start. These engines were made in sizes of 10, 15, 20, and 25 hp., the smallest having a 6¾-in. bore and an 8½-in. stroke and operated at 450 r.p.m. These engines had a compression pressure of approximately 200 lb. per square inch and operated on kerosene, distillates, gas oils, and certain of the lighter crudes. They were of the crank-case-scavenging type.

The horizontal engines were followed in 1913 with a vertical semi-Diesel. The original engine was also of the hothead type, with glow plug, and originally water injection was employed. It took only a few months to discover that the advantages of water injection were more than offset by the disadvantages, and its use was discontinued.

The engine had a bore of 14 in. and a stroke of 17 in. and was rated at 50 hp. per cylinder at 257 r.p.m. Compressed air was used for starting.

The single-cylinder engine was followed with two-, three-, and four-cylinder modifications; and a six-cylinder, 300-hp. engine was built in 1920. Improvements were being made continually until these engines, which originally had a full-load fuel consumption guarantee of 0.60 lb. per brake horsepower, reached as low as 0.45 lb.

The hot bulb was replaced by a water-cooled combustion chamber, with a compression pressure of 150 lb. Later the compression was raised, step by step, to 450 lb., and the combustion chamber became a small water-cooled cavity in the cylinder head (Fig. 28).

Cooper-Bessemer.—Cooper-Bessemer Corporation (then the Bessemer Gas Engine Company) entered the slow-speed, heavy-duty, high-compression, mechanical-injection Diesel field by building a line of engines after the Atlas-Imperial common-rail system. This was discontinued shortly and replaced by an original Bessemer design. The common-rail principle, which was retained, embodied the Cooper-Bessemer system of atmospheric relief of the spray valves, with pressure-lifted spray valves in place of mechanically operated ones.

Sun-Doxford Diesel.—Shortly after the World War the Sun Shipbuilding and Dry Dock Company began building the Doxford opposed-piston, solid-injection Diesel (Fig. 29). This engine has two pistons and three cranks per cylinder. The two outside cranks are provided with long connecting rods which are coupled to crossarms of the upper piston. The center crank is connected by the usual rod to the lower piston. The fuel is injected into the cavity formed by the pistons when they are at "in" dead center. A compression of 300 lb. is used, and the engine functions excellently after the initial warming by live steam. The engine is two cycle, with the scavenging air supplied by a "blowing tub" in the center of the engine frame.

Many of these units have been installed on shipboard; and although they need considerable headroom, they have the advantage of being balanced

Worthington Diesel.—Worthington Pump & Machinery Corporation embarked on the development of an extensive line of vertical, trunk-piston, air-injection Diesels during the period 1920–1930 (Fig. 30). The corporation also constructed several

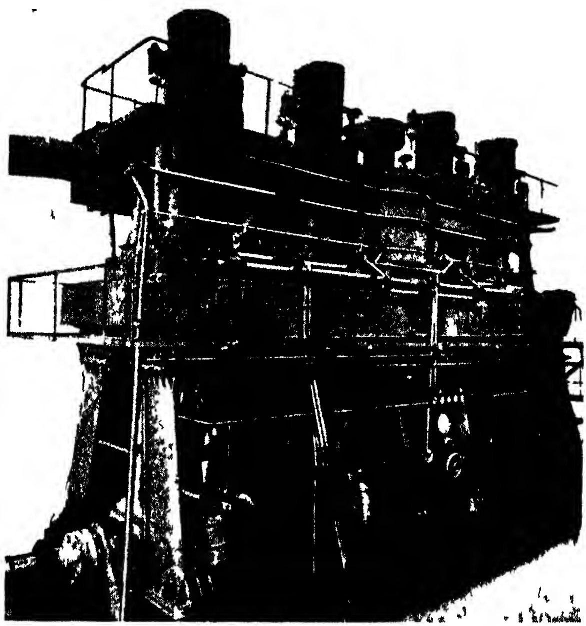


FIG. 29.—Sun-Doxford opposed-piston, two-cycle Diesel.

engines for the Shipping Board. These were double-acting, two-cycle engines embodying forged-steel cylinders and other novel details.

Other Builders.—Other builders of large air-injection Diesels were Hooven-Owens-Rentschler and Electric Boat Company, with a two-cycle, double-acting M.A.N. design.

Winton Engine Corporation expanded its range of four-cycle, air-injection Diesels up to 800 hp. in six cylinders, at 400 r.p.m. The corporation also developed a line of four-cycle mechanical-injection Diesels, employing a common rail.

McIntosh & Seymour.—Shortly after 1920 the company abandoned the A-frame engine and for units up to 500 hp. built a box-frame engine. This engine had uncooled pistons up to 175 hp. per cylinder capacity.

Beyond the frame itself the engine was modified in no great way, the details of valves, governors, etc., remaining as before;

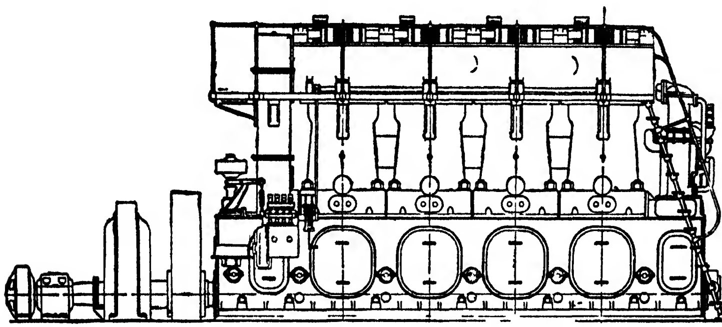


FIG 30—Worthington Type J four-cycle Diesel.

Fig. 31 is a view of the box-frame engine. These engines ranged in size up to 1,000 b.hp., a unit of six-cylinder construction.

For small powers this company manufactured a line of engines in which the frame and cylinder were cast in one piece with

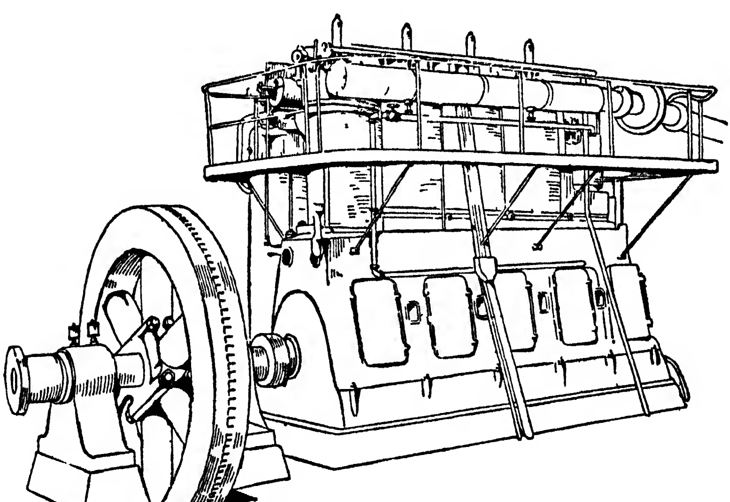


FIG. 31.—McIntosh & Seymour Type B box-frame Diesel.

inserted liners. It also built medium-speed, 300-r.p.m. designs up to 2,000 hp. of the type shown in Fig. 32.

The company built a crosshead-type vertical Diesel in sizes ranging from 750 to 3,000 hp. This engine had the camshaft running alongside of the engine at a level with the cylinder bases, the camshaft being driven by a train of spur gears from the engine shaft.

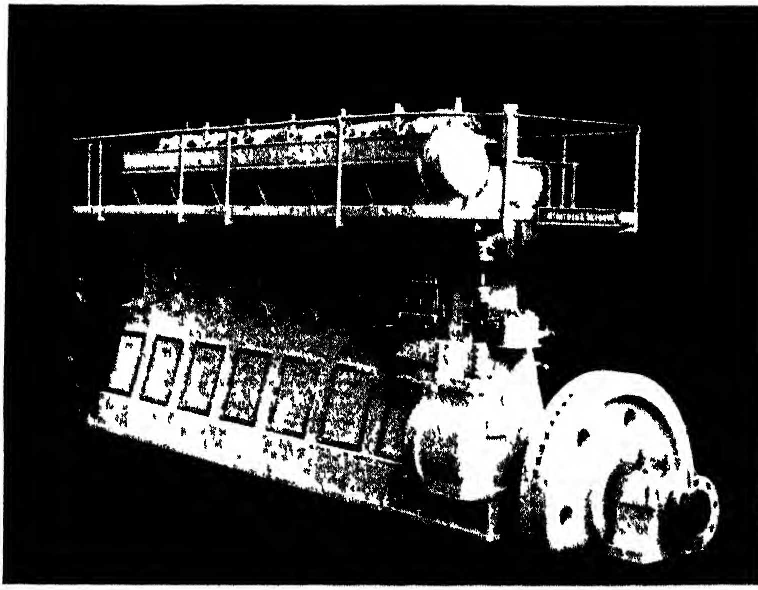


Fig. 32.— McIntosh & Seymour 300-r.p.m., air-injection, Diesel.

The valves were actuated by rocker arms and reach rods. The reach rods carried rollers which contacted with cams on the camshaft. The valves were in no way different from those of the Type A engine. Box girders consisting of a front and rear column cast together with a top cross member supported the cylinders. One of these frames supported the adjacent sides of two cylinders. This company is now a part of American Locomotive Company.

CHAPTER III

AIR-INJECTION DIESELS

General.—Up to 1930 the Diesel was, broadly speaking, an air-injection unit. Although scores of engines in which fuel is injected into the engine cylinder by the action of a blast of high-pressure air are still in service, almost all the Diesel builders have abandoned the system to build airless, or mechanical-injection, Diesels. It seems logical then, to deal with air-injection units before taking up modern mechanical-injection Diesels. Only the fuel-injection details will be discussed, for the other mechanical parts are identical with those of the mechanical-injection Diesels discussed in later chapters.

Air-injection Diesels.—Only three builders have constructed air-injection Diesels since 1935. One of these is the American Locomotive Company (formerly McIntosh & Seymour), which built a 2,850-hp., four-cycle, crosshead-type Diesel in 1937 for installation in the Rockville Centre, N. Y., Municipal Plant. This engine, with five cylinders of 32-in. bore and 42-in. stroke, turns at 120 r.p.m. A second engine of the same size was installed at Rockville Centre in 1938.

The second most important firm still building air-injection Diesels is the Nordberg Manufacturing Company. This company builds large two-cycle, air-injection Diesels to meet such conditions as low-grade fuels and a high load factor.

Busch-Sulzer Brothers have also built several air-injection Diesels.

Alco Air-injection Diesel.—The American Locomotive four-cycle, air-injection Diesel employs a subbase upon which rest vertical members, or frames. Across the tops of these frames are placed the separate engine cylinders. The base of the cylinder is extended to form a flange, and flanges of two adjoining cylinders rest upon each frame. A double-acting unit is shown in Fig. 33.

At the back of the engine a flat, vertical plate is bolted to each pair of frames. To this plate are bolted the guides for the crosshead. The crankshaft is supported by bearings resting in cross girders of the subbase. The piston is bolted to the crosshead. The interior of the piston crown is cooled by water which enters and leaves through telescopic tubes. The glands of these tubes are below the cylinders but above the top cover of the frame, so no water can leak into the frame, or crankcase.

Camshaft.—The camshaft is placed along the lower flanges of the cylinders and is gear driven from the flywheel end of the crankshaft.

Fig. 33.—McIntosh & Seymour double-acting, four-cycle Diesel.

Long pushrods from this camshaft operate the valve rocker arms.

Valves.—The admission and exhaust valves are in cages placed in the cylinder head. The cages and the exhaust valve are water-cooled.

Fuel Valve.—The fuel valve is discussed later.

Air Compressor.—The Alco injection-air compressor, of the three-stage type, is placed at the outer end of the engine frame and is driven by a crank at the end of the crankshaft.

Nordberg Air-injection Diesel.—Although Nordberg Manufacturing Company has built several different designs in the past, at present its air-injection, two-cycle engine follows the general appearance of Fig 66

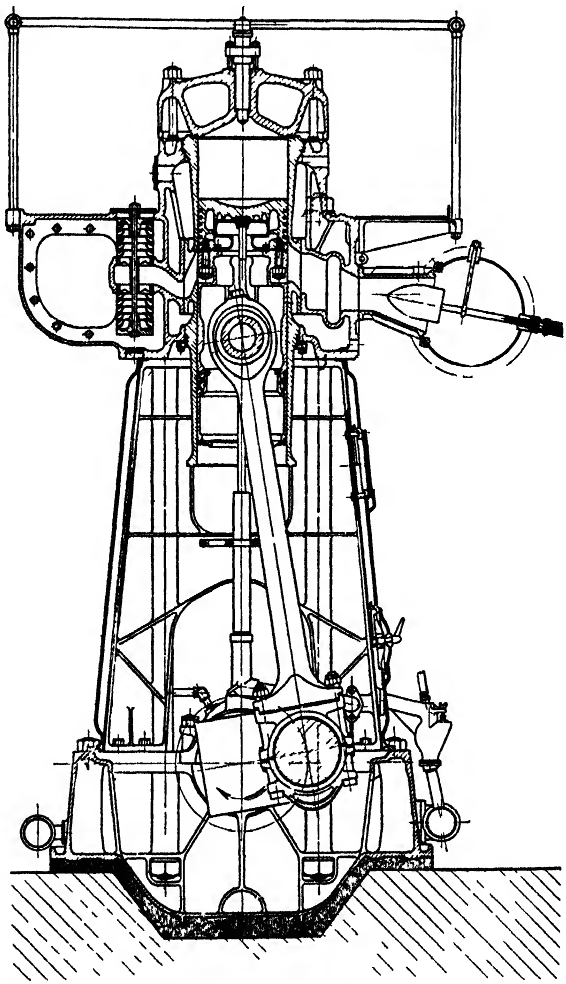


FIG. 34.—Section through Nordberg two-cycle, mechanical-injection, trunk-piston Diesel

A cross section of this engine is shown in Fig. 34. Although this shows a trunk-piston design, the same engine is obtainable with crosshead.

A subbase carries vertical frames, or stanchions, upon which are placed the cylinders. Each cylinder is a separate casting, but all are bolted together to obtain rigidity. The cylinder liner is provided with ports which register with exhaust and air ports in the cylinder casting.

It will be seen that the scavenging air enters and the exhaust gases leave through ports.

Piston.—The piston is two-piece construction. The crown is oil cooled by telescopic tubes; part of the lubricating oil reaching the piston pin through the hollow connecting rod flows up the piston rod to the piston crown and returns through a telescopic tube, as shown in Fig. 34.

Camshaft.—The cams lifting the fuel spray valve are placed alongside of the engine frame, with long pushrods swung from short links, pulling downward, to open the fuel-valve needle.

Lubricating System.—The crankshaft, crankpin, and piston-pin bearings are lubricated by oil under pressure. This oil is handled by a pump, chain driven from the crankshaft. A filter and oil cooler are placed between the pump and pressure header.

Earliest Design of Air-injection Fuel Valve.—Diesel first contemplated using coal dust as the combustible for his "rational motor," as shown in Fig. 35. The dust was to be deposited in a valve cavity and, by rotation of the valve, was to be dropped into the cylinder and burned. Nothing came of this idea, and it is only recently that a coal-dust engine, developed satisfactorily by Rudolph Pawlikowski of Gorlitz, Germany, has become a commercial proposition.

Diesel turned to oil for the fuel of his engine, and his initial designs embodied the mechanical injection of the oil. Hydrodynamics was not well understood at that time (1892–1893), however, and the inventor hit upon the happy idea of blowing the oil charge into the cylinder by highly compressed air. This worked exceptionally well and for years, until 1914, was the sole method used on high-compression engines of the Diesel type.

The first design of fuel valve and atomizer is surprisingly similar to designs employed on present-day engines.

Injection Action.—For the benefit of those unacquainted with the functioning of the fuel-injection valve the following brief explanation is given: The charge of fuel oil is pumped by some form of pump into a receptacle called the "fuel valve." An air compressor delivers a charge of air at about 900 lb. gage to this injection device. At the proper moment in the engine's cycle the needle valve of the injection device, or fuel valve, is opened, connecting the cylinder with the fuel supply. The high-pressure air then rushes into the cylinder, carrying the oil charge with it, the air having enough energy to carry the oil into the cylinder. This fuel, as it is forced through a device called the "atomizer," which is located in the valve housing, or cage, is broken up into fine foglike particles which will ignite when intermingled with the cylinder charge of air, which is at a high temperature. Figure 36 shows such a design, a fuel valve where

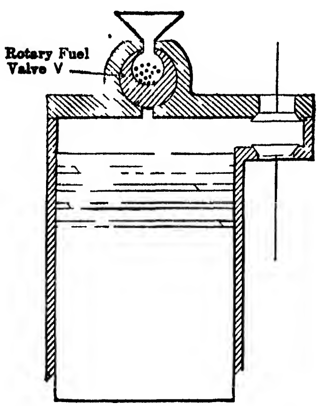


Fig. 35.—Fuel-feeding arrangement, Dr. Diesel's original patent.

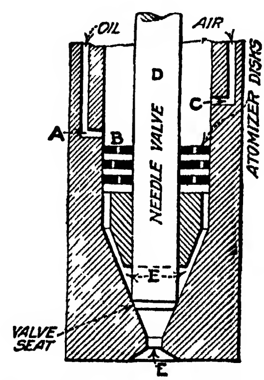


Fig. 36.—Typical air-injection fuel valve.

D is the valve stem, *A* the oil line, *C* the air line, *B* the atomizer, and *E* the opening into the cylinder.

It is evident that the fuel valve has two main functions. First, it must allow the oil charge to be introduced into the engine cylinder or combustion chamber at the proper time. Second, it atomizes, or breaks up, the stream of oil in such a thorough manner as to permit ignition. The cylinder temperature is high; the value corresponding to 550 lb. compression pressure should be at least 1000°F. when the engine is cold; after warming up, the temperature would be well above 1400°F. If the oil charge were injected as a solid mass into this highly heated air, the oil would vaporize and burn but at very slow rate. The air would not be thoroughly mixed with the oil, and the combustion would occur only on the surface of the oil mass in exactly the same manner as a pool of oil burns when ignited. The objection to this latter method lies in the slow rate of combustion and in the loss of fuel which escapes as unburned gas vapors. If the oil charge is separated into many minute particles, more surface area is presented to the air. If this oil-separation process is to be successful, the fuel valve must be provided with some means whereby the oil is broken into particles and mixed with the injection air before the oil enters the engine cylinder.

If the oil spray were to be finely divided and pumped into the cylinder without the air injection, the energy of the oil particles might not be enough to enable the oil drops to penetrate the air mass. The injection air supplies this energy.

The idea of using a charge of high-pressure air was taken up in the first experimental Diesels and has been retained on some Diesels to the present day, as it has certain advantages not possessed by the direct-pump system. The air charge in the fuel, or spray valve, being at a pressure 300 or 400 lb. higher than the cylinder pressure, provides an excellent means of forcing the oil through the valve passages into the cylinder as soon as the needle valve opens. The expansive power of the air, as it drops in pressure from 900 to 500 lb. on entering the cylinder, increases its volume and tears the oil into minute particles. A well-designed valve will break up the oil so finely that, if the engineer holds his hand in front of the fuel nozzle, the oil will not even wet the hand but will float as a mist.

By reason of the chilling effect of the air in expanding through the valve from its high pressure to the existing cylinder pressure, a temperature drop of 50 to 100° may be experienced by the cylinder air charge. This necessitates a slightly higher compression pressure and temperature than is required by the solid-injection oil engine. The chilling effect is of consequence only at the moment the needle valve opens and starts the flow of air and oil. After the oil ignites, any further chilling is of little importance, for the cylinder temperature upon ignition of the oil immediately rises to 2000 to 2700°F., and the chilling effect of the air injection is of little detriment. It is evident, then, that the elimination of chilling when the spray valve opens is desirable. This may be, and is generally, accomplished by so designing the valve that a small amount of oil enters the cylinder ahead of the air. This oil upon ignition raises the cylinder temperature high enough to avoid any bad effect of the air chilling. At the same time the amount of oil so entering must be small; for not being so finely broken up as the balance, it would, if of any great amount, cause soot or carbon deposits.

If the oil is hurled into the cylinder in a mass, its combustion, even though imperfect, would cause the pressure to rise to an abnormal value. The air-injection Diesel engine is presumed to burn the oil at constant pressure; to accomplish this the oil must be introduced at a comparatively slow rate. The two main offices the valve performs are that of "braking," or offering a resistance to the oil flow, and that of thoroughly atomizing the charge. The latter would not be difficult of achievement if no other considerations entered into the problem. The same conditions prevail as to the braking action. If this braking effect is obtained by the imposition of a series of improperly designed baffle plates, or disks, the air pressure required to force the oil through these resistances may become so great as to make the method impractical. The desirable fuel valve is one that thoroughly atomizes the fuel and exercises a control, or a braking action, over the fuel charge without any great loss of air pressure through the valve.

The design of the orifice between the needle valve and cylinder has a decided influence upon the combustion within the cylinder. The combustion space in all Diesel engines is limited in volume, since the compression ratio is large. If this volume (about one-fifteenth of the stroke volume) is in the shape of a cylinder between a flat piston top and a flat cylinder head, it is difficult for the oil to mix with the entire air charge. Instead, without proper nozzle design, the oil will be localized in the central portion above the center of the piston crown. It will be imperfectly burned because of the limited amount of air in contact with the oil spray. Carbon deposits will appear on the center of the piston head and lead to its overheating and fracture unless the piston cooling is good. If the oil is introduced along the axis of the cylinder head, as is the usual custom, it must be spread by some means. In view of the short depth of the combustion space of a flat-headed engine, the engineer will perceive that this spreading is by no means easily accomplished. It would, on first thought, seem that a series of outlets would distribute the oil better than a central orifice, but results of the two different types of openings apparently differ but little. It has been found on certain engines that the use of a multi-hole orifice, or flame plate, has improved conditions. With a flaring central orifice the oil seems to spread about as well in most cases.

Classes of Atomizers, or Fuel Valves.—The many fuel valves employed on the various makes of Diesel engine fall into two classes—the closed-nozzle and the open-nozzle valves. The former was the one adopted on the pioneer Diesels and is found on all vertical engines of the present day; in fact, this is necessary for structural reasons. The open nozzle, known in Europe as the "Lietzenmayer" nozzle, was used on the Allis-Chalmers horizontal engines.

Fuel Valve.—The earlier Diesels employed the closed nozzle, and it was exclusively used on practically all air-injection Diesels. Figure 36 outlines the basic principle of this nozzle. The fuel valve has a body in which a cavity is formed, inclosing the atomization device. The fuel needle valve is seated below this device and is actuated by a cam-controlled lever. The interior of the fuel valve is in connection with the air line through the passage and is at all times under an air pressure of 900 lb. per square inch or more. The fuel pump forces the oil charge through the line A, the oil settling around the valve stem at E above the valve seat. When the needle valve opens, the compression pressure of the engine is around 500 to 550 lb., whereas the air pressure in the valve body is about 900 lb. This pressure difference results in a rapid flow of the air into the cylinder. The air charge forces the oil along with it, and, in passing through the tortuous passages of the atomizer disks, the oil is completely nebulized.

This form of fuel valve has the advantage of depositing the oil in a receptacle entirely isolated from the influence of the hot compressed air in the cylinder. Furthermore, part of the oil, being immediately around the valve tip, enters the cylinder ahead of the air and ignites, even though it is not thoroughly atomized. This primary ignition provides a flame to fire the remainder of the oil, which enters the cylinder at a somewhat low temperature due to the expansion of the air charge at the valve tip. Unfortunately, with many closed-nozzle designs, an entirely too great percentage of the fuel enters the cylinder ahead of the air; in some it appears that all the oil is forced ahead of the air. If the disks are designed with perforations of small diameter to enable the air to mix with the oil, the "braking," or resistance, of the atomizer is increased, since the disks are entirely filled with oil at full load. This compels the employment of a higher injection pressure or prolongation of the time interval of fuel injection. On low loads, with small perforations, the fuel charge is of small weight and does not flow down around the valve

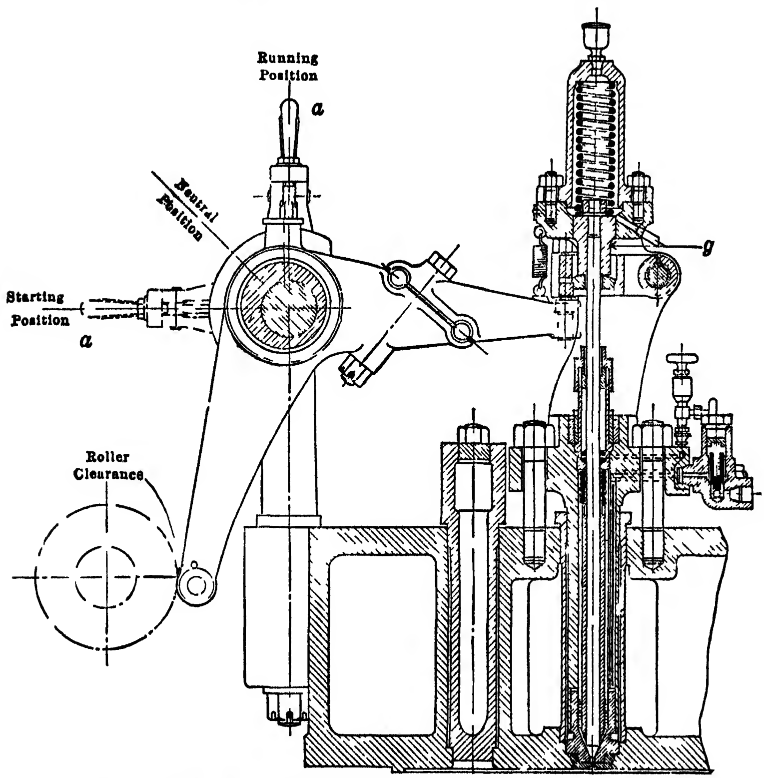


FIG. 37.—Fuel-valve arrangement of the Busch-Sulzer Type B Diesel.

seat. The consequence is that the first part of the injection consists of air only, which, in expanding, chills the nozzle tip and housing. This delays the combustion, producing a smoky exhaust.

Hardly any two Diesel manufacturers followed the same design in the fuel valve, although there was some degree of similarity in a few designs.

Busch-Sulzer Type B Diesel.—The fuel valve of this four-cycle Diesel, built by Busch-Sulzer Brothers up to about 1930, appears, with its rocker mechanism, in Fig. 37, and Fig. 38 gives a view of the lower part of the same valve. The valve consists of a cast-iron body with an extension which carries the spring, a needle valve, and an atomization device. The body, or cage, rests in a bushing, which is pressed into the cylinder head and is held by two studs. The needle valve is enclosed along the lower part, which is in the cage, by a bushing or barrel. This bushing rests, at its base, on the atomizer cone and is prevented from lifting by a coil spring at the top. As is seen in Fig. 38, there is a space between this bushing and the cage wall, which serves as the air and fuel cavity. The fuel charge flows down the passage A and enters the fuel cavity above the atomizer disks. The disks are several plates containing small perforations and are so placed as to stagger these holes. The oil, by its own weight, is forced to pass through these openings and fills the intricacies between the plates as well as the space around the needle-valve seat. The air enters the cavity d above the oil level.

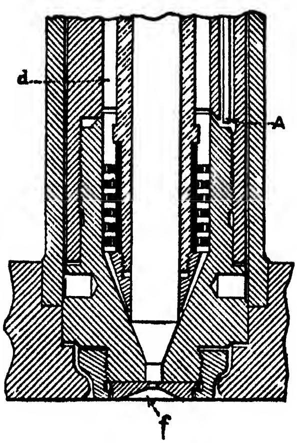


Fig. 38.—Busch-Sulzer Type B fuel valve.

When the needle valve is opened by the valve rocker, the air, which is at a pressure much higher than that existing in the engine cylinder, forces the oil charge into the combustion space. The oil, as it is broken up, passes from disk to disk and is mixed with the air that is flowing toward the cylinder. The emulsion is further increased as the air and oil issue through the single small opening in the atomizer tip f. This continues as long as the needle valve is open; this valve is closed at the proper time by the fuel cam (see Fig. 37), and the small amount of fuel still left in the atomizer flows down around the needle-valve seat. This insures that a small charge of oil will enter the cylinder at the next valve opening ahead of the air and produce ignition. With the usual fuel oil the fuel charge is able to pass through the disks under the influence of its own weight. When the valve opens, the oil is forced into the cylinder in front of the air. Since this oil is fairly light, it ignites even though poorly broken up. A heavy oil behaves quite differently. Its viscosity is such that the fuel does not readily flow through the disks but rests above them. The air must force the oil through the atomizer and, in so doing, thoroughly nebulizes the charge. This type, then, has the advantage of offering a mixing and atomizing effect in inverse ratio to the gravity of the fuel; the actual degree of combustion in the cylinder is fairly constant, regardless of the nature of the oil.

Fuel Cam.—The fuel cam is fitted with an adjustable steel nose. The nose is slotted and is held by two countersunk screws. This allows a considerable shifting of the nose. After the nose is set in position, it is locked by end shims. These shims are best made of wrought iron and should be hammered until the entire recess is filled; the surface should then be smoothed with a file to conform to the curvature of the cam.

Eccentric Rocker Bushing.—As has been mentioned, the fuel rocker arm is fulcrumed on an eccentric bushing, which also carries the starting rocker. On a four-cylinder engine the two inside cylinders are fitted with starting valves. When the engine is to be started, the eccentric lever *a* (Fig. 37) of these two cylinders is thrown into the starting position. This revolves the eccentric bushing until the fuel rocker fails to engage its cam while the starting rocker comes into contact with its cam. The levers of the outside cylinders (1 and 4) are set to "neutral," which disengages the fuel rocker and cam. As soon as the air-line valve is opened, the two starting cylinders turn the engine over. After one or two revolutions of the flywheel, the levers of 1 and 4 are moved to the running position, admitting fuel to the fuel valves. When these cylinders start firing, the levers on 2 and 3 are moved from the starting to the running position, cutting out the air-valve mechanism and engaging the fuel rockers.

Servomotor.—On some of the larger Busch-Sulzer engines the timing of the fuel injection is altered at load changes. This is accomplished through the agency of a cylinder, placed in front of the engine, containing a spring and piston. This cylinder is in communication with the low-pressure cylinder of the air compressor. The air-compressor suction is provided with a damper arrangement actuated by the engine governor. On low loads the air to the compressor is throttled. This results in a lower discharge pressure in the low-pressure cylinder; this, in turn, lowers the pressure existing in the servomotor. The spring then forces the piston downward. The piston rod moves a system of levers which actuates an auxiliary roller which is linked to the fuel-valve rocker. As the air pressure in the servomotor becomes lower, owing to a lighter load, the auxiliary roller moves upward, thereby allowing the cam nose to strike it slightly earlier in the engine cycle. The auxiliary roller is in contact with the fuel-rocker roller at all times, being held by links. The auxiliary roller is set in such a position that if the injection opening is early, as on low load, this roller remains in contact with the cam nose for a smaller interval. Consequently, on full load the period of injection begins later and extends over a greater crank angle than it does on low loads.

On starting the engine, the handwheel of the servomotor is raised to the half-load position while the air from the air compressor to the servomotor is cut off. This latter action is for the purpose of preventing the servomotor from moving the injection roller to the full-load position. Frequently, when the engine is using heavy oil, it is necessary to adjust the servomotor handwheel to provide for earlier admission than the air pressure would give, or vice versa with heavy oil on light loads.

Fuel-valve By-pass.—Each fuel line has a by-pass, or relief valve, mounted on a block at the camshaft cover. Before the engine is turned over under air pressure, these valves should be opened and left in this position until a solid stream of oil issues from each valve, thus indicating that all air in the pump or oil lines has escaped. When running, frequently one or more cylinders skip in firing or fail to fire at all; this, in most cases, is due to an air-bound fuel line.

McIntosh & Seymour Spray Valve.—The fuel valve of this engine, appearing in Figs. 39 and 40, is contained in a cast-iron housing, or cage. The valve, the needle end of which has a 60-deg. included angle, seats on the cage itself. The upper end of the cage carries the spring housing. This end of the valve stem is screwed and locked into a dashpot, against which the spring bears. The valve is opened by the movement of the rocker arm shown. The atomizer is the detail of the valve that differentiates it from other American fuel valves. This is a gun-metal barrel, the diameters of which are of irregular dimensions. The interior is bored taper, fitting the valve stem at the upper end, and

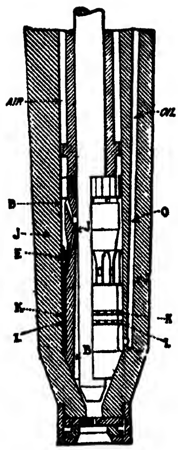


FIG. 39.—McIntosh & Seymour fuel valve.

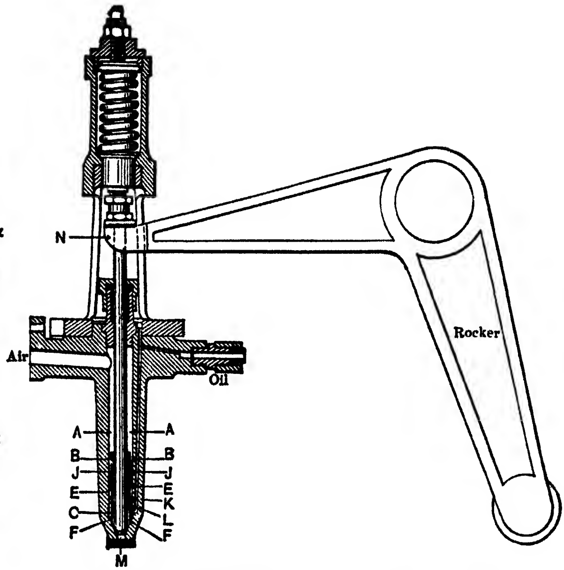


FIG. 40.—McIntosh & Seymour fuel-valve assembly.

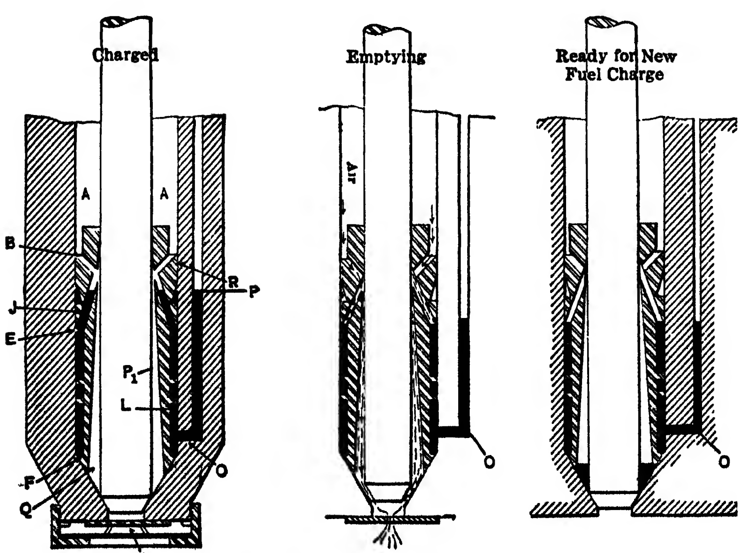


FIG. 41.—McIntosh & Seymour Hesselmann fuel-valve action.

a number of passages connect this taper bore with the outside surface. In operation a charge of oil is forced by the fuel pump through the passage *O* and surrounds the atomizer at the space *J*, issuing through the serrated fins at *K* and *L* until the fuel reaches the level *P* (Fig. 41). The injection air fills the cavity above the atomizer, and, when the needle valve opens, this air flows through the ports *B* and along the valve stem into the cylinder. The air pressure on the surface of the oil at *J* remains constant while the velocity of the air current along the valve stem reduces the pressure at the inner end of the oil ports *E* The consequence is a flow of oil through the oil ports *E* under the influence of the unbalanced air pressure. This oil, as it enters the stream of high-velocity air, is broken up and thoroughly atomized by the time it reaches the cylinder through the atomizer cap *M*. The flow of oil continues until the oil level falls below the ports *E*. If the fuel valve is properly timed, the valve should then close, preventing an excessive amount of air from blowing into the cylinder. Although this excess air may actually increase the mean effective pressure of the engine as figured from an indicator card, it represents a loss of power, since it has been compressed to 900 lb. per square inch and is allowed to cool to nearly room temperature, thereby losing the energy of the compression work.

On full load the closure of the needle valve should trap a small amount of oil immediately above the valve seat. This oil, on the next valve opening, is blown in ahead of the air charge, providing an initial ignition to balance the chilling action of the expanding air charge. The oil, if it contains dirt or a tarry base, gums badly until the fuel chamber is filled, forcing the oil to deposit around the valve stem. When this occurs, the oil enters the cylinder in a slug. This is indicated by loss of power and a smoky exhaust. The remedy is, of course, the cleansing of the atomizer.

It is claimed that this valve design permits operation with a lower injection-air pressure than with other types of atomizers. In practice it would appear that this advantage does not exist, as at full load a pressure of about 900-lb. gage is necessary.

This design of spray valve is superior in many respects. Of special advantage is its ability to handle successfully the heaviest of Mexican oils. Unlike many spray valves this one is able to atomize heavy oils without the formation of carbon on the piston crown.

Starting Fuels.—For starting purposes, it is customary for the manufacturer to furnish a two-compartment fuel tank, one compartment containing the fuel oil which is used, while kerosene is placed in the other part. This kerosene is supplied to the engine in starting, since it will ignite at a lower temperature.

Nordberg Diesel Fuel Valve.—The spray valve (Fig. 42) used on the large Nordberg Diesel is also found on the smaller type engines without change in design, save that the former has a small rocker which serves to transfer the motion of the large

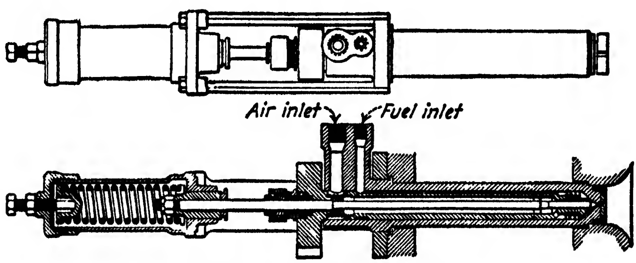


FIG. 42.—Nordberg fuel valve.

rocker to the needle valve. This rocker is necessary in the large engines, since the large rocker has its roller resting on top of the fuel cam rather than under or between the camshaft and cylinder head. On the small units the camshaft is below the cylinders, and the rocker is actuated by vertical pushrods, the rocker bearing directly upon the needle-valve lock nuts.

The fuel-valve body fits into an opening in the cylinder head, the water being sealed off by a thin bushing. Within the valve housing is placed a long pulverizer tube which carries a set of perforated disks at its lower end. The fuel flows down the clearance space between the body and the tube while the air flows between the needle valve and tube, emerging through drilled passages above the disks. The flame plate carries one central nozzle opening with flaring exit walls, thereby enabling the hollow cone of oil, formed by the flow along the conical needle-valve end, to spread over the combustion space. The large engines have flame plates with five holes to give distribution into the entire clearance volume. Later Nordberg engines have spray valves of a different design.

Maintenance of Fuel Valves.—Superior advantages are claimed for each type, but from the point of view of maintenance there is little choice. The spray-valve troubles most commonly encountered are scoring and cutting of valve and seat, causing valve to leak; jamming of stem in its guide, preventing valve from seating; valve-stem packing leaking; and atomizer plates or passages clogged, preventing flow of oil and air out of valve body. In the case of a leaky seat or valve, its correction may involve more or less regrinding, depending on the depth of the cuts or scores on valve and seat. In no case should recutting or facing be resorted to if it is possible to make the valve tight by grinding, but it should be remembered that frequent regrinding is likely to result in the seat wearing out of true or the valve face becoming uneven.

Regrinding Fuel Valves.—Regardless of the type of fuel valve, each, sooner or later, requires regrinding. In performing this process, an engineer should be very miserly with the amount of grinding paste used. The best compound is one of powdered glass and vaseline or of emery flour and vaseline. Only a very small amount should be placed on the needle, being spread out evenly over the entire seating surface. The entire valve, with the exception of the spring, should be assembled when grinding. This is to insure that it is aligned properly. It is unnecessary to secure more than a thin line contact at the seat—$\frac{1}{32}$ in. in width is ample. After grinding, it is advisable to disassemble the entire valve and cage and wash very thoroughly with kerosene. This is to remove all emery particles.

It is impossible to lay down a hard and fast rule as to the frequency of regrinding. If the fuel is fairly clean, once every 2,000 to 4,000 hr. of service seems to be about correct. Poor filtering equipment may cause valve leakage much before this.

Some engine builders supply a special reamer for truing the spray-valve seat; but if it is not supplied, the engineer should have one made. The reamer should be a duplicate of the valve tip, with the face of the cone fluted to form cutting teeth. The reaming should be very carefully done to obtain good results. The reamer should not be dropped down on the seat, nor should very much pressure be applied, as it will chatter and cut an irregular seat. Insert the reamer carefully; hold against the seat with a light pressure; and turn only in one direction. If it is turned backward, the chips collected between the cutting teeth will scratch the valve seat, and a great deal of grinding will be required to remove the scratches. The liability to scratch will be reduced if the cutting teeth are coated with heavy cup grease. This grease will catch and hold the metal chips and prevent them from being dragged under the teeth. Remove the reamer frequently; clean and recoat with grease; wipe out the valve seat; and continue the operation only long enough to obtain a good full seat without removing any more metal than is necessary.

It is general practice to make the valve seats of cast iron and the valves of high-grade alloy steel, hardened. For this reason it is usually the valve seat that requires recutting, and the valve need be ground only to the seat; but when the valve is scored enough to require recutting it is best done in the lathe, using a center grinder in the compound rest as a cutting tool. The rest is set to the angle of the valve seat, and very light cuts are taken, using a soft grinding wheel and revolving the valve about 200 r.p.m.

Needle-valve Wear.—The spray-valve stem guide is usually made as long as possible in order to reduce the wear and insure that the valve seats absolutely true. It is a common mistake among engineers to assume that sticking of the valve stem in the guide is due to a too close fit between stem and guide. As a matter of fact, it is in most cases caused by this fit being too loose. The spray-air pressure inside the spray-valve body tends to force the fuel oil up into the space between the guide and stem, especially when leakage of air past the valve-stem packing creates a condition of unbalanced pressure in the valve-stem guide. This fuel oil is spread in a thin film over the surface of the valve stem, and the heat of the stem gradually cooks it down to a gummy condition. This gum, or soft carbon, is very gritty, with no lubricating value, and will soon cause the stem to stick in its guide. When the spray valve is disassembled for overhaul, the stem and guide should be carefully cleaned with kerosene. Emery cloth should not be used to polish the stem unless it is too rough to be cleaned with oil, as it tends to reduce the diameter of the stem and increase its clearance in the guide.

In some cases the hydrogen sulphate in the fuel oil causes severe corrosion of the needle.

Spray-valve stems are packed with high-grade soft or metallic packing. In replacing, do not use packing that contains rubber.

Rubber is readily solvent in fuel oil, and the packing will be destroyed rapidly. If the stem shows cuts or abrasions where the packing bears, it should be smoothed and polished with emery cloth to prevent the packing from being torn.

Bent Needles.—If the valve stem is bent, it should be straightened and spun between centers to see that it runs true. Do not replace a stem that has been straightened without testing the seating of the valve. If the spring has taken a permanent set that shortens it enough to decrease its working tension, place enough washers under the spring or on top of it to give it the right tension. Since it is not practicable in the ordinary engine room to measure this tension, the engineer must necessarily set it by observing the action of the valve. The tension should not be so great as to put unnecessary load on the operating gear, but it should be enough to overcome the inertia of the valve levers and pushrods and permit the valve to snap back rapidly to its seat after the pressure of the valve lever is removed.

Lubrication of Valve Stem.—The valve stem gradually becomes coated with a thin layer of oil residue, this being more noticeable in non-cooled valves. To prevent binding of the stem, it should be constantly lubricated. A small amount of kerosene injected around the valve at least once every 24 hr. will remove any residue.

Leaky Fuel Valves.—A leaky injection valve usually betrays its presence by causing the engine exhaust to be smoky. Leaky valves also allow the fuel charge to seep into the cylinder during the compression stroke and so produce violent preignition.

Incorrect Fuel-valve Timing.—If the valve opens too early, a sharp metallic click or pound will be heard in the cylinder. This is evidence of premature combustion. If the valve opens late, a dull thump or pound, quite like a pound due to a loose pin bearing, can be heard. Furthermore, a smoky exhaust ordinarily accompanies this pounding.

Clogged Atomizer or Nozzle-tip Disk.—When either the atomizer or the disk at the end of the fuel valve, sometimes called the "burner plate," is clogged, the exhaust is smoky.

Sticking Valve Stem.—When the fuel valve stem sticks in the open position, the exhaust will be smoky, and the injection-air gage will show a drop.

This sticking may be due to a gummed stem or to excessive packing friction. The engineer is cautioned against allowing such a state of affairs to exist. There is great danger that the flame in the cylinder will flash back through the fuel nozzle, since the air pressure will be low by reason of the free passage into the cylinder. This flash-back will burn the flame plate or nozzle and the needle tip. In a few cases the explosion has wrecked the full valve cage and air lines. If the edges of the flame-plate

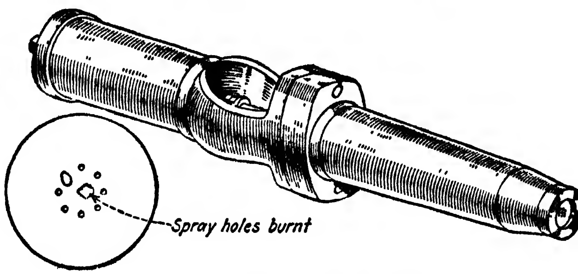


Fig. 43.— Burned flame plate.

orifices seem to be burned, as in Fig. 43, it is proof that there has been a flash-back.

Buried Needle Tips.—If the needle spring is too heavy or too light in compression, the needle will chatter in closing. Ultimately this will lead to a badly worn seat in the valve body. To rectify this, operators will re-ream the seat. If rereaming is

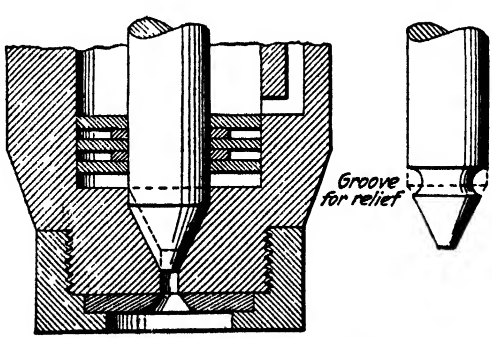


Fig. 44.—Buried fuel-needle tip.

continued, there is a likelihood of the needle seating so deep in the body, as shown in Fig. 44, that when it is lifted the round part is still below the top of the seat cone. If this happens, the fuel cannot pass by the needle stem and down the seat. The result is that the entire oil line to this spray-valve casting is filled with oil. On the next pump stroke the pump plunger comes up against this solid mass of oil, and something must give way. If the governor acts directly on the plunger, the governor becomes irregular. In other cases the oil line must relieve itself by leaking at the joint, or the high pressure may raise the needle valve at the wrong time. The danger in continued rereaming is obvious. As a remedy it is recommended that a groove be cut around the stem at the juncture with the tip. This groove will provide a clearance for the oil when the needle lifts. A better plan is to renew the lower end of the fuel-valve cage which forms the valve seat.

Fuel-valve Cooling-jacket Temperature.—The desirable temperature at which the discharge line from the fuel-valve water jacket should be carried depends on the characteristics of the fuels. If the oil is heavy and viscous, the discharge should be around 160°F. With fuel oil of 28°Bé. and higher, 120°F. is amply high, since the valve must be cool to prevent gassing of the light oil.

Adjustable Injection-air Pressure.—The engineer can appreciate the necessity of having a higher injection-air pressure when the engine is carrying full load than when under a light load. When the fuel charge delivered to the fuel valve is large, as on full load, the resistance, or braking action, of the atomizer is high and requires a high pressure to force the entire charge of oil into the cylinder. On light loads the oil occupies only part of the atomizing space, and, consequently, a light air pressure is sufficient. If the pressure is high on light loads, the oil is blown into the cylinder at an increased rate. The passage of the fuel would then require only part of the time during which the needle valve is opened. The remainder of the period of valve opening would be devoted to the passage of pure air. The high velocity of the free air as it left the nozzle tip would chill the tip and lower the entire cylinder temperature, causing a decreased cylinder efficiency as well as a direct loss of air that has been compressed at a considerable expense of power. Furthermore, if the air pressure is high on low loads, a sharp knock is produced in the cylinder which results from the inrush of air at a pressure far above cylinder pressure. Conversely, if the air pressure is too low, the engine will smoke, since the fuel has not been sufficiently atomized.

It is necessary for the successful operation of any Diesel that the injection-air pressure be altered to conform to load change. This adjustment can be under manual control of the engineer, as is the general practice. The manual control can be obtained in several ways. The McIntosh & Seymour marine engine and the crosshead-type stationary engine are provided with a clearance chamber on the low-pressure cylinder, whose volume can be altered, changing the air-discharge pressure. Other builders arrange for the operator to adjust the low-pressure suction, obtaining the required air-pressure control. On fluctuating loads, however, this entails constant attention and is more suit-

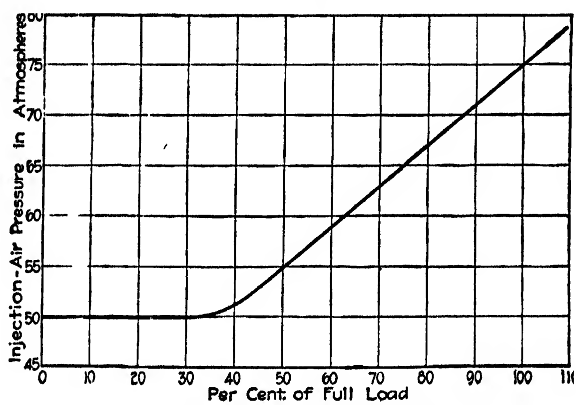


Fig. 45.—Injection-air pressure.

ably handled by some automatic arrangement. There are several designs of automatic injection control. The Busch-Sulzer Diesel throttles the compressor suction through a linkage from the engine governor. The Standard fuel-oil engines use a governor-controlled air by-pass valve.

Injection-air Pressure.—Each design of fuel atomizer requires a certain air pressure to inject the oil properly. This pressure will vary with the resistance or braking action of the atomizer disks. It follows that no set pressure values can be given that will apply to all engines. The curve in Fig. 45, however, gives the relation between engine load and injection pressure that is correct for practically all closed-nozzle engines. In making this statement it has been assumed that the difference between injection-air pressure and cylinder pressure was a constant. This holds perfectly true for similar types of engines but is not true for such widely different types as a high-speed submarine engine and a low-speed heavy-duty stationary or merchant-marine engine. Both types use about the same cylinder compression pressure (about 475 lb. per square inch), but the combustion pressure in the high-speed engine rises to perhaps 600 or even 700 lb. per square inch, whereas in the heavy-duty engines it is only 550 lb. or even no more than the compression pressure. The best injection-air pressures are found to be from 1,000 to even 1,200 lb. per square inch for the high-speed engines and generally about 800 lb. per square inch for the low-speed engines. These differences are due to the conditions of working. From the high-speed engine the highest possible power output for a given piston displacement is desired; but in the case of the low-speed engine, reliability and economy are the main objects sought.

In any case the lowest possible injection-air pressure that will give the results wanted is the best. The lower the pressure the easier the duty on the air compressor and the less the chilling effect on the combustion in the cylinder, due to the expansion of the air. With a new type of engine the best method to be pursued is to run a series of tests with gradually increasing sizes of nozzles and varying injection pressures with each nozzle. A point will soon be found where increased size brings no advantage, and that size with the proper injection pressure is the one to be adopted.

At full power the engine will run with a wide variation in injection-air pressure, and the economy will not be badly upset unless an extreme pressure is used; but for light loads the available range of pressure is very small, because too great a pressure will give such a chilling effect that the temperature in the immediate vicinity of the fuel blast will be lowered to the point where no ignition will result, and too low a pressure will not atomize the fuel. In fact, with a very low pressure there is danger of burning the spray-valve seat or even doing more serious damage. It is, therefore, never advisable to drop the injection-air pressure below 550 to 660 lb. per square inch under any circumstances. The available range of pressure at the low end of the scale is perhaps 50 to 100 lb. per square inch. For full-load conditions, if the very best pressure is, say, 850 lb., acceptable pressures would probably range from 825 to 900 lb. The economy is more seriously affected by a reduction in pressure than by an increase.

In order to overcome some of the difficulties just mentioned, certain engine builders arrange to have one-half of the cylinders cut out at very light loads. This increases the load on the other cylinders enough to give regular ignitions; and not only is the engine less sensitive to a variation in spray-air pressure, but, incidentally, greater accuracy of governing is secured. Some builders even go a step further and vary the spray-valve lift with the load, which practically eliminates any sensitiveness due to injection-air pressure variations. Under these conditions there is little difference between the best pressure at full load and that at light load, but this is under the control of the governor and does not need to worry the operator.

Marine-Diesel Air Pressures.—The marine-engine case presents greater difficulties than the stationary, since the propulsion power decreases approximately as the cube of the ship's speed, and the revolutions only directly. It is customary to fit rather large air compressors in order to have a surplus capacity for refilling the starting air tanks and also to provide a margin for reduced-speed running. As the speed is reduced, the total time that the spray valves are open during a second or a minute is the same, and, consequently, with the same air pressure the air consumption is constant in spite of the reduction in speed. The reduced air pressure that is used reduces the consumption to a certain extent; but since the production is only directly proportional to the speed of revolution, a point is reached at one-third to one-half speed where the supply fails to keep up with the demand. By this time the power required has also dropped to such an extent that ignition troubles begin, so that this becomes the lower operating limit for practical purposes with constant time of opening injection valves. With special care and a skilled operator this limit can be lowered, but under ordinary conditions one-third speed is about the limit. In most cases this is no disadvantage whatever, since the full speed is generally 10 to 12 knots, and one-third of this speed is barely steerage-way. To express it in another way, a marine engine operates at nearly full power and speed most of the time; and provided it can be slowed down on occasion and can be maneuvered satisfactorily, all requirements will be met.

Finding the Best Pressure.—The simplest method of determining the best injection-air pressure is to watch the exhaust temperature. Without a doubt, the exhaust-temperature pyrometer is one of the most important instruments used around an oil engine, because no matter what happens to upset the combustion, the result is immediately registered on the pyrometer. The operator may need all sorts of instruments and special tests to locate the trouble, but the exhaust temperature, if watched and compared, never fails to give a warning when something is wrong with the combustion. Consequently, the way to find the best injection-air pressure for a given set of conditions is to vary the pressure and watch the pyrometer. When it registers the lowest, record the pressure for future reference and use; once determined, it will always hold until some condition is changed. The whole procedure is so simple that there is no excuse for an operator not knowing the best pressure for any condition. It would be sufficiently important if it were only a question of economy, but it means reliability as well. Poor combustion and the resultant high exhaust temperature invite trouble all around.

Adjustable Fuel-valve Timing.—The usual Diesel-engine fuel valve is designed with a constant period of valve opening, regardless of load conditions. In the Otto-type explosive engine the efficiency of the engine depends on the maximum explosive pressure. With the Diesel engine the efficiency depends both on the combustion pressure, which should be, but never is, identical with the maximum compression pressure, and on the duration of the fuel injection. It is very clear that with load changes the time during which the fuel is injected should also vary. Since the rate of combustion should be constant, the period of injection must vary if the greatest possible efficiency is to be secured. Furthermore, a factor of operation also enters into the problem. On low loads the amount of oil is small and will be entirely blown into the cylinder long before the valve closes. The balance of the valve-opening period is taken up with the injection of high-pressure injection air. This air assists in no way toward the combustion. For these reasons several builders have adopted a form of injection-timing control.

Timing of Fuel Valves.—In timing a fuel valve the engine is pinched over until it is several degrees ahead of the desired point of fuel-valve opening. The air-line valve is "cracked," giving about 75 lb. air pressure on the fuel valve. The indicator plug is removed, and the engine is slowly barred over until the trammel cuts the opening mark on the flywheel. The injection valve should now start to open, as evidenced by the sound of injection air blowing into the cylinder. If the valve opens before the mark is reached, the rocker clearance can be increased, producing a later opening. If the valve opens late, the clearance can be reduced. The engine should be barred on to the closing mark, and the sound of the escaping air should cease as the mark is reached. Since the roller clearance has been altered to make the opening earlier, the closing point will probably be late. It then becomes necessary to turn the engine back ahead of the valve-opening mark and shift the cam nose. The nose should be shifted to produce the required opening with the roller clearance correct. Then, on checking the closing point, it should either be correct or early. If the latter, the nose must be shifted back a trifle and the roller clearance made less. This should produce the required opening and closure. If the nose is excessively worn, it is impossible to obtain a correct timing, and a new nose must be secured. This method works very well with small engines; but in the case of a large engine, in which the cylinder volume near the time of fuel admission is quite large relative to the area of the hole through the indicator cock, a slight movement of the piston causes quite a rush of compressed air through the indicator cock, and it is not always possible to determine the exact point of spray air admission.

There are several other ways of determining the exact time the valve lifts from its seat. An approximately correct method is to start revolving the cam roller when the cam toe is just about to make contact. When contact between the cam and its roller has progressed sufficiently to jam the roller so that it cannot be turned, the valve may be considered open. This method is not accurate, as the clearance in pin bearings, deflection in levers, etc., may result in the valve leaving its seat an appreciable time after contact between cam and roller is made.

The most satisfactory method of all is to make use of a dial test indicator such as is generally employed in machine shops. This instrument is shown in Fig. 46. It may be used on its base, or the arm *B* may be removed from its holder and clamped to any stationary part of the engine structure near the valve being set. Pointer *A* is placed in contact with the valve stem, and any movement of the stem will be registered by the hand on the dial. The dial can be read to one-half of one-thousandth of an inch, so the exact point of opening of the valve is easily determined. By this method all error due to lost motion in the valve gear is avoided, as only the actual valve movement affects the indicator.

The valve having been set to open at the correct point, as called for by the builder's valve diagram, the time of closing and

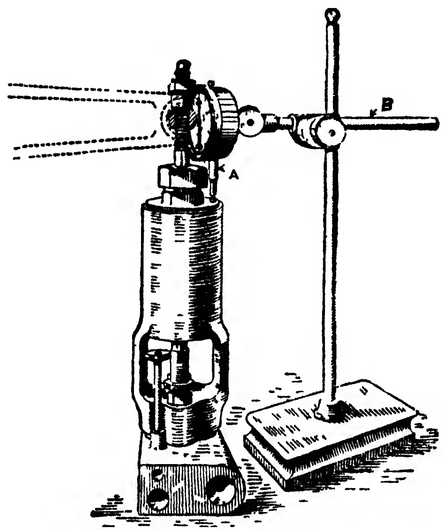


FIG. 46.—Setting fuel valve by dial gage.

total lift should be measured just as a matter of record. If the dial test gage is used, the maximum lift may be read directly from the dial. The same measurements should be taken for the astern cams.

Checking the Valve Settings with Indicator Diagrams.—After the fuel valves have been timed, the engineer should apply an indicator (discussed in a later chapter) to each engine. Any difference in the diagrams should be investigated, for it is possible that variation in the cam outline or in the roller clearance is the cause of the difference. It is not an unusual thing for the spray valves on an engine to vary as to time of opening. If each valve is set to open, say, at 5 deg. before top center, it may be found upon taking indicator diagrams that one or more valves may be slightly late or slightly early, and it may be necessary to vary the times by as much as 1 deg. in order to get a good combustion line.

Worn Cams.—Because of cam outline, sometimes the rocker roller does not follow the cam but jumps, causing ridges beyond the cam nose. This is also traceable at times to the valve spring being too weak or to the stem sticking.

Backlash.—The camshaft gears are not immune to wear, and in the course of 5 to 7 years of constant service the backlash between the gears may become noticeable. The clearance between the gear teeth has a very detrimental effect on the injection cam. As the cam nose contacts with the valve-rocker roller, the pressure that the spring offers against the rocker movement is considerable. As the roller travels over the surface of the cam nose and starts down along the back slope, this spring pressure forces the camshaft forward, causing the valve to close early. Since the wear is between the teeth, the camshaft is already behind its exact timing with the engine shaft, and consequently the opening of the fuel valve is late, and the closure is early. A new cam nose of greater length will partially overcome the defect, but new gears should be ordered to replace the worn ones.

Fuel Pumps.—The fuel pumps required for mechanical-injection Diesels are so different from the pumps of air-injection engines that the two types must necessarily be handled in separate chapters. The mechanical-injection pump must handle the oil at high pressures; and the discharge rate, and consequently the pump-plunger velocity, is much higher than that of the air-injection pump.

The fuel pump of the air-injection Diesel delivers against an injection-air pressure of, say, 900 lb. gage, and the discharge from the pump to the fuel valve may continue over a greater part of the engine's piston stroke. As a consequence, the pump can be built amply large and without undue complications as to plunger seal. For this reason the designer is presented with the comparatively simple problem of building a mechanism that is airtight, has serviceable valves and plungers, and has a proper control to enable the quantity of fuel going to the fuel-spray valve to be regulated according to the existing engine load.

Fuel-pump Capacity.—The ratio of the fuel-pump capacity to the cylinder volume of a Diesel engine depends upon the idiosyncrasies of the designer, but taking the average air-injection Diesel engine with 70 lb. m.e.p. per brake horsepower, the power-stroke volume per brake horsepower can be figured from the formula $PLAN/33{,}000$, and if we substitute for LAN, the power-stroke volume V, and for P use a value of 70 lb. brake m.e.p., we have a volume per brake horsepower of 3.3 cu. ft. per minute. Now, engine designers have the pump-stroke volume ranging from 1.6 to 2 times the actual volume of oil needed for the engine to develop its full horsepower. On the basis of 1.6 times the volume, and 0.44 lb. of oil per brake horsepower, the pounds of oil handled per hour per brake horsepower would be 0.7, which would be approximately 21 cu. in. per brake horsepower per hour; it would be 0.33 cu. in. per brake horsepower per minute. Since the power-stroke volume is 3.3 cu. ft., this gives a ratio of 1:17,280, and designers have used ratios as high as 1:25,000.

Types of Fuel Pumps.—Fuel pumps used on air-injection Diesels may be separated into three types or classes. The first of these covers the pump designs wherein the stroke of the fuel pump plunger is changed to meet the load variations, in this way controlling the amount of fuel handled by the pump.

The second class includes those pumps in which the suction valve is held open during a portion of the pump plunger's delivery stroke. In this way part of the fuel drawn into the pump-plunger cavity is allowed to flow back through the suction valve. Only the oil handled by the pump during the latter part of its pressure stroke and after the suction valve has closed is forced into the spray valve.

The third type or class includes those pumps having a separate by-pass valve, which by-passes part of the oil handled by the pump back into the suction line.

Busch-Sulzer Type B Diesel Fuel Pump.—Figure 47 is a cross section of the Type B pump, showing one plunger. This is along designs adopted by the majority of early European and English air-injection Diesel builders and is the pump used on Busch-Sulzer four-cycle engines built prior to 1930. In this construction the plunger A is driven by an eccentric B keyed to the vertical governor shaft C. The suction valve is mechanically operated by a dog F, which swings on an eccentric G. This eccentric is mounted on a small shaft controlled by the governor K through a linkage J and bell-crank I. The suction-valve plunger *D* is also driven by a fixed eccentric *E* on the vertical governor shaft, being 180 deg. behind the pump-plunger eccentric. In Fig. 47 the pump plunger is at the end of the delivery stroke, when the suction-valve plunger is at the extreme inner position. As the governor shaft revolves at one-half engine speed, the pump plunger moves to the end of its suction stroke; the suction plunger moves outward, lifting the suction valve off its seat; this allows the fuel to enter the pump cavity. As the pump plunger reverses and moves on its delivery stroke, the suction valve remains open the oil flowing back into the suction line.

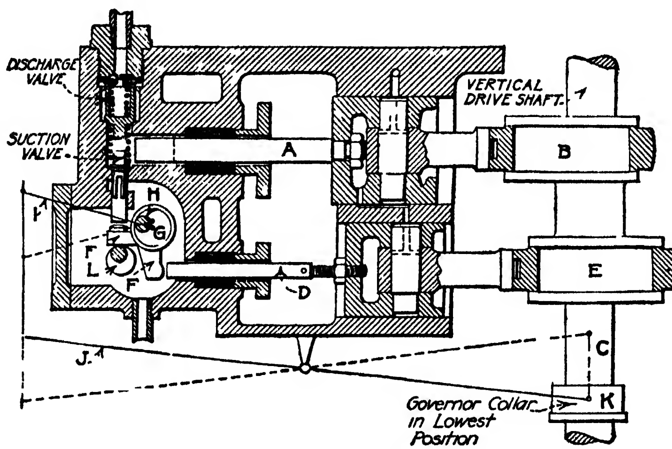


Fig. 47.—Busch-Sulzer Type B Diesel fuel pump.

At a stated point in the pump-plunger travel, the suction plunger moves out of contact with the dog *F*; the valve now closes, and the oil is forced out through the discharge valve. If the load decreases, the rising governor sleeve shifts the center of the eccentric-dog bearing. This allows the valve plunger to remain in contact with the suction valve for a longer interval, which permits more of the fuel charge to flow back into the suction line.

On these engines there is a pump plunger and suction-valve plunger for each engine cylinder. It is possible to use only one suction-valve plunger, but this individual suction-valve mechanism offers opportunity for a closer adjustment of the functioning of each pump. On some of the four-cylinder engines the four plungers are arranged in a single row, whereas on others there are two sets of two plunger cavities each, placed end to end with the valve block in the center.

This fuel pump gives the closest possible speed regulation while the reaction on the governor is at a minimum. The resistance offered to the movement of the governor sleeve consists of the suction-valve spring compression.

Filling the Fuel Pump.—To facilitate charging the pump and discharge line with oil, an eccentric shaft *L* is provided. This shaft is rotated by means of a hand lever, and thus the dog *F* is raised to its maximum lift. This lifts both the suction and the discharge valves.

Stopping the Engine.– A less angular travel of the eccentric shaft *L* lifts only the suction valve. This relieves the pumps of the oil charge, and the engine stops from the lack of fuel.

Setting Pump Valves.—The four plunger pumps deliver the fuel to the four cylinders on all four cycles, that is, suction, compression, expansion, and exhaust strokes. In timing the pump and the suction-valve opening, the engine is slowly turned over, and the inner and outer dead centers of the pump-plunger eccentric are marked on the plunger. The engine is then turned over until the pump plunger is $\frac{7}{16}$ in. from its discharge dead center. The suction-valve plunger, or regulating plunger, has at this point moved away from the dog or bell-crank, which leaves its contact with the suction-valve stem. The suction-valve plunger should be adjusted to give a clearance of 0.002 in. between the dog and valve stem when the pump plunger is in the position mentioned. In setting the valve, the governor collar must be central. In a two-plunger pump this means that the collar must be in the mid-point of its travel, whereas with a four-plunger pump the collar must be on its bottom position; to obtain the latter the governor springs must be removed.

Busch-Sulzer Type C Pump.—On two-stroke-cycle, air-injection Diesels Busch-Sulzer Brothers employed a pump of a different design. This is shown in Figs. 48 and 49.

This fuel pump, installed on engines built prior to 1935, is operated by an eccentric 1 mounted on the vertical cam drive shaft at the after end of the engine, the eccentric making one revolution for each revolution of the crankshaft. The pump is provided with plungers 2, each plunger serving an individual cylinder.

The plungers are of steel, hardened and ground and mounted on the driving yokes 3, which are connected by the side rods 4.

The plungers, yokes, and rods form a rigid combination, which receives its reciprocating motion from the eccentric.

The pump body is a forged-steel block, drilled to provide the necessary passage for the in- and outflow of the fuel oil and to

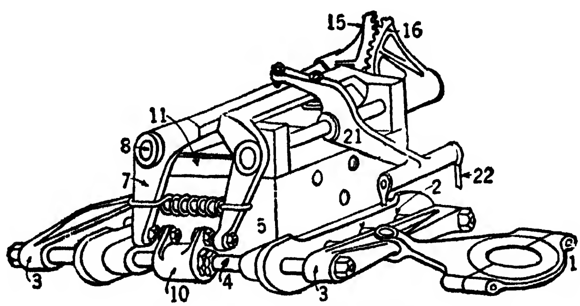


FIG. 48.—Perspective view of pump used on large Busch-Sulzer engine.

receive the plungers and valves. A separate passage delivers the fuel to each individual suction valve; and a separate passage discharges it from each individual discharge valve.

During the outstroke of a plunger, fuel is drawn into the plunger barrel through the suction passage 23, the suction valve

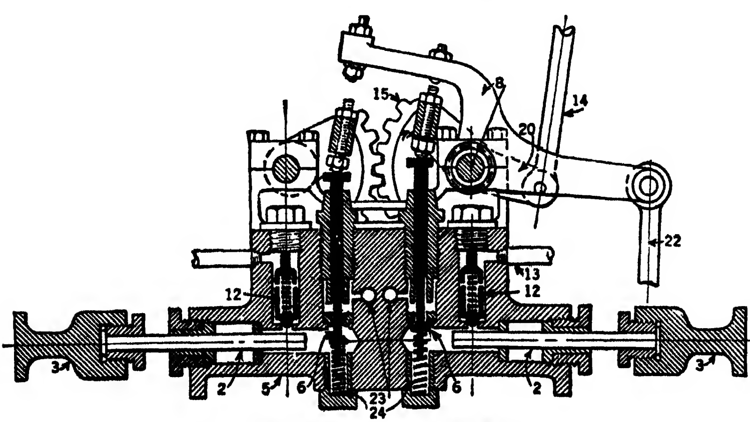


FIG. 49.—Cross section of Busch-Sulzer fuel pump.

being held open by the rocking lever 7 pivoted eccentrically upon the shaft 8. The rocking lever 7 receives its oscillating motion from the block 10, which is rigidly mounted upon one of the side rods and provided with adjustable tappet screws, which push against hardened screw heads on the lower ends of the rocking levers 7, the parts being held in contact by the spring 11.

During the instroke of the plunger, the fuel is discharged from the plunger barrel back through the suction valve, until the rocking lever 7 has moved away sufficiently far to permit the spring 24 to close the suction valve; the remainder of the fuel is then forced out of the plunger barrel past the discharge valve 12 through the discharge pipe 13 into the fuel-spray valve on the working cylinder.

The period during which the suction valve is held open is adjusted to suit the desired engine speed by the fuel-control lever at the operating station. The movement of the fuel-control lever is transmitted through the control rod 14 to the lever 20, which is keyed to the shaft 14. The teeth of the gear-segment lever 16 mesh with the teeth of the gear segment 15, the segments being keyed to the shafts 8. The turning of these shafts affects the movement of the eccentrically pivoted rocking levers 7, increasing or decreasing the length of the open period of the suction valves according to the direction in which the shafts are turned.

Early McIntosh & Seymour Diesel Fuel Pump.—The first Diesels manufactured by the McIntosh & Seymour Corporation were equipped with fuel pumps somewhat after the design appearing in Fig. 47. The engines were four-cylinder units, and the fuel pumps had two pumping plungers, one for each pair of cylinders. The fuel from one plunger cavity passes through the discharge valve and pipe line into a block, called the "distributor." This block contains two passages connected to the fuel lines leading to the two fuel-injection valves. The cross sections of these passages are controlled by needle valves. The fuel entering the distributor divides into two streams; the needle valves allow the operator to proportion the two oil streams properly.

The operating difficulty of this pumping system lies in the inability of the engineer to regulate the distribution of the fuel on varying loads. A setting of the needles that is correct for full load will not give the proper regulation at low loads, since the resistances of the passages vary, because of a smaller quantity of oil entering the distributor. Another factor that prevents proper proportioning of the fuel is the partial clogging of one line; this throws almost all the oil into one cylinder.

With this form of pumping mechanism it is imperative that the oil be filtered to prevent the clogging of a distributor. Furthermore, the fuel valves must be kept in perfect condition, since the smallest leak in a fuel-injection valve lowers the resistance of this particular fuel line, allowing this cylinder to receive too large a proportion of the fuel from the pump.

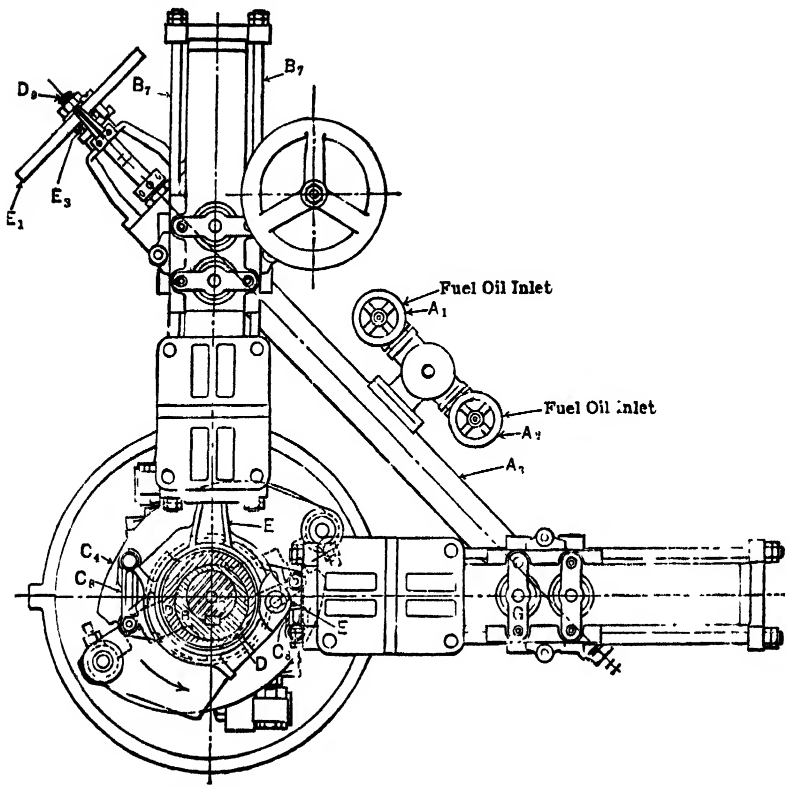


FIG. 50.—Governor and fuel pump, top view.

McIntosh & Seymour Fuel Pump.—The fuel pump mentioned above gave way to a design shown in Figs. 50 and 51. Unlike the distributor-type pump which followed Sulzer Brothers patents, Fig. 50 is of exclusive American design and was employed on all the company's trunk-piston, air-injection Diesels prior to 1930. This pumping apparatus consists of two pumps set at right angles, each being an outside-packed, double-plunger pump. The eccentric strap *E* driving the two plungers is mounted on an eccentric *D*, and the governor acts directly on the pump plungers, in this way regulating the amount of fuel pumped by varying the plunger stroke. The reaction on the governor of direct plunger-controlled pumps is considerable. McIntosh & Seymour partially avoid this by using two eccentrics; the eccentric *D*, controlled by the governor, drives the pump and is mounted on a second eccentric *B* which is keyed to the vertical governor shaft *C*. It is apparent that this offers a more accurate regulation of the pump stroke and a greater reduction in the reaction on the governor than can be secured by a single eccentric.

The pump suction valves are located below the discharge valves, being removed through the discharge-valve opening.

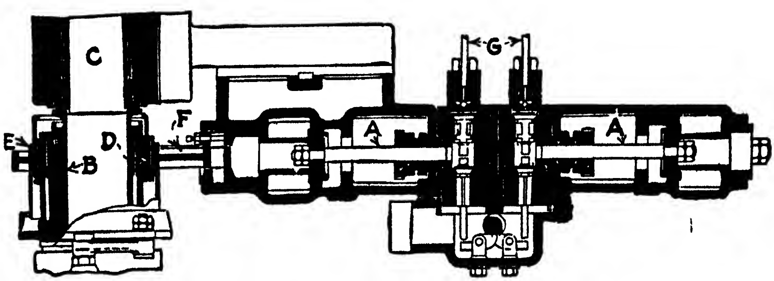


FIG. 51.—McIntosh & Seymour fuel pump.

The latter valves are accessible by the removal of the valve cap or plug *G*, Fig. 50. Below the suction valves of the pumps is placed a shaft *H*, which has two milled cams, as shown in the cross section. During the functioning of the pump the suction-valve stems clear the shaft by means of the depressions in the shaft. Rotation of the shaft lifts the suction valve, thereby filling the pump with oil.

Various arrangements of plungers are used at present in place of the right-angle one described. In some of the six-cylinder units, six pumps are placed radially, with the plungers connected to a common eccentric.

In other engines, especially eight-cylinder ones, the pump bodies are placed in a row along the top of the frame, with all the plungers hooked to side rods driven from one eccentric.

The advantage of this pump lies in the fact that the several plungers have a common governor linkage, and as a result each cylinder receives the same amount of fuel.

Pump of McIntosh & Seymour Large Diesel.—In the cross-head-type, air-injection engines this company employed the same design, save that all the pump plungers fitted into one pump-body forging, so arranged that three plungers entered one side and three the other side of the body, with all plungers connected to a common drive rod through side straps. The pump was placed at a level with the engine crankshaft, and the governor

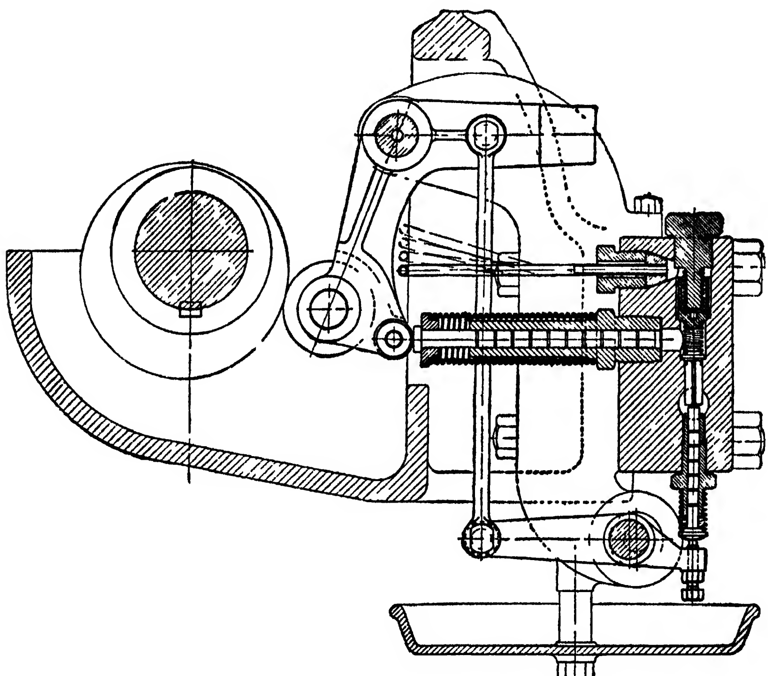


FIG. 52 – Section through pump of 17 by 25-in., Worthington, air-injection, Diesel.

was fitted to the engine shaft. It consisted of two weights keyed to a spider bolted to the crankshaft; and these weights shifted the eccentric driving the pumps. An outside spring-loaded lever was used to adjust the given weight tension.

Worthington Vertical Diesel Pump.—The vertical air-injection Diesel of this make used the fuel pump shown in Fig. 52.

The fuel pump was mounted on the front of the camshaft trough and below the Massey-Jahns governor. The body of the pump consisted of a single forging bolted directly to the camshaft trough, insuring rigidity and permanency of the fuel-pump adjustments. The individual fuel-pump plungers were of hardened steel ground and lapped to a perfect fit in separate plunger barrels. No packing of any kind was necessary. The pump plungers were operated by cams mounted on the main camshaft. The quantity of fuel delivered by each plunger was controlled by holding the suction valve open during part of the plunger discharge stroke. This was accomplished by a pushrod under each suction valve, mechanically operated by and interlocked with the fuel-pump plunger lever working on an eccentrically mounted shaft, the position of which was controlled by the governor, so that by shifting this eccentrically mounted shaft the point in the discharge stroke at which the suction valve closed was altered by the action of the governor. The control of the fuel pump could be taken away from the governor by a hand-operated throttle, and the amount of fuel delivered to each spray nozzle could be fixed by the proper setting of this throttle. Each plunger could be operated by a hand lever for filling the fuel lines with fuel before the engine was started.

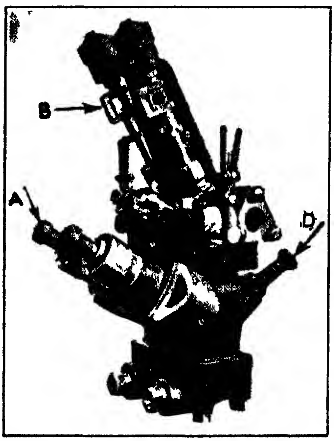


FIG. 53.—Nordberg FD fuel pump.

Owing to the spring loading of the pump-plunger bell-crank, all lost motion was taken up, and the pump operated quietly. The grooving of the suction-valve stem enabled leakage to be avoided. The design was a very practical one.

Nordberg Type FD Fuel Pump.—The FD pump is shown in Fig. 53. The pump was a double unit; that is, it was provided with two plunger and by-pass valves so that two engine cylinders were supplied with oil from each unit. The pump plunger *A* was a hardened-steel rod whose outer end was held in contact with the fuel cam on the camshaft by a spring inside the plunger housing. The regulation was obtained through the action of the by-pass valve. The valve rod of this valve was fitted with a scotch yoke *B*. A link connected this yoke to an eccentric mounted on the camshaft. The governor was connected to the fulcrum shaft of this link, and, by shifting, this eccentric fulcrum point altered the stroke of the by-pass valve A hand pump *D* enabled the oil lines to be primed before starting the engine

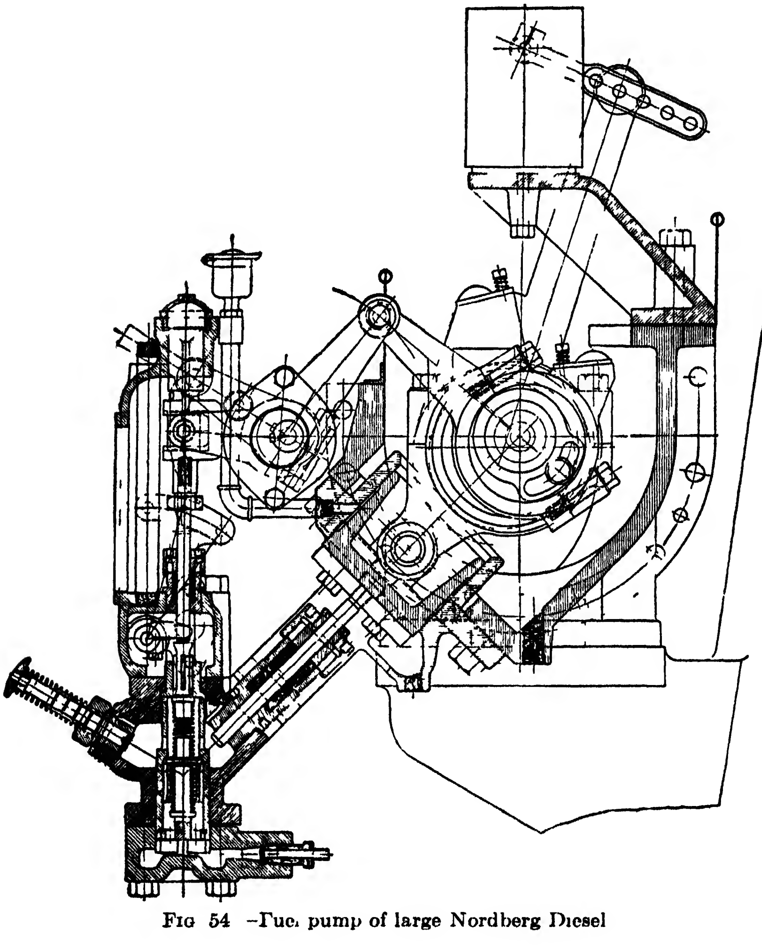


Fig 54 —Fuel pump of large Nordberg Diesel

Later Nordberg Diesels had a fuel pump of the design shown in Fig. 54. This was basically the same as that in Fig. 53, but the pump plungers were driven by eccentrics on the camshaft, as were also the suction valves. Control was through the shifting of the eccentric suction-valve bell-crank, as will be seen by an examination of the illustration Present-day Nordberg air-injection engines use Bosch fuel pumps.

Pump Maintenance.—After the valves of any design of fuel pump have been replaced, after grinding, etc., the discharge line should be disconnected from the injection valve, if no by-pass valve is attached. The pump should then be primed. This will effect the escape of any air that might be trapped in the pump cylinder. In case the engine, on starting, fails to fire on any particular cylinder, the pump is probably air bound, and the lines must be freed before oil will enter the fuel valve. At times the fuel-line check valve leaks, allowing air to flow down the pipe to the fuel pump. This will prevent the pump from delivering any oil.

Valve Grinding.—In grinding any poppet-type valve, powdered glass, or emery flour, and vaseline make the best compound. The operator must exercise judgment in the amount of pressure exerted on the valve as it is rotated. If the pressure is excessive, the compound will groove the valve faces.

When a discharge valve or suction valve is under suspicion, the best method of ascertaining if it actually leaks is to disconnect the discharge line and place on the coupling a high-pressure gage. The engine can be turned over until the pump-discharge pressure registers the usual value, 900 lb. with closed nozzle and about 200 lb. with open nozzle, assuming the engine is of the air-injection type. The engine should then be stopped, and the discharge pressure noted. It it falls, it can be taken as an indication of a leaky valve.

CHAPTER IV

MODERN HEAVY-DUTY, MECHANICAL-INJECTION DIESELS

General.—As was pointed out in the Preface, it is the intention of the author to devote most of his volume to operation and maintenance of mechanical-injection Diesel engines. In later chapters the design variation in the important engine parts will be pointed out, along with the functional duties of these parts, followed by outlines of handling, repairing, and adjusting the parts.

Before taking up engine parts, it seems highly desirable to give the reader brief descriptions, along with illustrations of the commercially important, mechanical-injection Diesels now being built in America.

Although this particular chapter will give only a general description of a particular engine, in later chapters the various important parts of the particular engine will be covered quite comprehensively.

Because of space limitations, only those engines being manufactured today will be covered.

Atlas-Imperial Diesel.—Atlas-Imperial Diesel Engine Company started building air-injection Diesels in 1914. This design was abandoned in favor of a mechanical-injection Diesel using a mechanically operated needle spray valve after the English McVickers design. This engine, consisting of a bedplate, frame, and individual cylinders, is still built. The company, however, developed a line of medium-speed, en-bloc-frame, four-cycle Diesels which are fitted with the common-rail fuel system and mechanically operated spray valves. Fuel injection is direct into the cylinder space; compression is from 450 to 500 lb. One of these engines appears in Fig. 55; this engine has all parts placed inside dustproof casings. It is not only applied to stationary power plants but also to boats, drag-line excavators, shovels, and other industrial machines. Cylinder sizes range from 6-in. bore and 7-in. stroke to 9-in. bore and 12-in. stroke.

Alco Diesels.—American Locomotive Company (purchaser of McIntosh & Seymour Corporation) builds both two-cycle and four-cycle mechanical-injection Diesels.

For its locomotives and for many stationary-plant applications, the company developed the line of four-cycle units. Sizes range from 300 to 900 hp. and to 1,200 hp. in a supercharged engine. Cylinders are from 9½-in. bore by 10½-in. stroke to 12½-in. bore

Fig. 55.—Atlas-Imperial medium-speed, four-cycle Diesel.

by 13-in. stroke. The same general design is also followed in a 17½-in. bore and 25-in. stroke unit.

Bosch fuel-injection pumps and differential-needle spray valves are used, with the fuel spraying into the cylinder space. A concave piston crown creates a semispherical combustion chamber.

The engine has an en-bloc frame resting on a subbase, with the latter supporting the crankshaft. Cylinder liners are inserted in bored recesses in the frame. The overhead valve gear on each cylinder head is concealed by a light steel cover. The camshaft, located in the frame, is gear driven from the crankshaft, while pushrods carry the cam action to the top-of-cylinder valve rockers. One of these units appears in Fig. 56.

Alco Two-cycle Diesel.—The American Locomotive Company secured the American license for the building of two-cycle Diesels after the Swiss Sulzer Brothers design.

An eight-cylinder, 14 by 23½-in., two-cycle, Alco-Sulzer engine of 1,400-hp. capacity at 277 r.p.m. appears in Fig. 57. The

FIG. 56.—Alco four-cycle Diesel with supercharger driven by the exhaust

engine consists of a bedplate and en-bloc frame. Cylinder liners are inserted in the frame, as is the standard custom.

Fuel is injected by a Sulzer by-pass pump, through a differential-needle nozzle placed in the center of the cylinder head. Each engine cylinder has its individual fuel-injection pump, which is placed in a compartment along the side of the frame.

Scavenging air is supplied to a reciprocating air pump driven from the end of the crankshaft. This air is discharged into a header which has connections to rows of air ports in each cylinder. Exhaust is through a second set of ports in the cylinder. In Fig. 57, the air pump is at the foreground end of the frame.

Buckeye Diesel.—The first oil engine built by Buckeye Machine Company was a horizontal, two-cycle, hot-bulb oil engine. This early design was abandoned, and at present the company builds a line of four-cycle, mechanical-injection Diesels (Fig. 58). This design embodies an en-bloc frame resting on a

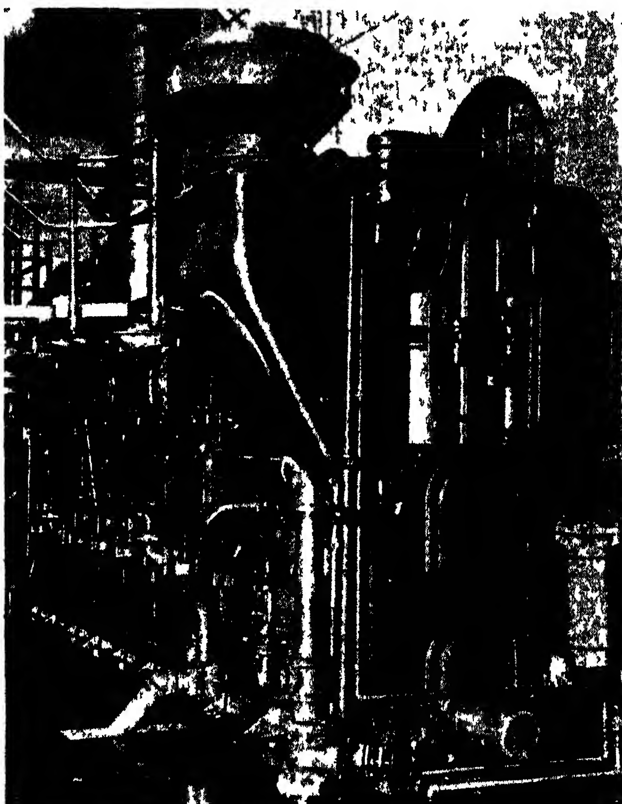


Fig. 57.—Alco two-cycle Diesel.

subbase which also carries the crankshaft in cross members. Removable liners are placed in the frame. The camshaft runs along the center of the frame side, in a compartment provided in the casting. In this space also are located the individual Bosch fuel-injection pumps. These are under control of the governor, whose weights are swung from the end of the crankshaft, with the circle of rotation vertical.

Each cylinder head carried an air, an exhaust, and differential-needle-spray valves. The fuel is injected directly into the cylinder space.

Busch-Sulzer Diesel.—Busch-Sulzer Brothers Diesel Engine Company was the first American firm to build Diesels, having procured the original Diesel American rights. In 1912 it began building Diesels after the Sulzer designs, but in 1935 Busch-Sulzer Brothers adopted the Hesselman mechanical-injection

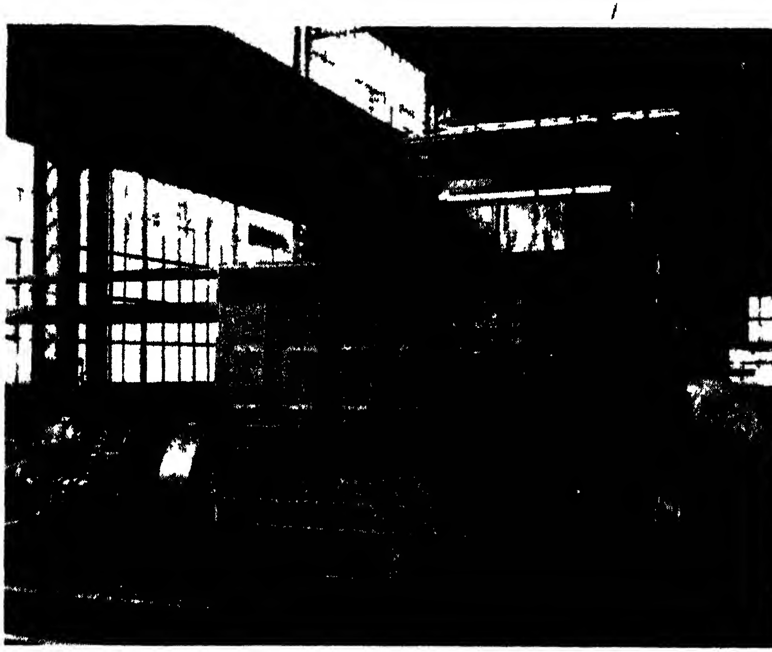


FIG. 58 —Buckeye four-cycle Diesel

combustion system and developed the largest mechanical-injection Diesel so far built in America, namely, a six-cylinder, 30 by 42-in. stroke, 3,750-hp., two-cycle unit. The company also built a 30 by 52-in., two-cycle, trunk-piston engine which, in multiple cylinders, is appropriate for powers up to 10,000 hp.

Employing the same combustion system, Busch-Sulzer Brothers constructed the world's largest locomotive Diesel of 2,000 hp.

The various sizes of Busch Sulzer-Hesselman engines incorporate a subbase, carrying the crankshaft, and an en-bloc frame

FIG. 59.—A 3,000-hp. Busch-Sulzer Bros two-cycle, mechanical-injection Diesel.

with removable cylinder liners. The several cylinder heads carry the central spray valves and air-starting valves.

The fuel pump is of the by-pass type and, placed along the side of the frame, is gear driven from the crankshaft. Scavenging air, supplied by a motor-driven blower, enters the cylinders through two rows of ports, while the exhaust gases pass out through a row of ports on the opposite side of the cylinder.

The two-cycle Diesel is available in capacities from 750 to 4,000 hp.

In addition, Busch-Sulzer builds four-cycle Diesels which also employ the Hesselman fuel system. Such engines are available in capacities from 300 to 1,000 hp. The company, however, concentrates on the larger two-cycle units.

De La Vergne Engine.—As has been mentioned heretofore, in 1917 De La Vergne brought out a horizontal, four-cycle, medium-compression, auto-ignition oil engine. This engine employed in the cylinder head a combustion chamber which accommodated both the inlet and the exhaust valves. The chamber was truncated in shape, and two spray nozzles, at opposite sides of the chamber, directed the oil spray diagonally toward the throat through which the air was forced by the piston action.

Later, the vertical design incorporating the same general arrangement was produced, and this unit is still being manufactured.

The vertical engine (Fig. 60) consists of a subbase and an en-bloc frame, although in the largest unit, 22 by 27 in., the cylinders are separate castings bolted to the top of the crankcase.

The engine has the advantage that the combustion of fuel and air takes place in the chamber, discussed in a later chapter, so the piston is not exposed to high temperatures; consequently even a 22-in.-diameter piston operates successfully without piston cooling. The chamber, however, introduces the disadvantage that by reason of wiredrawing of the intake air through the throat, the cylinder does not have a high volumetric efficiency. The horsepower developed is, then, somewhat below that of many other makes of engines.

The De La Vergne engine is provided with a by-pass type of fuel-injection pump of the company's own design. Since the spray valves are of the check-valve, or "open," type, the pump discharge pressure is low, around 1,500 lb. per square inch. The pump is chain driven from the camshaft, and in the smaller size engines the camshaft is placed along the frame, with pushrods actuating the overhead valve gear. In the 22 by 27-in. design, a

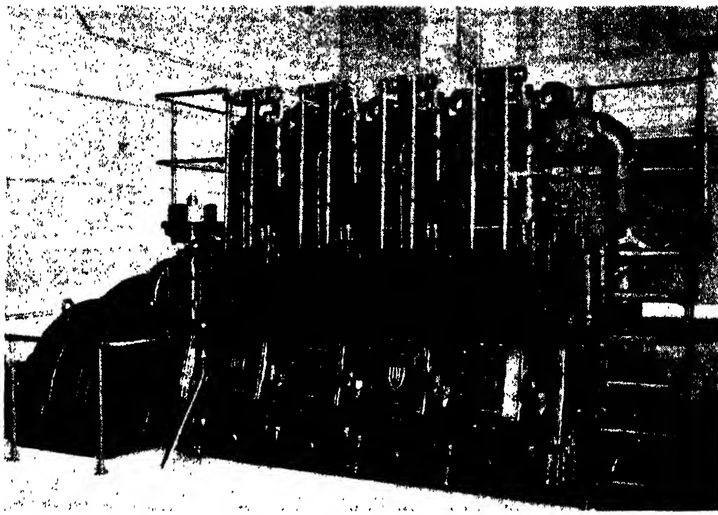


FIG. 60.—A 260-hp. De La Vergne four-cycle Diesel.

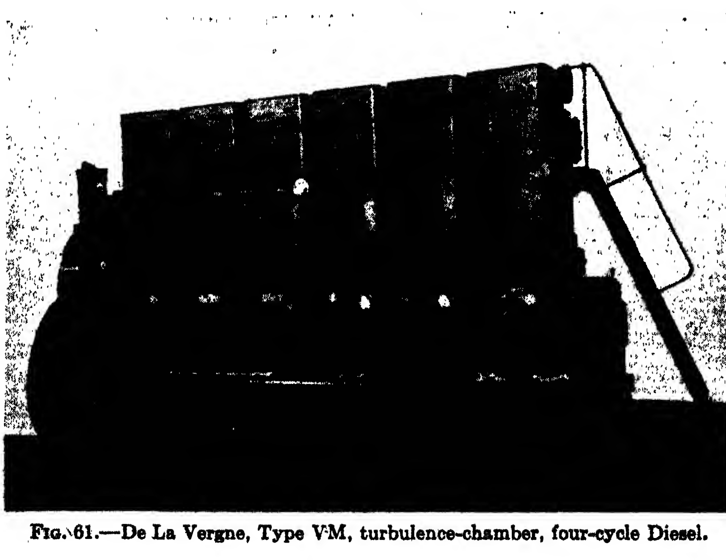


FIG. 61.—De La Vergne, Type VM, turbulence-chamber, four-cycle Diesel.

second chain drives the camshaft, which runs along the cylinder heads, with the valve rocker rollers resting on the cams.

Even at medium speeds, the wiredrawing prevents the employment of this design, so the company has developed another design (Fig. 61), employing a different combustion chamber, to be discussed in the following chapter. This new 16 by 20-in., four-cycle, trunk-piston design employs a pear-shape combustion chamber in the cylinder head into which the oil is sprayed through a single nozzle. This engine turns at 327 r.p.m., and its rating is based on 75.3 lb. brake m.e.p., although 90 lb. has been developed with clear exhaust.

Ingersoll-Rand Diesel.—Ingersoll-Rand built the Price combustion-chamber Diesel for a number of years. This system is the same as the De La Vergne already discussed, William T. Price being the chief engineer of De La Vergne at the time he patented the two-spray combustion chamber. This second builder of Price engines discovered the limitations of the design at medium speed; consequently, present-day Ingersoll-Rand Diesels employ direct fuel injection into the cylinder, with the combustion space formed between the cylinder head and the concave piston crown. The fuel distributor, a feature of early Ingersoll-Rand engines, has been superseded by individual pumps for each cylinder. This same general design has been applied to locomotives and stationary units.

The two spray nozzles have been retained in this Type *S* Diesel. These are placed on opposite points in the upper end of the cylinder, and the two sprays are directed inwardly and downward. Injection pressures are low, about 1,500 lb. per square inch.

An unusual feature of this engine (Fig. 62), considering its bore of 10 in. and stroke of 12 in., is that the crankshaft is underslung from the en-bloc frame. The camshaft is placed in a compartment cast in the frame; the overhead valve rockers are actuated by pushrods passing up through the frame. The side of this compartment is closed by a light steel cover.

Cooper-Bessemer Diesel.—The first Diesel produced by Cooper-Bessemer Corporation was a common-rail, mechanically operated unit, somewhat following the lines of the Atlas-Imperial. This engine was succeeded by the corporation's present designs, several in number, most of which employ a design of common rail with atmospheric relief and pressure-opened spray valve. This system, exclusive with Cooper-Bessemer, will be discussed later.

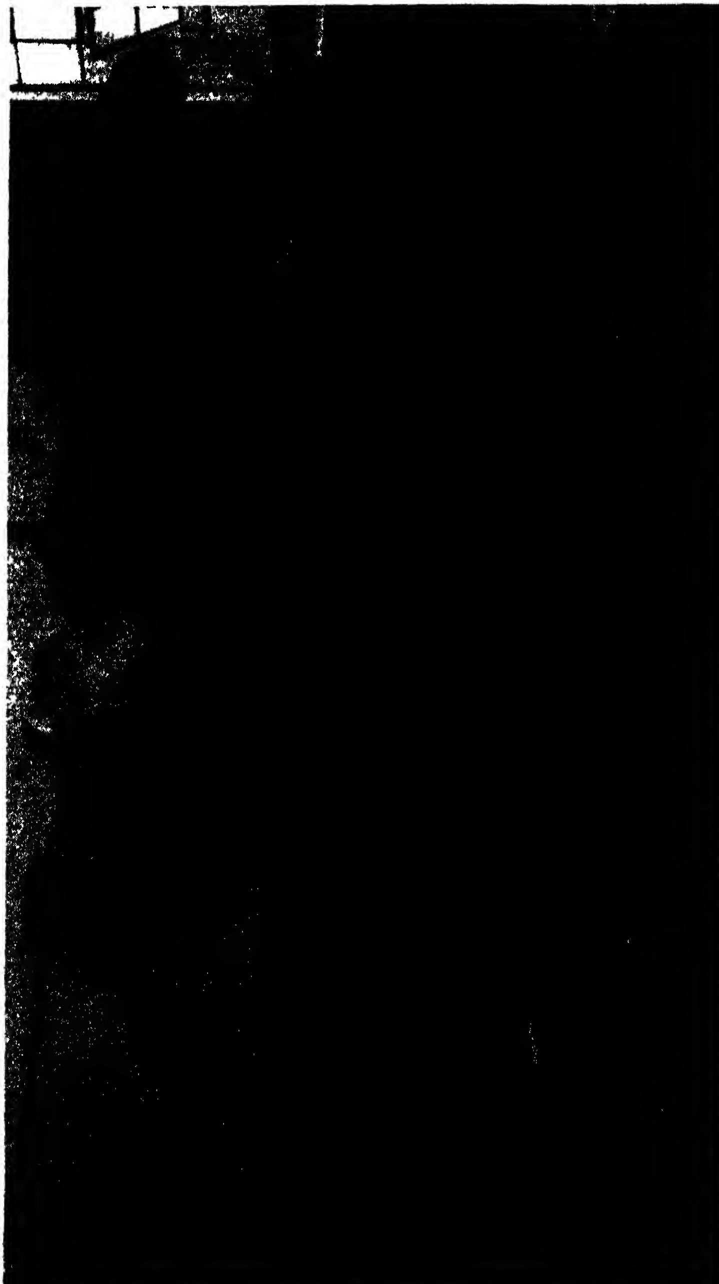


Fig. 62.—Latest design of Ingersoll-Rand four-cycle Diesel.

The structure of the larger units (Fig. 63) is made up of a subbase, carrying the crankshaft, a crankcase, and a cylinder block which contains the removable liners. The fuel-injection equipment is placed along the cylinder block, with a chain driven

FIG. 63.—Cooper-Bessemer Diesel.

from the crankshaft. A spring-loaded spray valve is placed in the center axis of each cylinder head, between the intake and the exhaust valves. The latter are operated by rockers and short pushrods from the camshaft, which runs in a trough alongside of the upper end of the cylinder block.

Fairbanks, Morse Diesel.—For heavy-duty slow and medium speeds, Fairbanks, Morse & Company has adhered to the two-cycle design.

Although retaining the principles of the two-cycle Diesel which the company had developed prior to 1930, Fairbanks, Morse has introduced many minor changes in detail. These have permitted the 14 by 17-in. unit (Fig. 64), formerly rated at but 50 hp., to develop a full-load rating of 75 hp. per cylinder at 300 r.p.m.

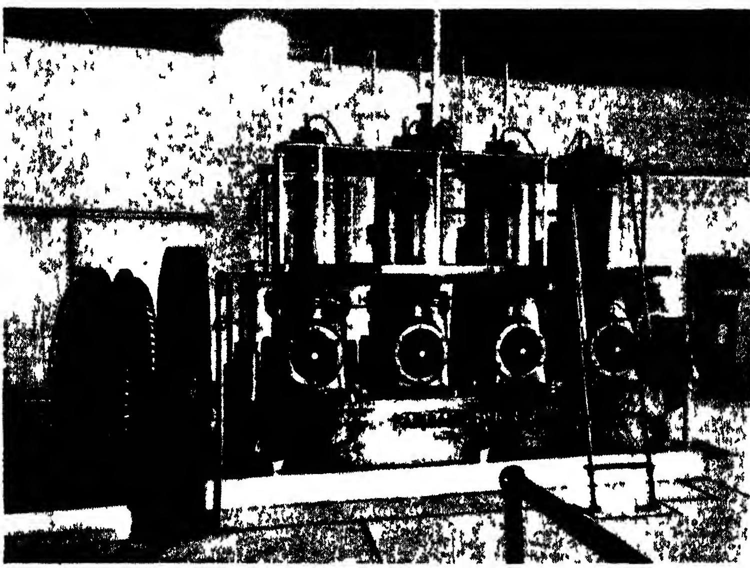


FIG. 64 —A 14 by 17-in., four-cylinder, two-cycle Fairbanks Morse Diesel

The company has also developed a 12 by 15-in., two-cycle cylinder, in sizes up to eight cylinders, fitted with pump scavenging and oil-cooled pistons. At 400 r.p.m. the cylinder rating is 100 hp.

For the largest cylinder the company builds, 16 by 20 in. (Fig. 65), the engine has been increased to eight cylinders operating at 300 r.p.m.

Fairbanks, Morse & Company has also engaged in the construction of opposed-piston, two-cycle engines, with two crankshafts geared to a common driven shaft. The smallest units, 300 hp., have been sold for commercial applications, and the

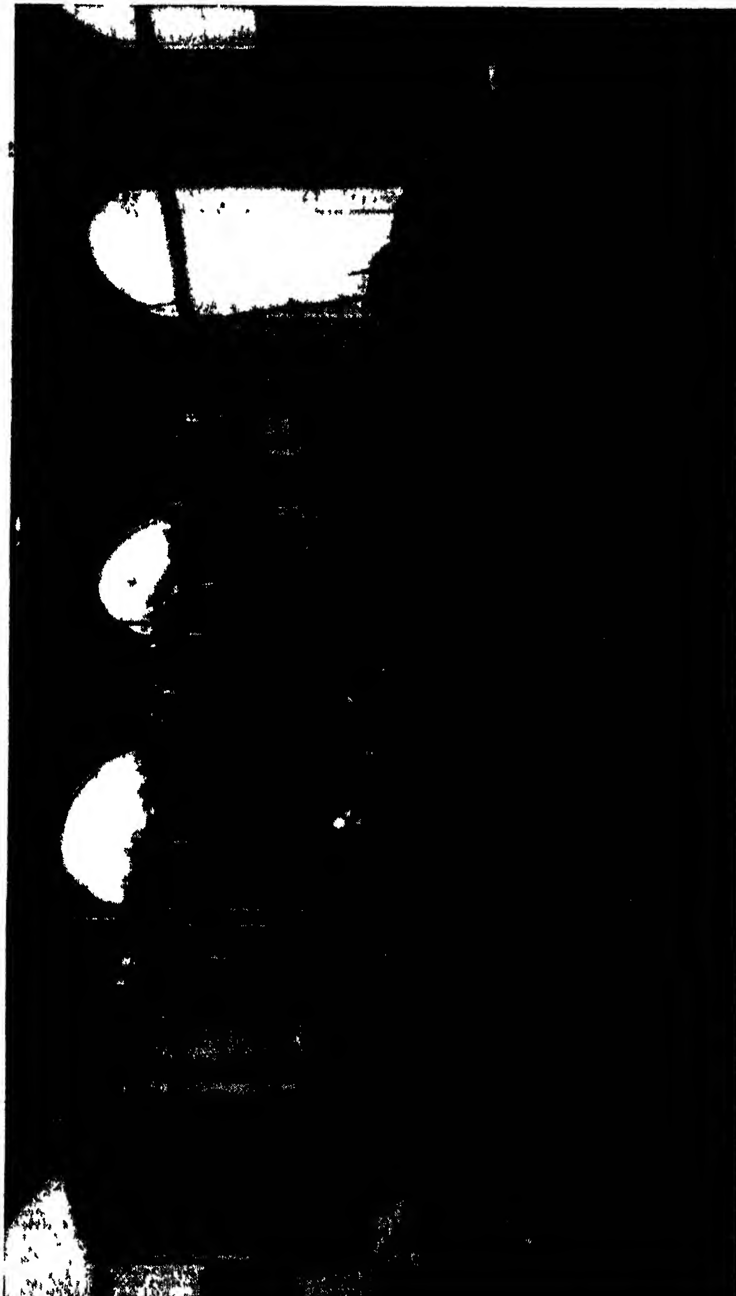


FIG. 65.—Fairbanks, Morse & Co. 16 by 20-in., two-cycle Diesel.

FIG. 66. —Nordberg mechanical-injection, two-cycle Diesel.

largest have been for the Navy, and several 750-hp. units have been for high-speed trains.

Nordberg Diesel.—At present, Nordberg builds, in addition to the air-injection Diesel heretofore mentioned, a line of two-cycle, mechanical-injection, 17 by 25-in. and 19 by 27-in. Diesels, of both the trunk-piston and crosshead-piston types. In mechanical structure these engines (Fig. 66) are almost identical with the air-injection Diesels.

The Nordberg structure has a bedplate carrying the crankshaft; upon this bed is placed vertical A-frames which in turn support the cylinder blocks. Each cylinder is cast separately, and the square bases of adjacent cylinders rest upon a common A-frame. All are tied together by long through bolts.

The Bosch fuel-injection pumps are placed along the crankcase; the spray valves are located centrally in the cylinder heads.

If the engine is provided with crossheads, the chief difference is the increase in the height of the A-frame to accommodate the crosshead guides. In both types of construction the piston crowns are cooled by lubricating oil. This oil flows up the connecting rod to the wristpin, and part reaches the piston crown. The oil flows out through telescopic tubes.

Scavenging air is furnished by a reciprocating air pump driven from the crankshaft end. The air reaches the cylinder through ports, whereas the exhaust gases exit by way of other ports. A 2,250-hp. Nordberg Diesel is shown in Fig. 66.

Fulton Diesel.—The present-day Fulton Diesel (Fig. 67) is a four-cycle, mechanical-injection unit. Structurally, the engine is made up of bedplate and en-bloc frame with removable cylinder liners.

Each cylinder head carries an intake and an exhaust valve, with the spring-loaded spray valve between these two.

The fuel-injection pump is of the by-pass type of Fulton design. It is placed along the frame, chain driven from the crankshaft.

Worthington Diesel.—Another firm to abandon entirely the use of air injection in favor of mechanical injection was Worthington Pump & Machinery Corporation which, shortly after 1930, out of all the knowledge and experience gained from the building of many types of engines and their resulting operations in different classes of service for which they were designed, developed a new line of unified and standardized engines.

FIG 67 —Fulton four-cycle Diesel

One of the strongest features of this new line is that in the design only time-tried elements are embodied; everything experimental or of a radical departure from recognized good practice has been rejected. Furthermore, a complete modernization of shop production has brought manufacture up to the latest improved and economical method of handling.

One of the Worthington eight-cylinder, 12 by 20, 1,000-hp., Type EE Diesel engines, with enclosed valve gear, appears in Fig. 68.

The unit consists, structurally, of a subbase supporting the crankshaft, a crankcase, and a cylinder block. In some of the units the camshaft is inside the crankcase, with pushrods operating the overhead camshaft. In the larger engines the camshaft is at a level with the cylinder top, with short pushrods connected to the valve rockers.

Bosch fuel-injection pumps are employed in all the types. The spray valve is in the center of the cylinder head.

Venn-Severin Diesel.—Venn-Severin Machine Company has always adhered to the two-cycle, crankcase-scavenging principle The present Venn-Severin engine (Fig. 69) employs 250-lb. compression and a combustion chamber in the cylinder head. In starting, it is necessary to heat the fuel charge by an electric coil. The fuel-injection pump and spray nozzle are of the company's design. The pump is driven by cams placed on a vertical shaft at the end of the crankshaft and gear driven from the latter.

The crankcase acts as the scavenging pump; the air reaches the cylinder ports by a transfer passage cast in the cylinder. Exhaust is through a second set of ports.

Superior Diesel.—For medium speeds and heavy duty, the National Superior Company builds a line of four-cycle engines illustrated by Fig. 70. Some of the units, depending upon size and speed, employ the common-rail fuel system, with cam-operated spray valves; in other engines direct pump injection is followed.

The engine is of the vertical, multi-cylinder, cold-starting, mechanical-injection type, working on the four-cycle principle. Two cylinder sizes are available: 12½ by 15 in. and 14½ by 18 in.

The engine is designed to develop its rated horsepower at 75 lb. per square inch of piston area, which is very conservative, providing a liberal reserve of power, up to 25 per cent, to take care of

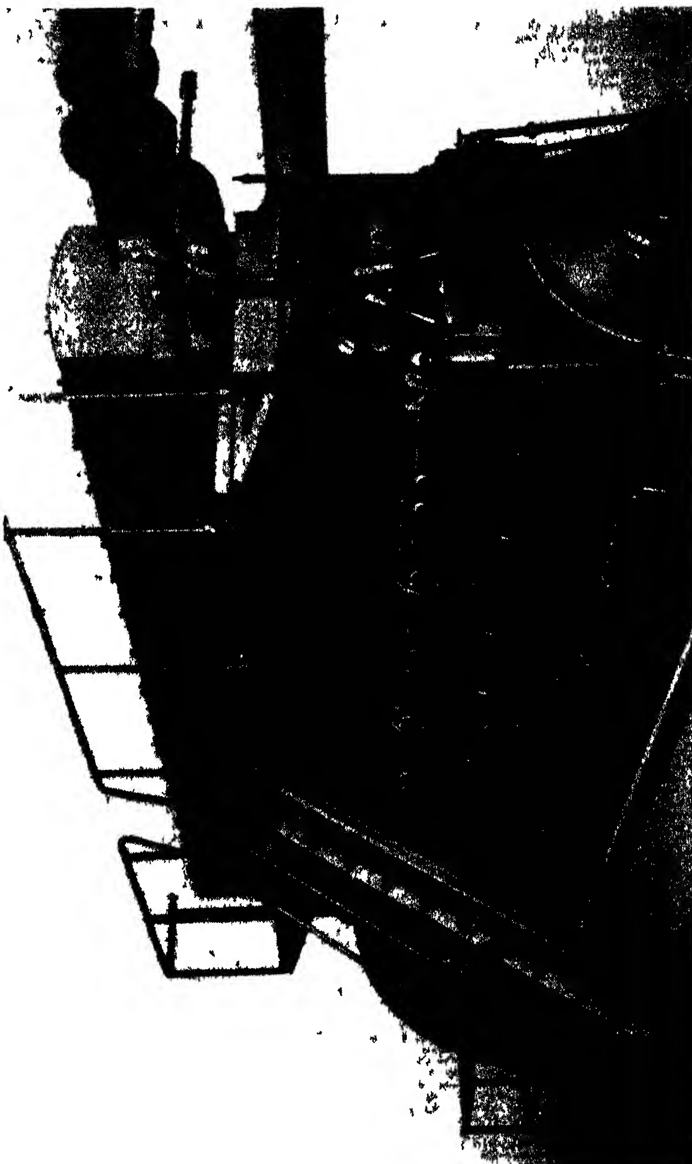


Fig 68 —Worthington four-cycle Diesel

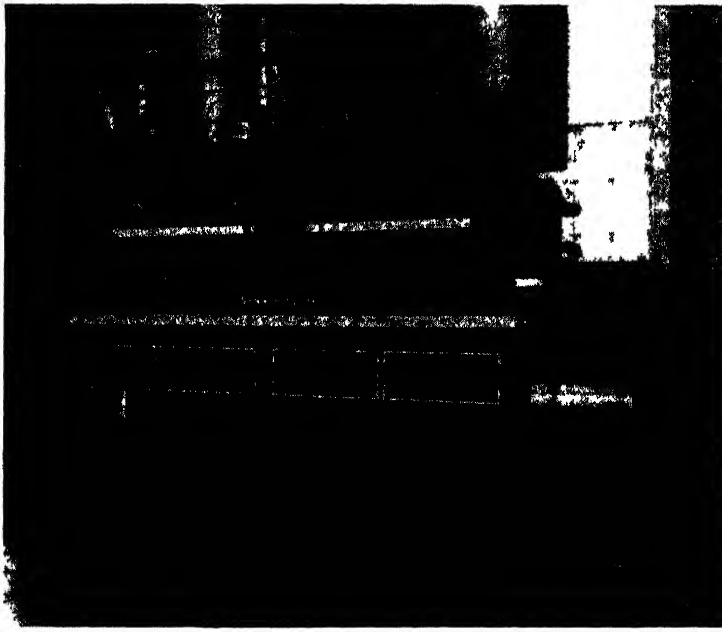


Fig. 69 — Venn-Severin two-cycle Diesel

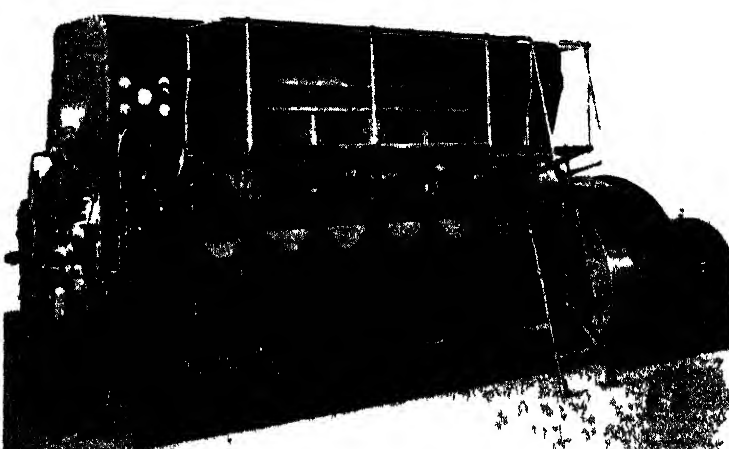


Fig. 70 —Superior medium-speed, four-cycle Diesel.

temporary overloads and emergencies. It is, however, not recommended to run an engine continuously under overload, as this may cause serious trouble, shortening the life of various engine parts and running up repair bills.

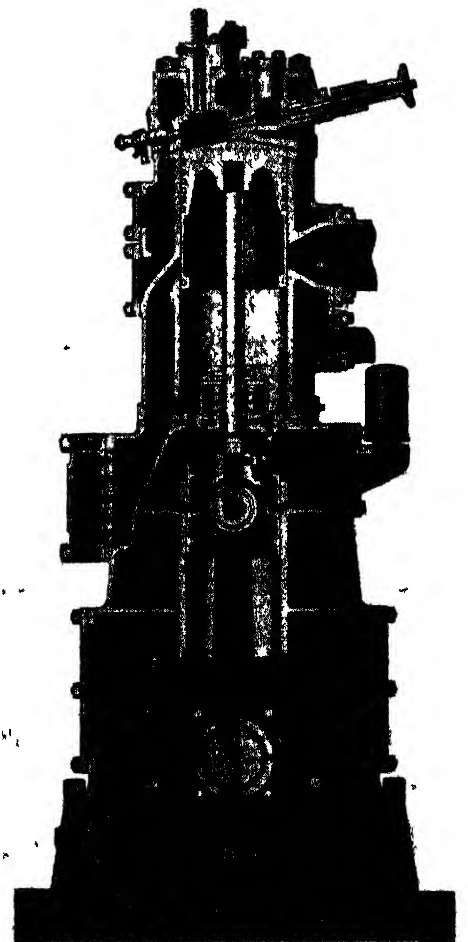


FIG. 71.—Superior vertical two-cycle Diesel.

The engine is of the trunk-piston type, with overhead, chain-driven camshaft. Dual inlet and exhaust valves, the latter water cooled, are provided. Removable cylinder liners of the wet type are used. The mechanical fuel injection is of the constant-pressure rail design. The engine is totally inclosed, equipped with force-feed lubrication, and is started cold, by compressed air of approximately 200 to 250 lb. per square inch pressure, in a few seconds. Safety devices of the alarm, or stopping, type are optional. All engine controls are centralized at floor level.

The cylinder block and frame, of one-piece, extremely rigid construction, has strong box sections with heavy transverse braces and carries the cylinder liners. All vertical strains are transmitted to the base by heavy alloy-steel through bolts, giving great strength and stiffness.

Each cylinder head carries an intake, exhaust, full-spray, and air-starting valve. The camshaft is placed at a level with the cylinder heads and is chain driven from the crankshaft.

The fuel pump is of the by-pass type with a constant plunger travel. The governor controls the timing of this by-pass, so that the amount of fuel delivered by the pump does not greatly exceed the engine's requirements. A relief valve continually by-passes part of the pump's discharge, thereby maintaining a constant common-rail oil pressure of 7,500 lb.

Superior Two-cycle Diesel.—In 1928 Superior brought out a vertical, two-cycle Diesel in which was incorporated the "dual-combustion" system the company originally had applied to its high-speed engines. A cross section appears in Fig. 71.

CHAPTER V

HIGH-SPEED DIESELS

General.—The Diesel of today is predominately a machine of high speed. For this reason due attention must be given to the various designs of this type of Diesel.

Probably the initial step should be a definition of "high speed." Basically, piston speed, in feet per minute, should be the factor defining high speed. Certainly, an engine of 24-in. stroke and turning at 350 r.p.m., giving a piston speed of 1,400 ft. per minute, is operating at high speed compared to the same engine turning at 200 r.p.m., giving a piston speed of only 800 ft. per minute. But this 24-in., 350-r.p.m. engine is not regarded as a high-speed engine when compared to a 1,400-r.p.m., 6-in.-stroke engine, although both engines have the same piston speeds.

Certain designers do classify engines by piston speed on the grounds that wear of cylinders, piston, and piston rings is proportional to rubbing speed. On the other hand, the general opinion is that wear of piston and liner depends more upon the number of reversals of travel than upon rate of rubbing speed. Certainly, rotative speeds determine inertia stresses, valve pounding, etc. Consequently, the author adopts the "revolution per minute" as the basis of speed classification.

The position is that in the chapter on slow-speed and medium-speed Diesels, there were included engines of speeds up to 600 r.p.m. In this chapter the engines are those turning from 900 r.p.m. and upward. Naturally, these high-speed engines are also lightweight as well, for those applications entailing high speeds require light weights.

Cummins Diesel.—The Diesels built by the Cummins Engine Company are based on the original conception of C. L. Cummins.

The earliest engines built by Cummins employed the Hvid combustion chamber. From this design Mr. Cummins developed an engine in which fuel was deposited in a "cup," or vaporizer, as in the Hvid, but this fuel is injected into the cylinder by position action of a displacement plunger, in place of a primary ignition of part of the fuel, as was the basis of Hvid system. The first Cummins Diesel was a single-cylinder marine auxiliary engine. In 1932 Cummins brought out a multi-cylinder truck engine which in general lines followed automotive-gas-engine practice.

One of these six-cylinder, four-cycle Diesels is shown in Fig. 72. The frame is of the en-bloc type, with an underslung crankshaft, and is carried up to form the cylinder jacket in which are placed the removable cylinder liners.

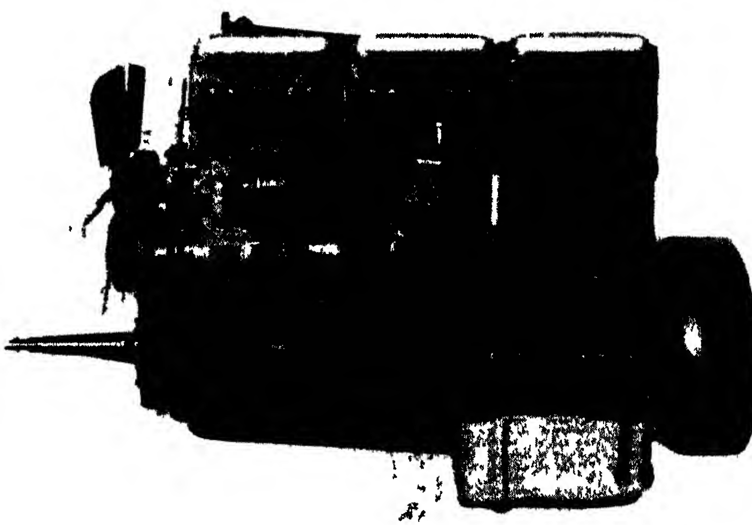


Fig. 72.—Cummins Diesel for trucks.

The underslung crankshaft is supported in precision-type bearings. All bearings, including crankpin and pistonpin bearings, are pressure lubricated. All auxiliaries, such as charging generator, water pump, lubricating-oil pump, and the fuel system, are built into the engine.

The fuel-handling system is assembled in a case at the side of the engine and includes transfer pump, filter, measuring pump, and distributor. As will be discussed later, the Cummins Diesel employs a single fuel-injection pump which delivers fuel in proper sequence to the vaporizers of the several engine cylinders through a rotary distributor. The vaporizer receiving this fuel charge contains a plunger which, when forced downward by a rocker and cam, sprays the oil charge into the engine cylinder.

Other than its truck units, Cummins builds larger Diesels of the six-in-line and V-8 types for locomotive, marine, and stationary applications.

Buda.—Originally Buda Company embarked on the construction of high-speed Diesels with a license to use the M.A.N. double-nozzle, direct-injection system, later the company

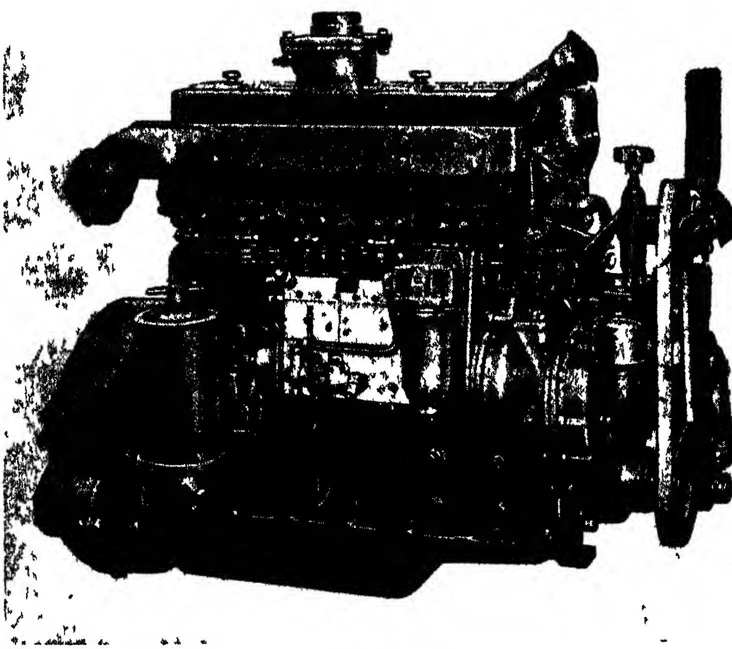


FIG. 73.—Buda-Lanova six-cylinder Diesel.

adopted the M.A.N. auxiliary air-cell system. At present Buda Diesels (Fig. 73) are fitted with Lanova combustion systems, which include an energy cell, to be discussed later.

The Buda frame is of the en-bloc type, with liners pressed into the bored recesses. Crankshafts are underslung, with all bearings on the engine pressure lubricated. The several auxiliaries are driven through gear trains from the crankshaft.

Waukesha-Hesselman.—Although the Waukesha-Hesselman oil engine is not a true Diesel, or compression-ignition engine, it does use the same cheap oil, injected by a pump as in the Diesel;

compresses pure air only as with the Diesel; and is competitive with the Diesel. These factors justify inclusion of this engine in this volume.

In the Waukesha-Hesselman engine (Fig. 75) air is drawn in during the suction stroke, and on the next piston stroke this air is compressed to about 150 lb pressure. Oil is injected into this air charge before the end of the compression stroke. Since 150 lb.

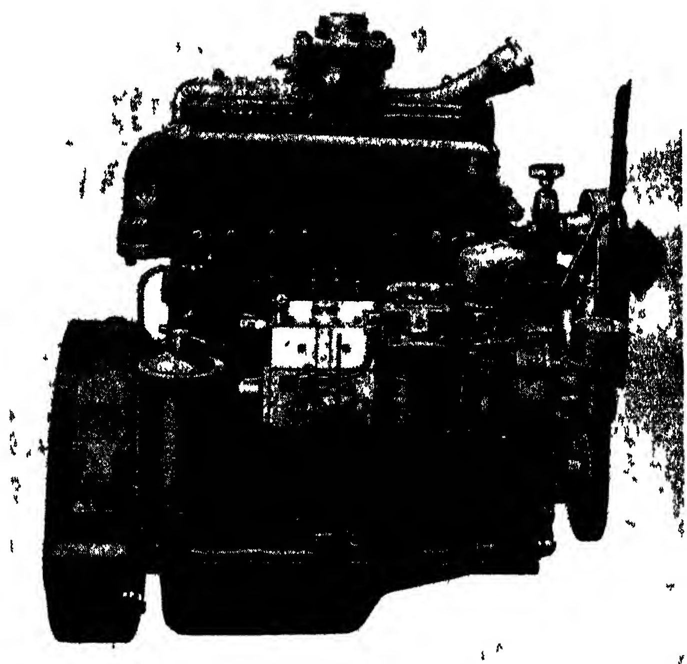


FIG. 74.—Buda-Lanova Diesel for Ford trucks.

compression does not give a self-ignition temperature to the air, a spark plug is added to ignite the fuel-air mixture. Combustion ensues, followed by expansion of the gases on the power stroke and evacuation of these burnt gases on the fourth, or exhaust, stroke of this four-cycle engine. The combustion system will be covered more completely in a later chapter.

The engine frame follows automotive gas-engine lines, including underslung crankshaft and en-bloc frame. Cylinder liners, however, are removable.

For fuel injection, the Bosch and Hesselman pump are standard. All bearings are pressure lubricated, a system that also includes the valve gearing. The camshaft is placed in the upper section of the crankshaft, with pushrods passing up through the frame, to operate the rockers of the valves, which are in the cylinder head. The spray valve is placed in the head at one angle so that the fuel spray sweeps down into the hollowed piston crown and then across to the spark plug, which is at the opposite side

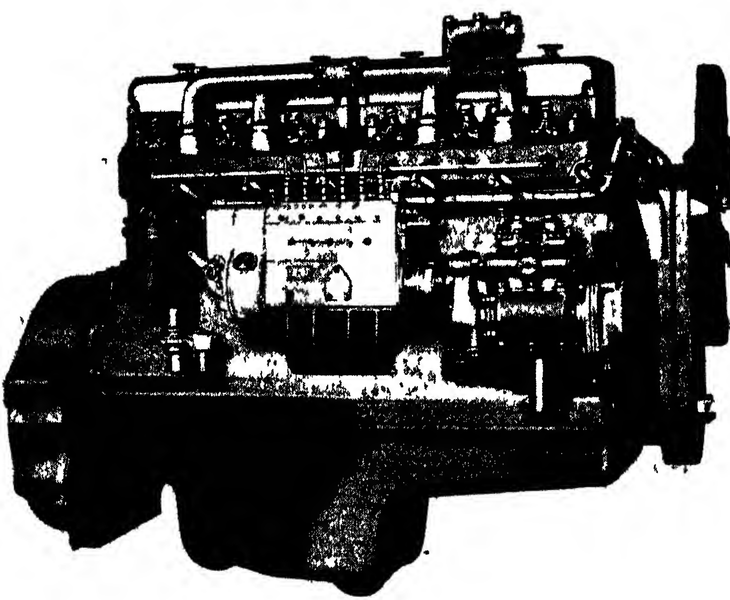


Fig. 75.—Waukesha-Hesselman spark-ignition oil engine.

of the head in an inclined position. To give the spark, the usual automotive distributor, coil, and battery are placed on the engine. The battery, large enough to operate the starting motor, is charged by a generator, a part of the standard equipment.

Caterpillar Diesel.—A considerable portion of the tractors built by Caterpillar Tractor Company used to be sold abroad, in India, Australia, Egypt. These were powered with gas engines and faced the competition of German and English Diesel-powered tractors. The situation prompted Caterpillar to initiate a program of research into Diesel-engine types, with the view to adopting the most favorable combustion system. After extensive tests on almost all existing designs, Caterpillar developed its own four-cycle, precombustion-chamber engine; the first such engine was finished in 1931.

Each year Caterpillar has extended the application of this Diesel until at present Diesel tractors represent about 95 per cent of the Caterpillar tractor output. The eight sizes ranging from 160 to 35 hp. are also used by 102 manufacturers of shovels,

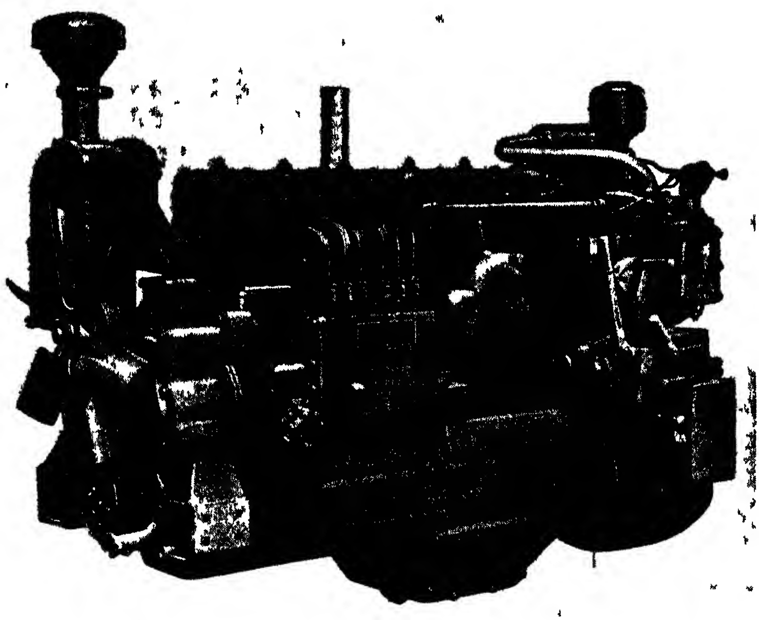


FIG. 76.—A Caterpillar six-cylinder, precombustion-chamber Diesel.

locomotives, crushers, etc., and thousands have been installed for stationary power purposes.

These engines normally turn at 850 r.p.m. for tractors, but 900 is a standard speed for other applications. The engine (Fig. 76), then, is at the low end of the high-speed classification. Consequently, its weight per horsepower is considerably more than the weight of the typical high-speed Diesel.

The frame is en-bloc, with underslung crankshaft and removable cylinder liners. The crankshaft is in the crankcase, with valve pushrods placed outside the frame but enclosed in tubes.

The fuel pump is built by Caterpillar and follows the general lines of the Bosch pump; it will be discussed later.

All parts are pressure lubricated. The piston is, as a rule, of cast iron, but recent units have been equipped with aluminum pistons.

To start this Diesel, a small two-cylinder gasoline engine mounted on the Diesel is hand cranked. Its shaft then engages, through a Bendix clutch, a gear on the Diesel flywheel. The Diesel is thus turned over several times, until compression of the air raises the air temperature to the ignition point of the fuel oil.

Hercules Diesel.—Hercules Motors Corporation was one of the first to realize and appreciate the tremendous market that awaited the development of high-speed, light-weight Diesels.

In 1930 actual work, under the direction of O. D. Trieber, was under way, and, as a result, a single-cylinder test engine was built for experimental purposes. The Hercules combustion chamber was developed in this engine. Late in 1932 the first multi-cylinder engine was built. This was the DHXB, six-cylinder Hercules engine with a 5-in. bore and a 6-in. stroke. Except for minor refinements which were incorporated into the engine as newer and better materials were made available, the present DHXB is the same as the first engine of this series. No fundamental changes in the Hercules combustion method have been made since the original, single-cylinder engine was built in 1930. Other cylinder bores and strokes were incorporated in other engine sizes.

In the development of its complete line of high-speed, heavy-duty Diesel engines the Hercules Motors Corporation has not been restricted to the use of these engines in one type of application or confined by their use for only one purpose. Hercules designed these modern Diesel engines for a vast variety of applications where high-speed gasoline engines had previously been the only available prime mover.

The engine (Fig. 77) has an en-bloc frame, underslung crankshaft, removable cylinder liners, and pressure lubrication. Both Bosch and Ex-Cell-O fuel-injection pumps are used.

These engines are used by leading American truck and bus manufacturers and by leading manufacturers of agricultural, oil-field, general industrial, and marine equipment.

European interest has been exceptionally high, many European manufacturers coming across the ocean for the first time for their Diesel-engine source and selecting Hercules.

McCormick-Deering Diesel.— Demand of tractor owners for a cheap-fuel engine was the incentive that prompted International Harvester Company to develop its McCormick-Deering precombustion Diesel (Fig 78) The first of these was put out in 1933; this was a four-cylinder, 4¾ by 6½-in , four-cycle

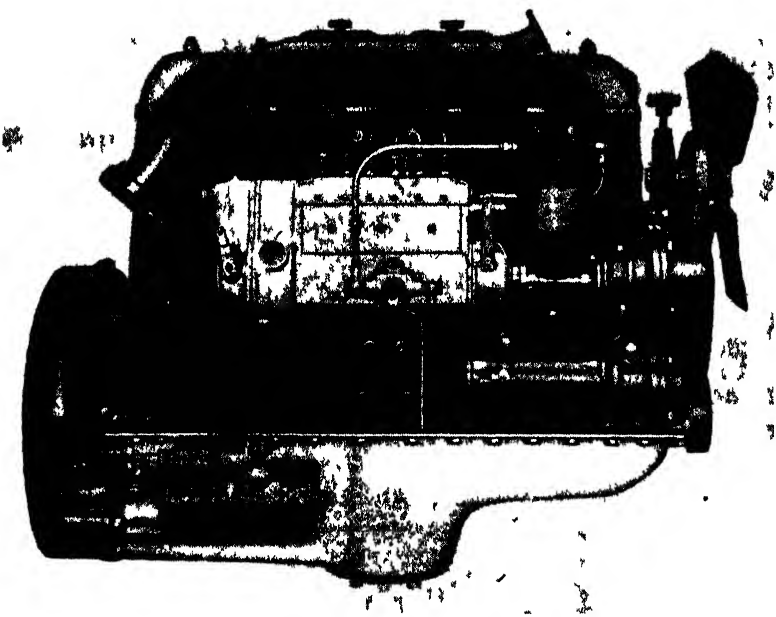


Fig 77 —Hercules six-cylinder, four-cycle Diesel

engine Later, 4¾ by 6½-in , six-cylinder and 4½ by 6½-in., four-cylinder units were developed

These engines are started by gasoline, which is introduced into the engine cylinder by means of a carburetor and fired by a spark. The necessary lower compression is brought about by the opening of a valve which places an additional clearance volume in connection with the cylinder.

The frame is similar to the gasoline en-bloc design. The camshaft in the crankcase actuates pushrods which act upon the overhead valve rocker arms. All parts are pressure lubricated.

The fuel-injection pump of the PD 80 is of McCormick-Deering design whereas the PD 40 unit has a Bosch fuel-injection pump, to be discussed later.

Murphy Diesel.—Another Diesel which is started by exploding gasoline in an enlarged volume of the combustion space is the Murphy Diesel (Fig. 79).

The frame is en-bloc, with underslung crankshaft and removable liners. All the cylinder heads of the unit are embodied in

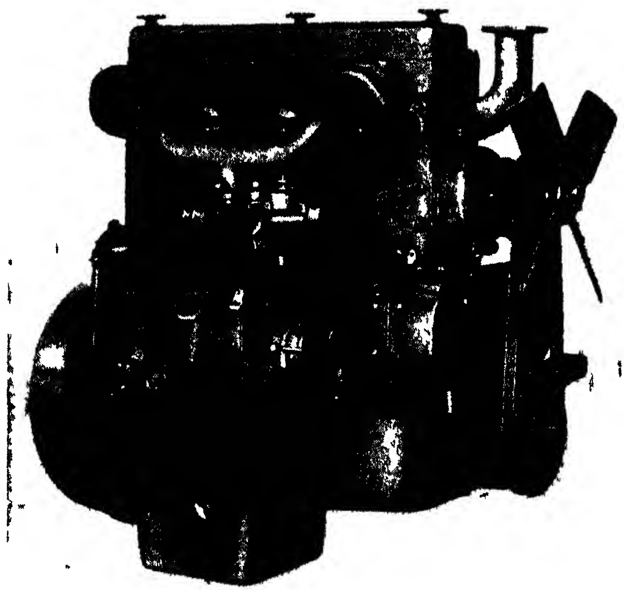


Fig. 78.—McCormick-Deering Diesel, showing the gasoline carburetor used in starting

a single casting which is provided with links, so that it may be swung away from the frame with little effort.

The fuel pump and spray nozzle are combined into a single unit, at the cylinder head. The plunger is operated by two cams, one of which is governor controlled, to regulate the plunger travel.

Fairbanks, Morse Diesel.—For years Fairbanks, Morse & Company built a line of two-cycle Diesels and was regarded, and still may be regarded, as the largest builder of heavy-duty Diesels. For many installations the costly, slow-speed, heavy-

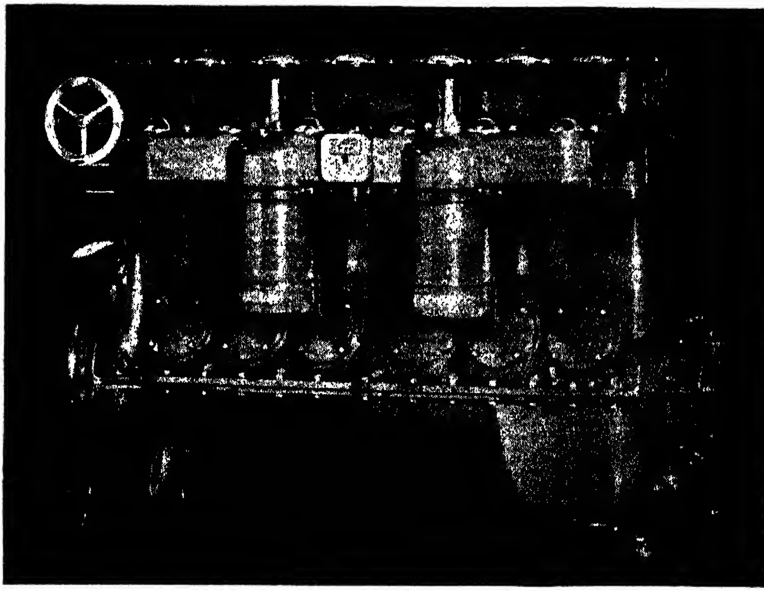


FIG. 79.—Murphy four-cycle industrial Diesel.

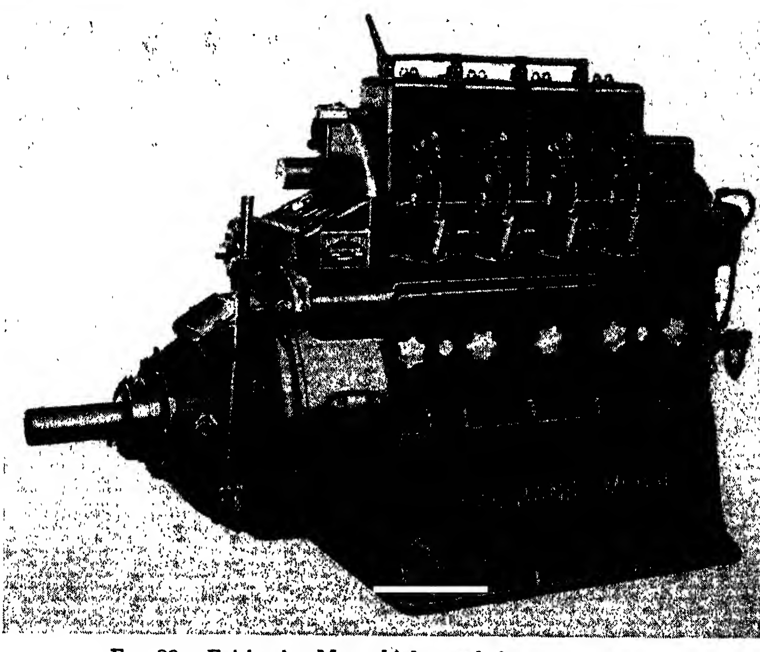


FIG. 80.—Fairbanks, Morse high-speed, four-cycle Diesel.

duty Diesel was unacceptable; consequently the company developed three sizes of four-cycle, high-speed Diesels (Fig. 80). In cylinder sizes these are 4½-in. bore by 6-in. stroke, 5½-in. bore by 7-in. stroke, and 8½-in. bore by 10-in. stroke.

For these units a turbulence-chamber combustion system was chosen. Fuel injection is carried out by Bosch pumps, through Bosch spray valves.

Unlike most high-speed designs, these engines have a subbase carrying the crankshaft, with the en-bloc frame bolted to this base.

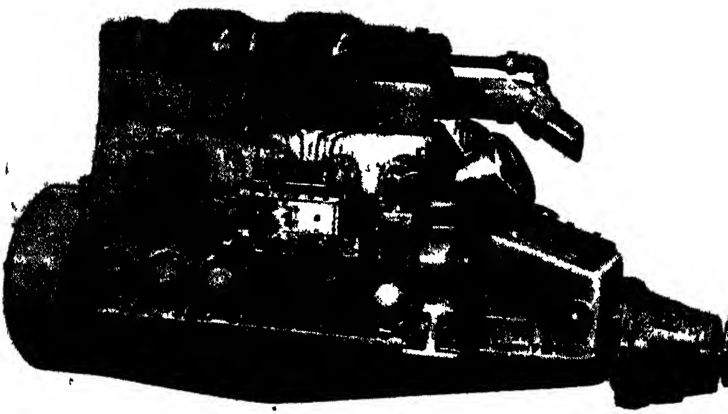


Fig. 81.—National-Superior high-speed, four-cycle marine Diesel.

Cylinder liners are removable. Valves are in the cylinder head, operated by rockers and pushrods from the camshaft placed in the crankcase. All parts are pressure lubricated.

The standard speed is 1,200 r.p.m. for the two smaller units and 900 r.p.m. for the 8½ by 10-in. engines.

National-Superior Diesels.—National Supply Company has developed a four-cycle, high-speed Diesel of 12 hp. per cylinder rating, at 1,200 r.p.m. Top speed is 1,600 r.p.m.

This engine (Fig. 81), has a "dual-combustion" system, in which the main oil charge is burned in the cylinder, and a small part of the charge enters an air cell, where part ignites and, flowing out into the cylinder, serves to give a final turbulence to the main fuel-air mixture, thereby completing combustion.

The frame, of the en-bloc type, supports the underslung crankshaft and carries removable cylinder liners. All moving parts are pressure lubricated, the lubricating system includes an oil cooler and a filter

Atlas-Imperial High-speed Diesel.—After centering its attention for years upon medium-speed, heavy-duty Diesels, Atlas-Imperial Diesel Engine Company developed a four-cycle, high-speed unit (Fig 82) with a fuel-injection system that embraces a single pressure pump, a governor-controlled "interruptor," and a rotary distributor

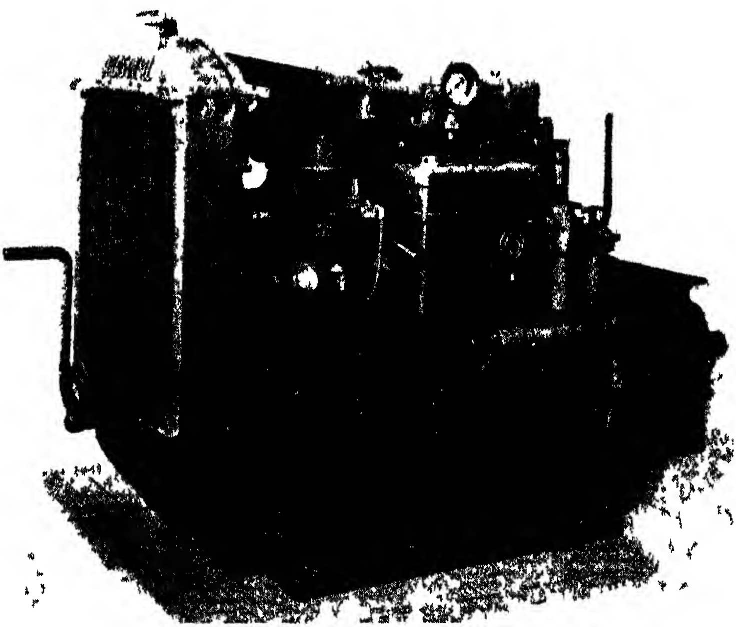


Fig 82 —Atlas-Imperial high-speed Diesel

The engine, when intended for power-plant application, is provided with a subbase The en-bloc frame carries the underslung crankshaft and the removable cylinder liners Valves, including the spray valve, are located vertically in the cylinder heads, with the camshaft in the crankcase operating the intake and exhaust valves through pushrods, which pass up through the frame

All parts are pressure lubricated In Fig 82 the box on the side of the frame contains the entire fuel-handling apparatus, to be discussed in a later chapter.

Allis-Chalmers Oil Engine.—For its tractors, Allis-Chalmers Manufacturing Company adopted a spark-ignition, pump-injection, four-cycle oil engine of 150 lb compression One of these units is shown in Fig 83 In principle the engine follows

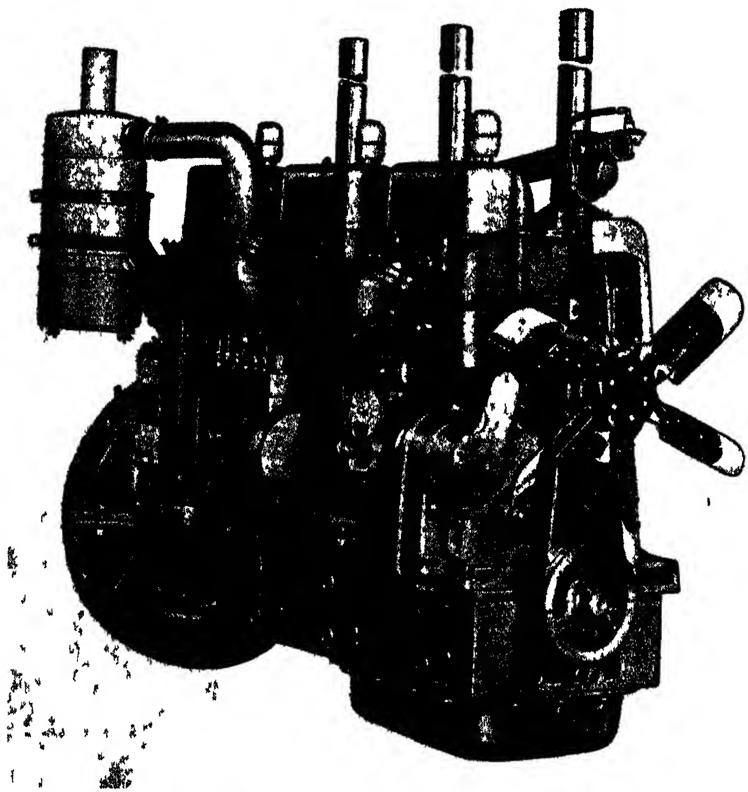


Fig 83 —Allis-Chalmers spark-ignition oil engine

the Hesselman-engine action, although there is supposed to be some difference in respect to the fuel-air mixture at the spark plug.

The engine frame is the same as the A-C gasoline tractor engine, with underslung crankshaft and pressure lubrication of the rotating and reciprocating parts.

The fuel-injection pump is the Deco, discussed later. The spray valve is of the differential-needle type.

Starting is accomplished by injecting a spray of gasoline into the air-intake manifold, so that the engine draws in a mixture of air and gasoline vapor.

Atlas-Thornburg Diesel.—The Atlas-Thornburg Diesel Engine Company has developed a new line of small high-speed Diesels employing the Lanova combustion system; Fig. 83 *A* shows a

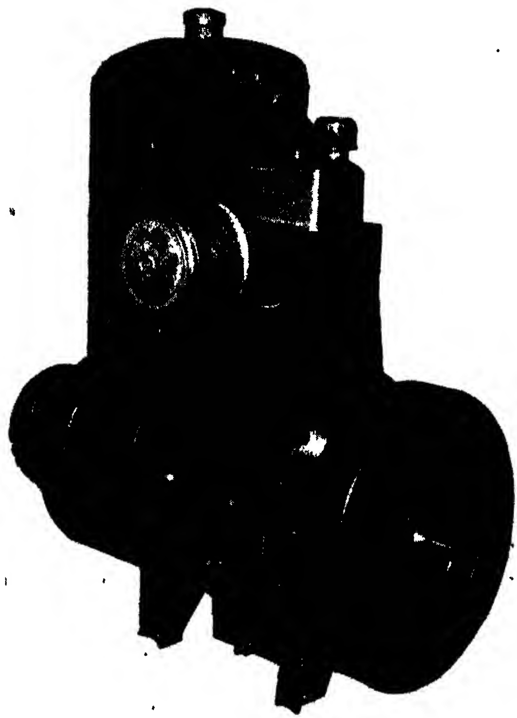


FIG. 83 *A*. —Atlas-Thornburg high-speed Diesel.

single-cylinder unit. All models have a $3\frac{1}{8}$-in. bore and $3\frac{3}{4}$-in. stroke, developing 5 hp. per cylinder. With one to four vertical-in-line cylinders, the engines range in size from 5 to 20 hp. Industrial engines are rated at 1,800 r.p.m., and marine engines are rated at variable speeds up to 2,000 r.p.m. Diesel light plants ranging from 3- to 15-kw. capacity are also available.

Fuel is injected into the Lanova combustion chamber by a Bosch pump through single-orifice, non-clogging, pintle-type, low-pressure nozzles. A centrifugal-type governor, contained in the end of the camshaft gear, controls the position of the fuel-injection pump helix.

Cylinder block and crankcase are cast integral of nickel-chromium, electric-furnace cast iron, as are also the cylinder heads. Valves of special heat-resisting, nickel-chromium alloy are in the cylinder heads Rocker arms are steel drop forgings

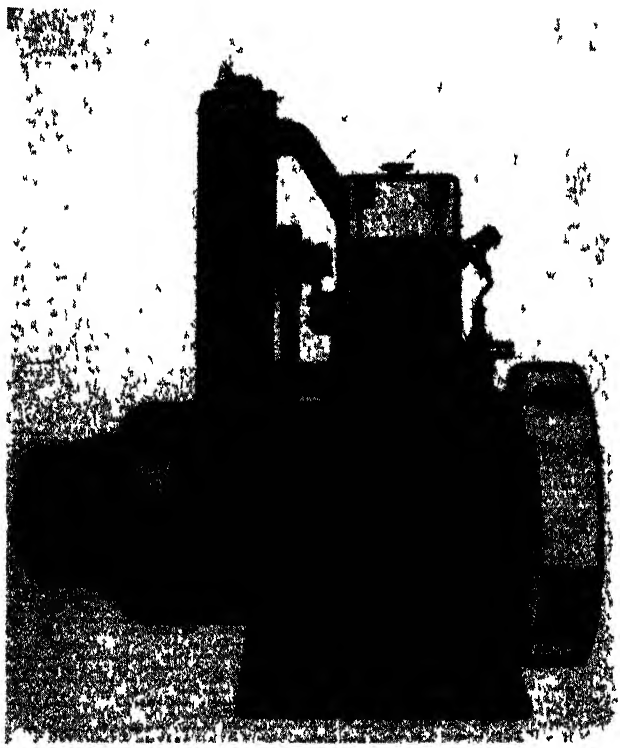


FIG 84 —Witte single-cylinder, high-speed Diesel.

and operate on steel-backed, babbitt-lined bushings. An aluminum cylinder cover incloses the valve mechanism and is easily removable.

Pistons are aluminum, with three compression rings and one oil ring. The crankshaft, of special electric-furnace cast iron of high tensile strength, is drilled for pressure lubrication of main- and connecting-rod bearings. Main bearings, 2 in. in diameter, are replaceable, steel-backed, babbitt-lined, precision type.

The engine is designed for cold hand starting, but electric starting is supplied, using a 12-volt Bendix-starter drive.

Witte Diesel.—Although European builders have been marketing miniature Diesels for years, American manufacturers hesitated about designing them. Finally Witte Engine Company ventured to redesign its 6-hp., four-cycle, single-cylinder gasoline engine,

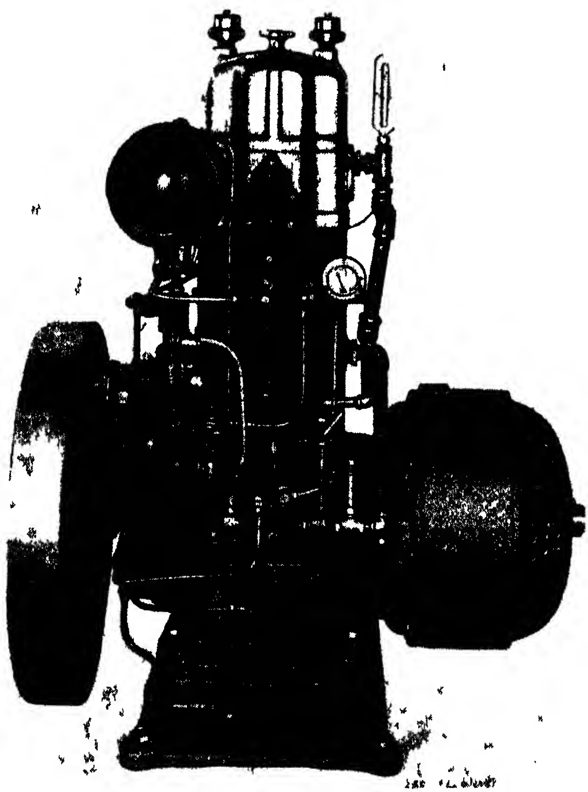


FIG. 84 A.—Stover single-cylinder Diesel

converting it into a 5-hp. Diesel. This horizontal engine has its frame and cylinder cast in one piece. The crankshaft is supported by roller bearings.

Fuel is injected by a Bosch pump, through a differential-needle valve, directly into the cylinder space. The piston crown is provided with an air cell, of tube shape. This insures complete combustion. In 1938 Witte brought out its vertical Diesel, shown in Fig. 84. In this design the frame and cylinder is a one-piece casting. A removable liner is held in by the cylinder head.

A precombustion chamber was adopted in order to eliminate combustion knock. This chamber is placed inclined in the cylinder head and is a forging separate from the head casting.

Lubricating oil is supplied under pressure to the bearings by a gear pump. The oil supply is carried in a tank in the subbase.

Stover Diesel.—Another firm building a small vertical Diesel is Stover Engine & Manufacturing Company. This engine is

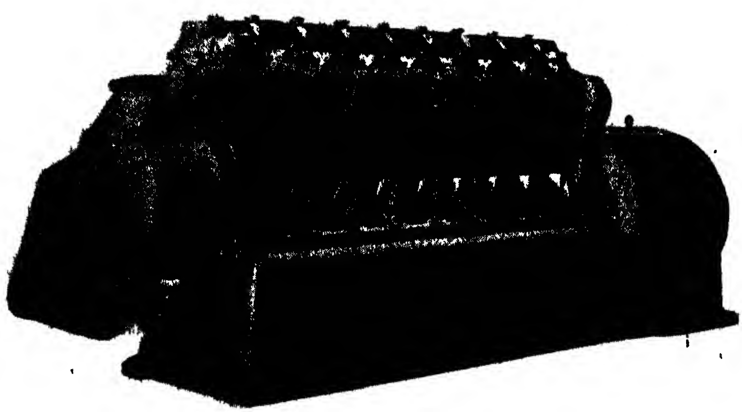


FIG. 85.—General Motors two-cycle locomotive Diesel.

shown in Fig. 84 *A*. The combustion system includes an Acro air cell in the piston crown. The engine turns at 900 to 1,000 r.p.m.

General Motors Diesel.—The first Diesel developed by General Motors was initiated by Winton Diesel Engine Company long before the latter firm was purchased by General Motors.

Winton's first high-speed Diesel was a two-cycle Diesel of 8-in. bore and 10-in. stroke. Two 720-r.p.m., 600-hp. units of this design (Fig. 85) were shown at the Chicago World's Fair in 1933.

The distinguishing features of the design were threefold: (1) The frame was welded steel plate; (2) the air entered through ports, and the gases exhausted through valves in the head; and (3) the fuel pump and spray valve were combined into a single element.

Immediately after the Fair the Chicago, Burlington & Quincy Railroad became interested in Diesels for high-speed railroad trains. The first was so satisfactory that the C. B. & Q., Union Pacific, and other roads purchased Winton-powered railroad trains up to 15 cars in length.

The Santa Fe started with a Winton-powered locomotive of 1,800-hp. capacity, followed by two 5,400-hp. locomotives pulling standard coaches and sleepers.

Early in 1939 General Motors announced the advent of a small high-speed engine following the Winton general lines.

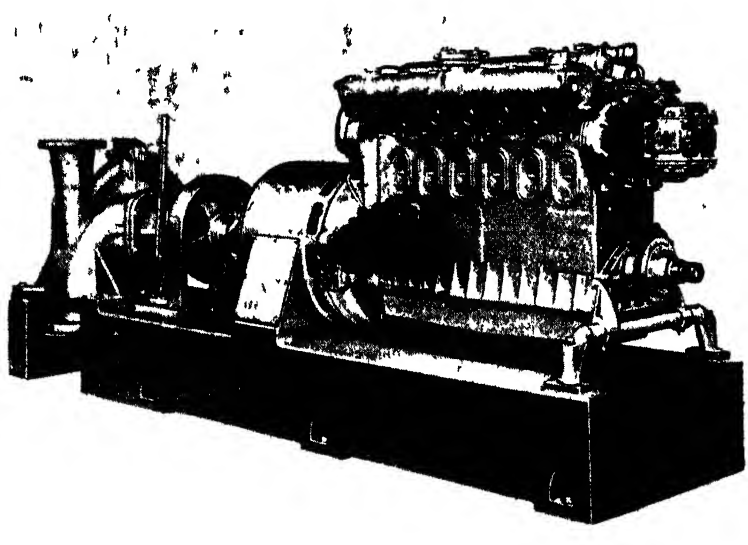


FIG 85 *A* Small General Motors two-cycle Diesel.

These are rated at 22 hp. per cylinder, and the builder will not only apply these to trucks but also sell them completely equipped as "packaged power."

General Motors Small Diesel.—The 22-hp.-per-cylinder General Motors referred to is shown in Fig. 85 *A*. The frame is a one-piece casting. Separate cylinder liners are carried by this frame, which carries, on its undersurface, bearings for the underslung crankshafts. As with the large General Motors engine, scavenging air, supplied by a lobed blower, enters the cylinders through ports, while the exhaust gases pass out through cam-operated valves in the cylinder head. The fuel injector, combining pump and nozzle, is placed in the cylinder-head center; it will be discussed in a later chapter.

Already about 400 of these engines have been installed in urban buses

Mack-Lanova Diesel.—Mack-International Motor Truck Company, Inc, has entered the Diesel field with a four-cycle, six-cylinder, $4\frac{3}{8}$ by $5\frac{3}{4}$-in. engine developing 131 hp. at 2,000 r.p.m. with 110-lb. brake m.e.p. The new Diesel (Fig. 86)

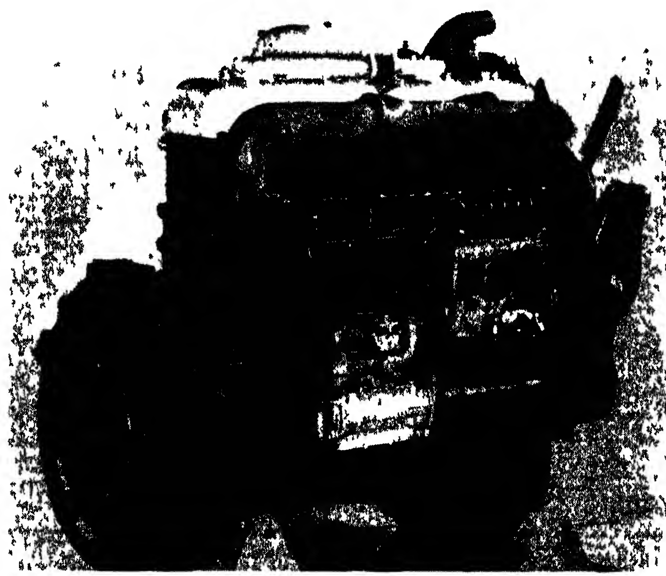


Fig. 86.—Mack-Lanova six-cylinder, four-cycle truck Diesel

employs the Lanova energy-cell, "controlled-ignition" combustion system. Fuel is supplied to the combustion chambers by new flange-mounted Bosch injection pumps driven by a self-aligning coupling. A special Pierce governor on the same shaft prevents overspeed. Fuel flows first through a Purolator filter, then to a transfer pump and two filters between the transfer pump and injection pumps.

Cylinder block and crankcase are cast integrally of high nickel-chromium semisteel and heat treated in electric continuous furnaces. Dry-type cylinder liners of high nickel-chromium semisteel, and honed to a high finish, are inserted in the cylinder block. Cylinder heads are cast in two interchangeable blocks.

Valves in the cylinder head are operated by rocker arms and tubular ball-ended push rods. Exhaust valves seat in Permafit inserts of niferrite, copper plated and faced with stellite.

The camshaft runs on seven bearings and is driven by helical timing gears of which there are four, driving an accessory shaft and the fuel-pump shaft.

The cast aluminum-alloy pistons are without internal ribs. Piston pins are full floating and run in shell bushings in the con-

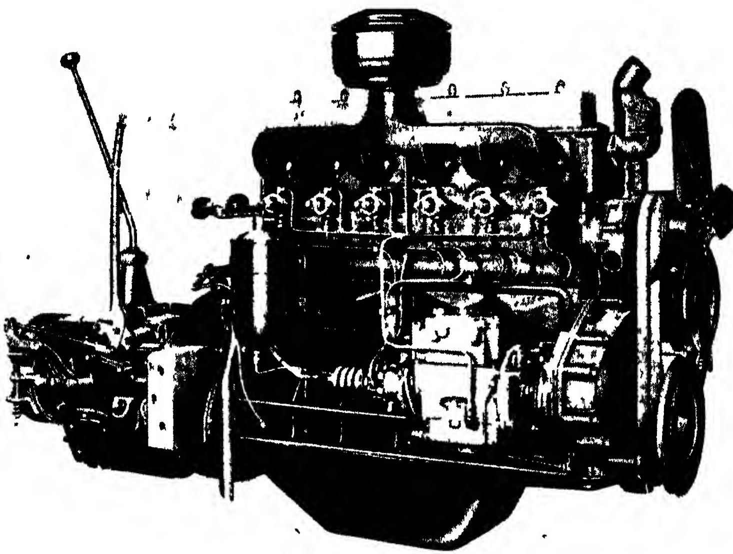


FIG 86 *A* —Dodge-Lanova truck Diesel.

necting rods and directly in the piston bosses. To lubricate the pin, three oil holes collect surplus oil from the cylinder walls.

Full-pressure lubrication is provided for main, connecting rod, and camshaft bearings. The main oil-distribution gallery is bored in the cylinder block adjacent to the full-length water jacket so that oil is directly water-cooled before entering the bearings. A by-pass W.G.B. lubricating oil filter is built so that passages to and from the filter register with drilled connections in the crankcase, thus eliminating external piping.

Dodge-truck Diesel.—The new Dodge-truck Diesel engine is a six-cylinder, four-cycle, 3¾ by 5-in. Diesel developing 95 hp. maximum at 2,600 r.p.m. (Fig. 86 *A*).

The Lanova combustion chamber, of double-0 shape with a double-cell energy chamber, is formed in the cylinder heads. Nozzles of the familiar pintle type extend through the side of the cylinder heads opposite the energy cell and placed so the fuel spray is directed at the mouth of the energy cell.

Fuel is supplied to the injection nozzles by an Ex-Cell-O fuel-injection pump in which is incorporated automatic control of injection timing in accordance with engine speed and also speed-governed regulation of maximum fuel quantity. Fuel oil is filtered by duplex, edge-type Purolators and a combination bag and edge-type Purolator.

Cylinders are cast en bloc of nickel-molybdenum alloy. The design embodies full-length water jackets; chain-driven camshafts and mounting dimensions are identical with the Dodge gasoline engine.

Tin-plated pistons are employed, permitting close piston fits. Three compression and two oil rings are used, the upper rings being ⅝ in. below the top of the piston. Rings are tin-plated to prevent scuffing during "wear-in."

CHAPTER VI

COMBUSTION OF FUELS

General.—In spite of the fact that petroleum oils have been on the market for 70 years and the molecular structure is not difficult of analysis, it must be confessed that the behavior of an oil within the cylinder of a Diesel cannot be determined with any exactitude on the basis of its known characteristics. In fact, just what is the characteristic that makes an oil undesirable is a subject of dispute. Some engineers stress the residue content; others, the ash; and still others are influenced by the aromatical factor. In the end an actual service test is the only trustworthy guide. Just what the important characteristics are will be discussed on succeeding pages.

Combustion.—The process of combustion that takes place in the cylinder of an internal-combustion engine is simply a chemical reaction. In actuality the cylinder is a chemist's retort wherein the atoms of hydrogen and carbon, which make up the body of the fuel charge, unite with the oxygen contained in the air charge, forming oxides. The carbon in its union with oxygen forms either carbon monoxide (CO) or carbon dioxide (CO_2). In this latter chemical reaction an atom of carbon unites with two atoms of oxygen, forming one molecule of carbon dioxide; this combustion releases 14,600 B.t.u. per pound of carbon and raises the remaining unburned carbon to the incandescent point. Unless sufficient oxygen is present to unite with this incandescent carbon, the latter unites with one of the oxygen atoms of the carbon dioxide (CO_2), causing the entire carbon oxide to assume the form of carbon monoxide (CO). Since the reaction is incomplete, the heat released by the formation of carbon monoxide from carbon and oxygen is less than that produced by the complete reaction (CO_2). The value is approximately 4380 B.t.u., making evident the heat loss when the combustion is not complete. All Diesel cylinders are of ample volume to give sufficient oxygen for complete combustion.

If the chemical reaction is not fully carried out, it is due to causes other than an insufficient supply of oxygen. In most instances the defect is traceable to poor atomization, wherein the oil charge is not separated into particles of such minute dimensions that each carbon atom contacts with the required oxygen atoms. If the oil droplets entering the cylinder are of fairly large diameter, the oxygen is in direct contact with only the carbon at the surface of the droplet. The carbon atoms within the droplet must receive their oxygen from the carbon dioxide formed at the surface, causing the CO_2 to break into CO and O_2 to supply oxygen to the inner carbon atom. For this reason the engine's efficiency depends on the degree of atomization of the fuel charge. The chemical reactions taking place probably follow this order:

$$C + O_2 = CO_2 \tag{1}$$
$$C + CO_2 = 2CO \tag{2}$$
$$2CO + O_2 = 2CO_2 \tag{3}$$

The hydrogen atom of the oil molecule also unites with the oxygen, forming water (H_2O), or rather water vapor, commonly called "steam," the reaction being as follows:

$$2H_2 + O_2 = 2H_2O \tag{4}$$

This reaction generates 62,100 B.t.u. per pound of hydrogen. The equations do not refer to the actual weight of the carbon, oxygen, and hydrogen but merely indicate the relation of the atoms. Since the atomic weights of the various substances differ, it follows that the weight of each substance entering into the reaction depends on its atomic weight wherein a hydrogen atom has unity weight, carbon 12, and oxygen 16, as listed in Table I,

TABLE I.—ATOMIC WEIGHTS

Element	Symbol	Atomic weight	Molecular symbol
Hydrogen	*H*	1	H_2
Oxygen	*O*	16	O_2
Nitrogen	*N*	14	N_2
Carbon	*C*	12	*C*
Sulphur	*S*	32	*S*

As Table I shows, where oxygen and hydrogen are not in combination with other gases, both oxygen and hydrogen have their atoms, or most minute particles, in groups of two or more. Equation (1) can then be written

$$\begin{aligned} C + O_2 &= CO_2 \\ 12 + 32 &= 44 \end{aligned}$$

indicating that 12 parts by weight of carbon combining with 32 parts of oxygen from 44 parts of carbon dioxide; then 1 lb. of carbon requires $^{32}\!/_{12}$, or 2.67, lb. of oxygen to be converted into 3.67 lb. of CO_2. Since, as shown in Table II, a pound of air contains $^{23}\!/_{100}$ lb. of oxygen, 11.6 lb. of air is required in the combustion of 1 lb. of carbon. Air of 62°F. has a volume of 13.14 cu. ft. per pound; the 11.6 lb. then has a volume of 152.4 cu. ft. The hydrogen reaction (4) can be written as follows:

$$\begin{aligned} 2H_2 + O_2 &= 2H_2O \\ 4 + 32 &= 36 \end{aligned}$$

where 1 lb. of hydrogen requires 8 lb. of oxygen, or 34.78 lb. of air.

TABLE II.—CONSTITUENTS OF AIR[1]

Element	Per cent by weight	Per cent by volume
Oxygen	23.1	20.9
Nitrogen	76.9	79.1

[1] This contemplates dry air, devoid of water vapor, and the rare gases, if present, are included in the nitrogen percentage.

While the petroleum oils contain hydrocarbons of a varied structure, the equation below, covering the ethylene series of hydrocarbons, outlines the process of equating the reactions taking place.

$$\begin{aligned} C_2H_4 + 3O_2 &= 2CO_2 + 2H_2O \\ 28 + 96 &= 88 + 36 \end{aligned}$$

where 28 lb. of ethylene requires 96 lb. of oxygen to form 88 lb. of carbon dioxide and 36 lb. of water or steam. Since 23 per cent of the air is oxygen, 417 lb. of air is required to consume the 28 lb. of C_2H_4, of which 77 per cent, or 321 lb., is nitrogen, which experiences no chemical reaction. Then, for perfect combustion, the percentage by weight of the exhaust products would be:

CO_2 Carbon dioxide	H_2O Water	N Nitrogen
20	8	72

These are the theoretical percentages and are quite different from those obtained on an actual Diesel-engine test where the percentages obtained were as follows:

CO_2 Carbon dioxide	CO Carbon monoxide	N Nitrogen	O Oxygen
7.2	0.2	81.6	11

The H_2O percentage was not obtained in the test.

In Diesel construction the designer always provides a cylinder volume which gives a large excess air capacity. It is usual to figure on 18 to 28 lb. of air per pound of fuel.

Volume Calculations.—In many cases it is desirable to know the volume of oxygen or air needed to burn hydrogen or carbon and the resulting volume of waste products. This is possible by adoption of the mol system.

Mol.—Avogadro's law states that equal volumes of any gases under the same pressure and temperature conditions contain the same number of molecules. It follows, then, that the ratio of weights of equal volume of two gases under these conditions is equal to the ratio of the molecular weights.

A mol is that volume of gas which at 14.7 lb. absolute pressure, or zero gage, and 32°F. temperature has a weight in pounds equal to the molecular weight of the gas, or 359 cubic feet.

For example, the molecular weight of a molecule of hydrogen H_2 is 2; consequently, 359 cu. ft. of hydrogen at 32°F. and 14.7 lb. absolute pressure weight 2 pounds.

If, then, we set down the reaction of carbon and oxygen as given in Eq. (1),

$$C + O_2 = CO_2$$

we have by weight

$$12 + 32 = 44$$

By volume we have, remembering that the oxygen molecule has two atoms,

$$\begin{array}{ccccc} C & + & O_2 & = & CO_2 \\ 1\text{ mol} & + & 1\text{ mol} & = & 1\text{ mol} \end{array}$$

or changing the Mol to its cubic foot volume 359, we have

$$C + O_2 = CO_2$$
$$359 + 359 = 359$$
$$\text{(Cu. ft.) (cu. ft.) (cu. ft.)}$$

This shows that the volume of the resulting carbon dioxide is equal to the volume of the original oxygen and equal to but one-half the combined volumes of the original carbon and oxygen.

It must be understood that, so far, only pure oxygen has been embraced. In air, there is 20.9 cu. ft. of oxygen to every 79.1 cu. ft. of nitrogen, or a ratio of 3.78 cu. ft. of N_2 to 1 cu. ft. of O_2. Consequently, if the total volume of the gases, including the inert nitrogen, is desired, the nitrogen content must be included in the computation. In place of

$$\begin{array}{ccccc} 1\ \text{mol} & + & 1\ \text{mol} & = & 1\ \text{mol} \\ C & + & O_2 & = & CO_2 \end{array}$$

the equation, including the nitrogen, becomes

$$\begin{array}{ccccccccc} 1\ \text{mol} & + & 1\ \text{mol} & + & 3.78\ \text{mol} & = & 1\ \text{mol} & + & 3.78\ \text{mol} \\ C & + & O_2 & + & N_2 & = & CO_2 & + & N_2 \end{array}$$

Changing this into cubic feet,

$$\begin{array}{ccccccccc} 359 & + & 359 & + & 1{,}357 & = & 359 & + & 1{,}357 \\ \text{(Cu. ft.)} & & \text{(cu. ft.)} & & \text{(cu. ft.)} & & \text{(cu. ft.)} & & \text{(cu. ft.)} \\ C & + & O_2 & + & N_2 & = & CO_2 & + & N_2 \end{array}$$

Combustion of Actual Fuel.—The method of calculating the combustion of hydrogen and carbon may be extended to the study of an actual fuel oil.

It will be observed that 1 atom of carbon calls for 2 atoms of oxygen and that two atoms of oxygen are in each oxygen molecule, so that to burn a hydrocarbon it is necessary to have as many molecules of oxygen as there are carbon, for a molecule of carbon has only one atom.

For every hydrogen atom H, there is needed ½ oxygen atom, since the combination is H_2O, so there will be needed one-half as many oxygen molecules (O_2) as there are hydrogen molecules (H_2), for each hydrogen molecule has two hydrogen atoms, or ¼ oxygen molecule for each hydrogen atom.

Taking, for example, a hydrocarbon of C_nH_m, and using the mol system already explained, there will be the following volume of the gases oxygen and nitrogen before combustion:

$$\begin{array}{ccccccc} 1 \text{ mol} + n \text{ mols} & + & 0.25m \text{ mols} & + & 3.78(n + 0.25m) \text{ mols} \\ C_nH_m + O_2 & + & O_2 & + & N_2 \end{array}$$

The n mols of oxygen are needed for the carbon, and the $0.25m$ mols for the hydrogen.

It will be seen that since there are 3.78 times as much nitrogen as oxygen in the air, that the nitrogen mols equal 3.78 (oxygen) = $3.78(n + 0.25m)$ mols.

The gases after combustion will be

$$\begin{array}{ccc} n \text{ mols} + 0.5m \text{ mols} & + & 3.78(n + 0.25m) \text{ mols} \\ CO_2 + H_2O & + & N_2 \end{array}$$

Apply this to the hydrocarbon C_7H_{16}. We have 1 mol of C_7H_{16}; also, since the number of mols of oxygen in the air supply equals the number of carbon atoms in C_7H_{16}, the value of n is 7. Also, we must have an amount of oxygen mols to take the hydrogen equal to $0.25m$, where m is the hydrogen atoms in C_7H_{16}, or a total of $0.25 \times 16 = 4$ oxygen mols. In addition, we will have nitrogen in the air supply equal to $3.78(n + 0.25m) = 3.78(7 + 0.25 \times 16) = 41.58$ mols of nitrogen. One may write the gases before combustion as

$$\begin{array}{ccccccc} 1 \text{ mol} & + & 7 \text{ mols} & + & 4 \text{ mols} & + & 3.78(7 + 4) \text{ mols} \\ C_7H_{16} & + & O_2 & + & O_2 & + & N_2 \end{array}$$

The gases after combustion are

$$\begin{array}{ccccc} 7 \text{ mols} & + & 8 \text{ mols} & + & 3.78(7 + 4) \text{ mols} \\ CO_2 & + & H_2O & + & N_2 \end{array}$$

It will be observed that before combustion, assuming the hydrocarbon to be in the gaseous state, there were

$$53.58 \text{ mols} = 53.58 \times 359 = 19{,}235 \text{ cu. ft.}$$

After combustion the volume of the gases at 60°F. and atmospheric pressure is 56.58 mols = 20,312 cu. ft.

In other words owing to combustion, there is a volume increase of 3 mols, or 1,078 cu. ft.

Actually, however, the fuel before combustion is a liquid, so its volume is not one mol but is insignificant. The gain in volume by combustion is then 4 mols, or 1,436 cu. ft.

This explains why fuel injection gives a superior efficiency over a carburetor at the same compression ratio, assuming both engines to operate on the constant-volume cycle. With a carburetor engine the fuel vapor would occupy 1 mol, or 359 cu. ft., so the volume increase is, as shown, from 19,235 to 20,309 cu ft. In the case of a fuel-injection engine the volume of the liquid fuel is insignificant; consequently the volume before combustion would not include the mol of gas fuel and, so, would be $53.58 - 1 = 52.58$ mols, or 18,776 cu. ft. The volume after combustion would be the same with both engines, namely, 20,309 cu. ft. The volume increase with fuel injection is, then, 1,533 cu. ft., and with the carburetor engine 1,074 cu. ft. During expansion, then, the fuel-injection Diesel will deliver more power, since the gases experience over and above the thermal expansion a greater volume expansion due to the chemical reaction.

Of oxygen and nitrogen, there are needed $359 + 1{,}357 = 1{,}716$ cu. ft. to burn 1 mol, or 359 cu ft. of carbon, but 1 mol of carbon weighs 12 lb., so 1 lb. of carbon requires $1{,}716 \div 12 = 146$ cu. ft. of air at 32°F. and 14.7 lb. absolute pressure, for complete combustion.

Combustion Characteristics.—It is necessary, if good combustion is to be obtained in an engine, that the fuel be mixed with the entire mass of air trapped in the cylinder-clearance volume To accomplish this, the following actions must occur:

1. Atomization of fuel
2. Mixing with air charge

Atomization is accomplished either by the action of the spray nozzle or by the action of the air charge. If the oil issuing from the spray has sufficient energy to carry it to all parts of the combustion chamber and is firmly atomized by the nozzle design or the low surface tension of the oil drops, no more can be expected of the nozzle.

It is a fact, however, that at high-piston speeds, atomization by nozzle effect cannot be depended upon entirely. For this reason, various devices to encourage the air to meet the oil have been developed. These include special piston crowns, precombustion chambers, turbulence chambers, and antechambers. The effect of each will be discussed later.

Combustion Events.—In a Diesel cylinder the fuel must be injected, a part vaporized and ignited, and then the mass of the fuel burned.

If the first portion of the injected fuel does not ignite before the entire fuel mass enters the cylinders, combustion of all the fuel will occur practically simultaneously. This resulting pressure wave set up travels at the velocity of sound, so there occurs a sharp knocking, termed "combustion" knock. This knock is not only indicative of the existence of dangerous pressures but also indicates that the efficiency is low.

The problem is to obtain ignition of the first particles before the main mass issues from the spray valve. In a slow-speed Diesel, no difficulty exists, for the rate of injection is relatively low, and the primary ignition occurs. At high speeds, the time interval is so small that knocking may occur. It has been found by several investigators that, if the cylinder pressure rise is not greater than 25 lb. per degree of crank travel, knocking will not occur, but that at a 50-lb. rate of rise, knocking will be noticeable.

The claim has also been advanced that the "measured knock," which is equal to the extreme rise of pressure *EPR* times the rate of rise in pounds per degree of crank travel *RR* divided by 10,000, should have a value of 2 or less, if knocking is to be controlled. Stated in an equation,

$$EPR \times RR \times 10^{-4} = 2 \text{ or less}$$

Ignition Lag.—Experiments prove definitely that the compression pressure has an influence on the lag in ignition. This is to be expected, since the higher the compression the higher will be the air temperature; naturally the higher the air temperature the faster the initial oil charge will be vaporized and ignited.

As pointed out by Prof. P. H. Schweitzer, when knocking occurs in a spark-ignition engine, the rapid pressure rise takes place and the knock occurs at the last part of the combustion. It is the "last gas" to burn that knocks. The first portion of gas ignited by the electric spark certainly could not knock, because only gas adjacent to the spark plug ignites, and this is of minute quantity. The propagation of the combustion is a gradual process for a while. The flame front expands and in doing so compresses the unburnt portion of the mixture ahead of it. Finally, according to Janeway, a point is reached when the unburnt portion is compressed to its self-ignition temperature.

This charge, then, ignites in its entire mass, causing a very rapid pressure rise and a sharp knock, frequently called "detonation."

In a Diesel engine, on the contrary, the combustion *begins* with autoignition. But self-ignition need not necessarily cause detonation, as thousands of smooth-running Diesel engines testify. The determining factor is *how much* fuel ignites simultaneously. The more fuel igniting in the cylinder practically *at the same time*, the steeper is the resulting pressure rise, and the greater the knock.

The fuel injected does not ignite instantaneously. It takes ordinarily 0.001 to 0.004 sec. for the cylinder air to heat up the injected oil to its self-ignition temperature. This delay gives the fuel opportunity to accumulate in the cylinder before any ignition occurs.

If the injection is uniform over 20°, more than half of the fuel charge accumulates in the cylinder during the ignition lag. This accumulated fuel ignites practically at once, as if thousands of spark plugs had set it afire simultaneously. The result is a violent pressure rise and an audible knock. The longer the delay before ignition the steeper the maximum rate of pressure rise, and the greater the knock. This fact was confirmed by Rothrock, who found a linear relation between ignition lag and maximum rate of pressure rise.

Although the fuel is the most important factor with regard to Diesel knock, frequently the same fuel will knock in one engine and run smoothly in another.

In designing a Diesel engine for smooth running, one governing principle should be the shortening of the ignition lag. The problem is essentially one of heat transfer between the hot air and the cold oil drops. The greater part of the ignition lag is the period during which the oil globules are warmed up to their self-ignition temperature. This period increases with decreasing compression-air temperature, with decreased relative velocity between the hot air and the oil globule, and with increasing oil-drop size. An effective way to shorten the period is to direct the oil spray so that it reaches the hottest part of the air. In an engine with a turbulent-type combustion chamber, the spray should first meet such part of the air stream as has not been chilled very much by flowing with high velocity near cooled surfaces. Such devices as high intake-air temperature, high cylinder-head temperature, and extra-high-compression ratio have a beneficial effect on the ignition lag. Of course they may be objectionable for other reasons.

Cutting the ignition lag is one way to reduce the knock in a Diesel engine. Another is by reducing the amount of fuel introduced during the ignition-lag period. If the rate of injection, instead of being uniform during the entire injection period,

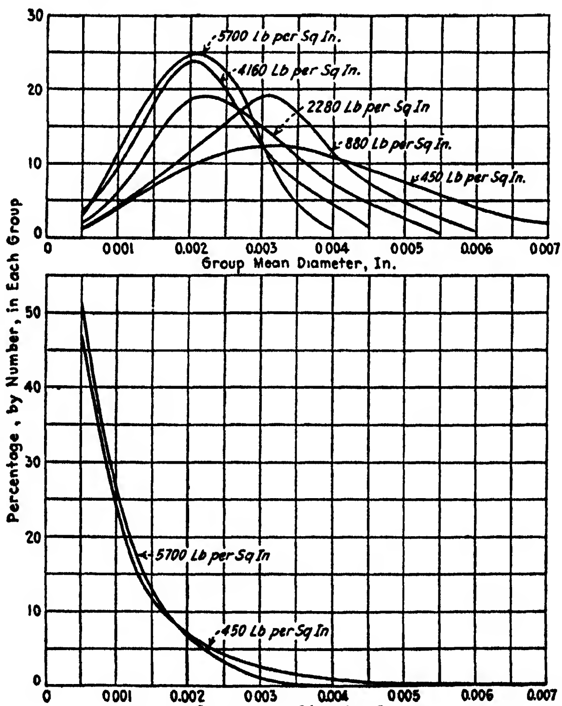


FIG. 87.—Size of fuel particles.

is low in the beginning and increases later with progressing injection, the portion of the total injected during the delay period will be relatively less. It is most desirable to have injection begin very gradually. Unfortunately, in most engines this condition cannot exist. The rate of injection is in the very beginning as high as or higher than after the ignition lag.

Size of Oil Drops.—In a test at Langley Field, Dana W. Lee found that the size of the oil drops after leaving the nozzle was dependent upon the injection pressure, as shown in Fig. 87. Furthermore, the nozzle diameter had some effect, but the effect of the kind of spray, centrifugal or plain nozzle, was not marked, as Fig. 87 *A* shows.

Fuel Characteristics.—With the advent of the cracking still in refineries, whereby the crude oil was "cracked" to take up the heavy hydrocarbons, to form lighter oil molecules, that is, gasoline, the supply of "straight-run" distillates for Diesel engines has shrunk. As a result, blended oils came into use; these consisted of part of the stock recirculated in the cracking

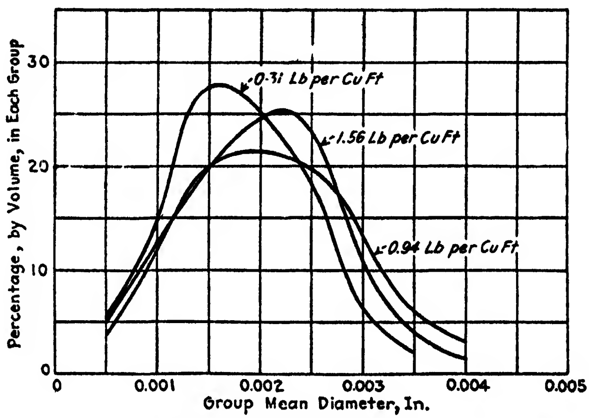


FIG. 87 *A*.—Influence of fuel quantity on droplet size.

still, combined with some distillate. In addition, crude oils come from a hundred sources, and the behavior of the refinery products cannot always be predicted.

The development of the high-speed Diesel brought with it certain limitations in the fuel that such engines could burn. At first, the general belief was that the fuel oils sold for house oil burners could be burned in high-speed Diesels. That this was not always true was evidenced by a number of engine failures. It was apparent that Diesel fuel-oil specifications must be developed, for which several engineering associations appointed committees.

Finally, a set of tentative specifications was evolved by the American Society of Testing Materials. These have been changed slightly from time to time, and in 1938 the committee discarded specifications 5 and 6 (for heavy oil). In Table III appear the specifications as of 1938 and old grades 5 and 6.

TABLE III.—TENTATIVE CLASSIFICATION FOR FUEL OILS

Characteristics	A distillate oil for use in engines requiring a low-viscosity fuel	A distillate oil for use in engines requiring a medium-low-viscosity fuel	An oil for use in engines requiring a medium-viscosity fuel	An oil for use in engines permitting a medium-high-viscosity fuel	An oil for use in engines of special design for high-viscosity fuels and after engine manufacturers' recommendations only
Grade No.	1-D[a]	3-D[b]	4-D[c]	5-D[d]	6-D[d]
1. Flash point, °F., min	115[e] or legal	150[e]	150[e]	150[e]	150[e]
2. Water and sediment, per cent by volume, max	0.05	0.2	0.6	(to be inserted later)	
3. Viscosity, sec.:					
Saybolt Univ. at 100°F., min	35	30			
max	50	70	500		
Saybolt Furol at 122°F., max	....	....	....	100	300
4. Carbon residue, per cent, max	0.2	0.5	3.0	6.0	10.0
5. Ash per cent by weight, max	0.02	0 02	0.04	0.08	0 12
6. Pour point °F., max	35[f]	35[f]	35[f]	40[f]	40[f]
7. Ignition, quality:					
A. Diesel-index No., min	45	30	20	[g]	[g]
B. Cetene, No., min	45	40	30	[g]	[g]
C. Viscosity—gravity No., max	0.86	0.89	0 91	[g]	[g]
D. Boiling point—gravity No., max	188	195	200	[g]	[g]

[a] Grade 1-D is recommended for mechanical (solid) injection engines of the high-speed type; in general, for engine speeds over 1,000 r.p.m.

[b] Grade 3-D is recommended for mechanical (solid) injection engines of the medium-speed type; in general, for engine speeds from 360 to 1,000 r.p.m.

[c] Grade 4-D is recommended for air-injection engines, both two- and four-stroke cycle types with speeds not over 400 r.p.m. Grade 4-D can be used for mechanical (solid)-injection engines with cylinder diameters over 16 in. and speed under 240 r.p.m., but approved heating equipment furnished by the engine manufacturer is recommended.

[d] Grade 5-D is recommended for air-injection engines of the slow-speed type; speed under 240 r.p.m. Manufacturer should be consulted for approved heating equipment. Grade 6-D is not regularly used for Diesel engines and is not recommended unless tested and approved by engine manufacturer. The purchaser should be advised regarding high maintenance cost of engines and operating problems involved in the use of this grade of fuel. Grades 5 and 6 are no longer standard.

[e] Minimum flash as stated or as required by local fire regulations, fire underwriters, or state laws.

[f] Lower pour points may be specified whenever required by local temperature conditions to facilitate storage and use, although it should not be necessary to specify a pour point of less than 0°F.

[g] No ignition quality is specified, but burning quality should be determined by actual engine test.

[e] So far as known, sulphur content need not be considered as regards combustion characteristics. However, a maximum of 2 per cent is suggested to purchasers operating engines in intermittent service.

Most of the various specification items are self-explanatory. Mention, however, will be made of the three factors *Diesel Index*, *Cetene*, and *C.C.R.* (critical compression ratio).

All three of these factors are indicative of the ignitibility of the the fuel oil, and various tests show reasonable correlations.

Combustion Knock.—According to present view, combustion knock in a Diesel engine is caused by the accumulation of fuel in a cylinder prior to ignition. Even under conditions that insure ignition, the fuel does not ignite instantaneously but only after a definite delay. The longer this delay the more fuel accumulates in the engine, which then burns more or less simultaneously, accompanied by an audible knock. If the delay is extreme, the engine will fail to operate with the fuel. It is significant that practically every factor that tends to aggravate knocking in a spark-ignition engine tends to suppress it in Diesel engines. Fuels of paraffin base, consisting chiefly of saturated straight-chain hydrocarbons, give the smoothest combustion in Diesel engines, whereas naphthenes and aromatics burn roughly.

Ignitibility.—As a consequence, there has been a practical unanimity among Diesel engineers that ignitibility is a most decisive factor. Methods of ascertaining the relative ignitibility are of both the laboratory and the test-floor types, although actual engine tests are viewed as the most satisfactory.

The success of the gasoline knock-testing engine, whereby gasoline is rated by its performance, duplicating the performance of a test fuel composed in part of octane, encouraged the search for a method of determining Diesel fuel quality by its behavior in the engine itself.

A variety of methods have been used by Le Mesurier & Stansfield, Boerlage & Broeze, and Pope & Murdock; in principle, however, all of these agreed. The "ignitibility" as such is not measured by any engine test, only determined indirectly on the basis of the behavior of the fuel in the test engine in regard to such characteristics as combustion knock, ignition lag, rate of pressure rise, or ease of starting.

No matter what testing method is adopted for measuring ignitibility, it is essential to determine the manner in which the rating of the fuel shall be expressed.

One method, used for knock rating of gasolines, was proposed for rating the ignitibility of Diesel fuels, based on two standard reference fuels, one having a high, the other a low, ignitibility. The test fuel was to be matched with a fuel consisting of the two reference fuels in such a proportion that its ignition quality (in the arbitrary scale of the test) would be equal to that of the test fuels. The rating of the fuel would be expressed, then, as the percentage concentration of the high-rate reference fuel.

Boerlage & Broeze proposed, as a primary standard, mixtures of cetene ($C_{16}H_{32}$) and alpha menthil naphthalene ($C_{11}H_{10}$). The cetene number in the specifications indicates the percentage of cetene in the reference fuel, a blend of cetene and alpha methyl naphthalene, which has the same ignition characteristics as the fuel to be used in a particular Diesel. The engine builder will be in a position to state this value to the engine owner.

It is, of course, understood that engines having different designs of combustion chambers, different speeds, and different compression pressures will require fuels with different cetene numbers. All these factors influence the ignitibility of a fuel.

The Test Engine.—The C.C.R. (Critical Compression Ratio) is by definition the compression ratio, in the C.F.R.-Waukesha test engine with the new turbulence-chamber cylinder head, at which the oil tested will fire within a definite time interval when the engine is motored.

The procedure, originated by Pope & Murdock, is to drive the engine with an electric motor as an air compressor with no fuel admitted. When the motor attains steady running conditions, fuel is admitted for a short interval (3 sec.); and if firing occurs, the compression ratio is lowered until a further lowering of the compression ratio suppresses firing. The lowest compression ratio at which firing takes place is used as the basis of rating. This method is referred to as the "C.C.R. method."

The operating conditions of the C.C.R. engine are as follows:

Speed, r.p.m.	600
Fuel feed, c.c. per min	9
Injection advance angle, deg. before top dead center	12
Injection pressure, lb. per sq. in.	1,500
Jacket temperature, °F	212
Air-inlet temperature, °F	100

The C.C.R. is obtained by motoring the engine at the selected speed and noting the compression ratio at which firing occurs (as judged by the sound of the exhaust) within 3 sec. after the fuel is injected.

It may be felt by some that an actual engine may need an entirely different compression to insure ignition. This is entirely true, but the engine builder is able to run tests on his engine with a given fuel and then ascertain from the refiner the C.C.R. value of the fuel. This will permit the builder to specify the corresponding C.C.R. of his engine.

Aniline Number.—Various tests have indicated that there is a definite correlation between the aniline number of a fuel and its ignitibility. This has been taken advantage of by A. E. Becker, who advanced a factor that he terms "Diesel index." The expression is

$$\text{Diesel index} = \frac{\text{aniline point (°F.)} \times \text{A.P.I. gravity}}{100}$$

"Aniline point" is the lowest temperature at which equal parts by volume of freshly distilled aniline and the test sample of oil are completely miscible. It is determined by heating such a mixture in a jacketed test tube to a clear solution and noting the temperature at which turbidity appears as the mixture is cooled.

This test is indicative of the amount of paraffin-base oil in the test fuel. As has been stated previously, a straight paraffin oil gives the best results from an ignition standpoint, with an oil showing increased unsatisfactory performance as the percentage of paraffin decreases. Becker has shown that there is a close correlation between Diesel index and "cetene number." On the other hand, certain petroleum technologists point out that the correlation is close only in a narrow range of cetene percentages, around 50.

It is the opinion of most engineers who have studied the problem that any one of the three factors may be employed, so that all three need not be included in any specification. The Diesel index is the easiest to discover, but many feel that cetene is the best criterion of the three.

These standards should be tested by all engine builders, for if corrections or alterations are to be made, they should be done at an early date. The various oil refiners are willing to supply the necessary fuel for such test purposes.

The committee evolving these standards, and especially Lee Schneitter, the chairman, deserves the commendations of the entire petroleum and Diesel industries.

Viscosity.—Viscosity, or body, of an oil, as it is more generally understood by the average operator, is a measure of the relative fluidity at the temperature of observation. It can also be regarded as that inherent property by virtue of which flow is retarded, owing to the resistance which the particles or molecules of an oil will offer to one another as they flow through the fuel system. It is possessed by all oils to a varying degree, according to their extent of refinement and source. Viscosity will vary inversely with temperature; that is, the colder an oil the heavier or more sluggish will it become. In contrast, as the temperature is raised, the same oil will become more and more fluid.

Viscosity largely determines the degree of atomization experienced by the oil as it sprays out through the fuel-spray orifice. If sufficiently low at the pump's pressure, the oil stream will have a turbulent flow; consequently the oil particles will be broken up by their turbulent action. A more viscuous oil will retain a non-turbulent flow at the same pressure and so will not possess many self-atomization characteristics. Atomization of the heavy oils must, then, depend upon centrifugal action of the nozzle, air turbulence, or reduction of viscosity by heating the oil.

Viscosity Test.—The Universal machine is used to determine the viscosity of light and medium oils, standard temperatures of test being 100, 130, or 210°F. The Furol machine is used for heavy fuel oils or other oils of similar viscosity at standard temperatures of 77, 122, or 212°F. The essential difference between these machines is in the diameter of the outlet tube or orifice, the Furol tube being the larger to accommodate flow of heavier oils.

The Furol machine is about ten times faster than the Saybolt machine; as a result, a numerical reading by the Furol instrument would be one-tenth as great as shown by the Saybolt machine for the same oil; a viscosity of 75 sec. Furol is equal to 750 sec. Saybolt.

Carbon Residue.—A certain percentage of a fuel may be incapable of vaporization and combustion. This includes any ash remaining after combustion of all organic substances and, likewise, those hydrocarbons and free carbon particles in the oil which are non-combustible at working temperatures. Since most of this residue is carbon, it is customary to call it "carbon residue," or "Conradson carbon residue," after the name of the test equipment.

Value of Conradson Test.—Theoretically, the Conradson test should be a guide to the amount of carbon and other non-combustibles in the oil. However, certain oils contain heavy gumming hydrocarbons which are impossible to burn in a Diesel cylinder. These will be driven off during the Conradson test; consequently an oil may show, say, 2 per cent Conradson while actually containing as much as 10 per cent of constituents impossible to be burned in the engine. At present many employ a modified Conradson test; the oil is heated, and the vapor ignited. After the flame dies out, owing to the absence of any more combustible vapor, the remainder in the crucible is termed carbon residue and is always greater than the residue obtained by the Conradson test.

Flash and Fire Points.—When the temperature of a petroleum product is raised, a point will be reached where enough surface vapor is developed to ignite for a moment upon the application of a flame. The temperature of the oil at the moment of flash is regarded as the "flash point" of the product under test.

The temperature to which the oil must be further raised to cause the oil to continue to give off a vapor sufficient to maintain the flame is termed the "fire point."

This test for flash point is arbitrary in that it depends upon the type of apparatus used, that is, whether this latter employs an open or a closed cup. In the United States the open cup prevails in the testing of lubricating oils, the closed-cup device being more largely confined to foreign usage or to oils of intermediate flash point.

Ash and Water.—Ash, unless in unusual quantities, is not objectionable, although the specifications for No. 1 fuel oil should be considered as 0.02 per cent. The Navy is even more stringent, calling for a maximum of not over 0.01 per cent. Ash is the refuse after combustion and may include rust, scale, and oxide, all of which may cause damage to cylinder liners and, more especially, to the spray orifices. Usually the test is combined with the water test.

CHAPTER VII

COMBUSTION SYSTEMS

General.—The greatest development during the last few years in oil-engine design is found in that class usually termed "solid-injection," or "airless-injection" Diesels. Although these engines make up the greater part of the present output of the many Diesel factories and show marked variation in design and, for that matter, in behavior, actually they may be divided into three classes.

Although we might classify these engines under the heading of two-stroke-cycle and four-stroke-cycle, a better and more logical basis is that of combustion arrangement, and these may then be broken down into a classification based on the fuel-injection systems used by the engines. Furthermore, each of the combustion arrangements give definite and almost invariable operating characteristics to the engine; if one desires to select an engine, one will do well to understand the several arrangements, or, shall we say, principles, and their effect on engine operation.

Combustion-chamber Classification.—The three general principles are shown in Fig. 88 at *A*, *B*, and *C*. That shown at *A* consists of a plain cylinder with a flat head carrying the admission, exhaust, and fuel-spray valves. It is essentially the identical design of cylinder embodied in the design of practically all Diesels using air injections and differs in no great measure from a gas-engine cylinder. Its ancestor may be placed as the original Diesel engine. The air is drawn in through the admission valve on the suction stroke and is compressed on the compression stroke. At the end of this stroke, oil is sprayed into the charge of hot compressed air, is ignited by the air temperature, and combustion ensues.

With the arrangement shown at *B*, the cylinder head carries a cavity of a volume sufficient to contain the entire cylinder charge when the piston reaches the end of the compression stroke. Into this cavity the oil is injected and, mixing with the air charge

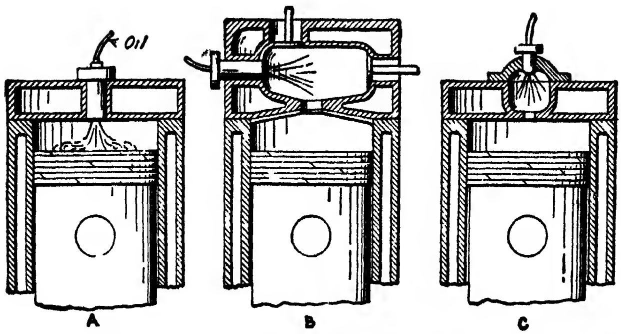


Fig. 88.—Three combustion systems used with solid-injection Diesels

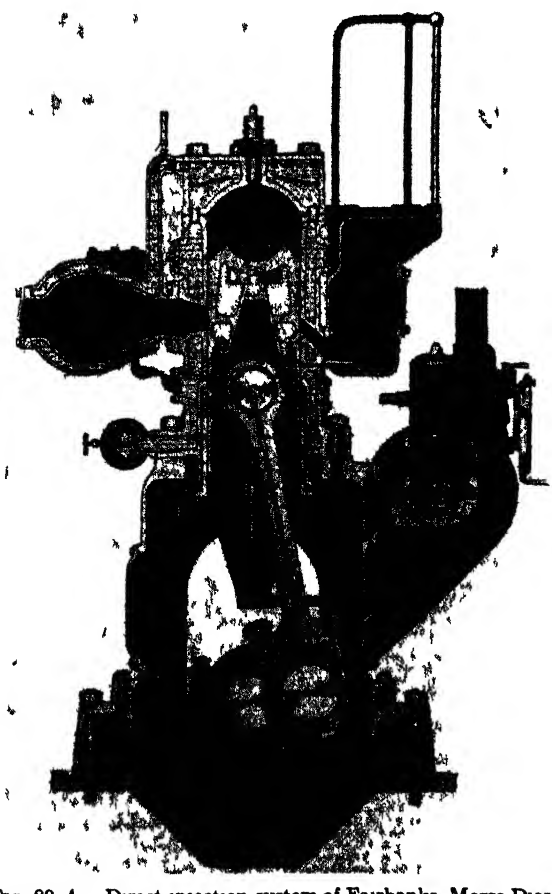


Fig. 89 *A*.—Direct-injection system of Fairbanks, Morse Diesel.

in the cavity, ignites and burns. Combustion takes place, or should take place, entirely within this cavity; and as this must occur practically instantaneously and while the piston is at the end of the compression stroke, ready to begin its power stroke,

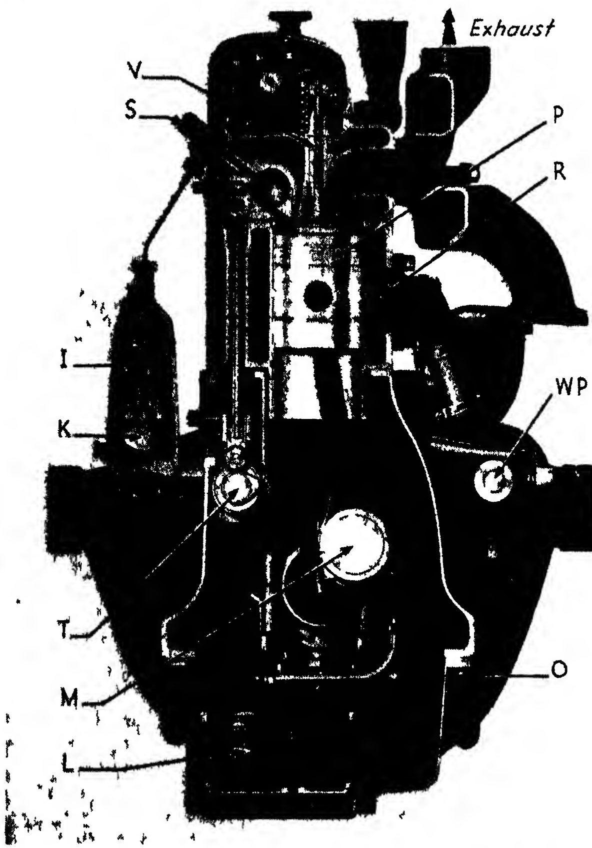


FIG. 89 *B*.—Hercules turbulence-chamber Diesel. V = exhaust valve, S = fuel spray nozzle, I = fuel pump; K = fuel pump cam; T = camshaft, M = crankpin; L = lubricating-oil pump, O = lubricating-oil piping; WP = cooling water pump; R = lubricating-oil cooler, P = piston.

the pressure rises much above the compression pressure. In actual engines combustion is not completed instantaneously but continues during the early part of the power stroke. The variation in the shape of the chamber and position of the fuel spray valves will be discussed later.

In the modern engine the combustion chamber is moved to one side of the cylinder center line and the valves are in the cylinder head and not in the chamber. Such a turbulence chamber is shown in Fig. 89 *B*.

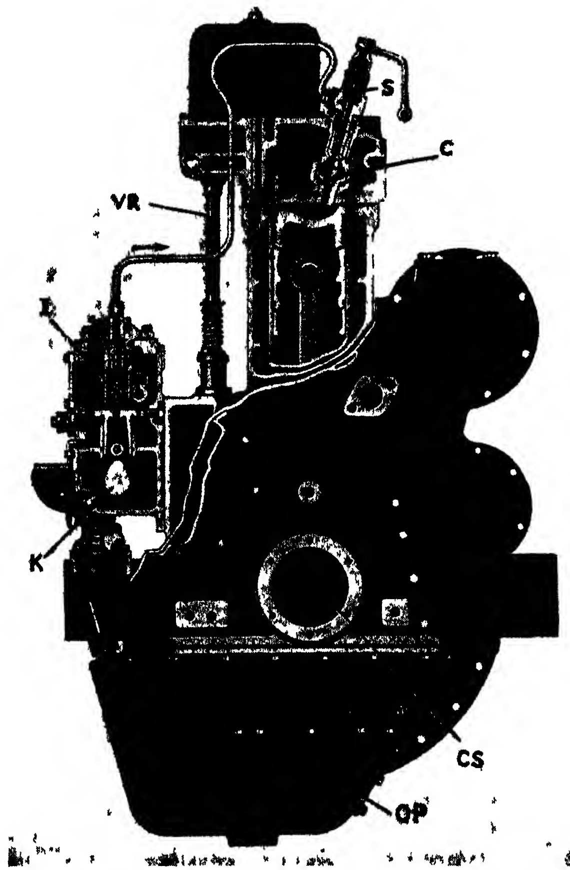


FIG 89 *C*.—Caterpillar precombustion-chamber Diesel. *C* = precombustion chamber; *S* = fuel spray valve, *VR* = valve push rod, *I* = fuel pump; *K* = fuel-pump cam; *CS* = crankshaft, *OP* = oil pan.

At *C* is shown the so-called "precombustion-chamber" design. In the cylinder head is located a small chamber connected to the working cylinder by an orifice, either a single opening or a multiplicity of small openings.

As the air can flow through the orifice during the compression stroke, the chamber is charged with the hot compressed air, and at the proper time fuel is sprayed into the air. Because of the volumetric limitations of the chamber, only a small amount of oil can find air with which to combine, and the remainder vaporizes and blows through the orifice into the working cylinder Here it meets the main air charge, and combustion follows. In some designs oil rather than oil vapor is injected into the cylinder, but all these engines depend for their action upon the prior combustion of a small portion of the fuel charge

A modern design of precombustion chamber appears in Fig 89 *C*, page 149.

Referring again to type A (Fig. 88), the majority of large-bore mechanical-injection Diesels incorporate this combustion system

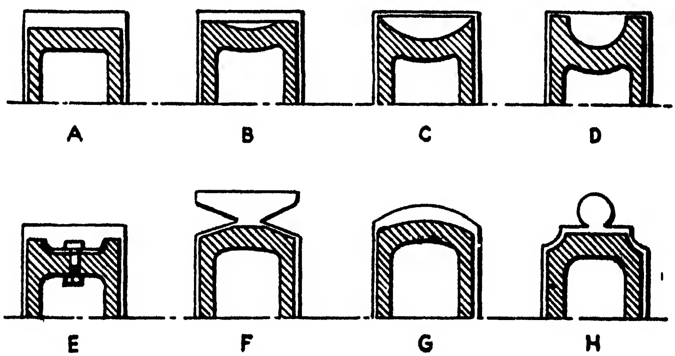


FIG. 90.—Different designs of piston crowns

in their design. The distance between the spray nozzle and the piston head is great enough to allow the fuel spray to diffuse throughout the mass of hot air. This prevents any great amount of the oil from striking the piston crown, cracking, and throwing down carbon. When oil does strike the piston, a great amount of heat is absorbed by the piston. This may be taken care of by a shield of steel bolted to the piston top or by cooling the piston crown interior.

To provide sufficient depth between the spray and piston on many engines, the piston is provided with a concave crown. In Fig. 90 are shown several different arrangements in general use.

When the cylinder bore becomes small, say from 4 to 6 in., the distance between nozzle and piston crown is small, and direct injection is not always successful. In some engines the spray nozzle has its orifice spaced almost horizontally, so that the oil will jet out in a flat sheet parallel to the piston crown.

The advantage of injection of the fuel directly into the cylinder is that it is not necessary to carry the compression pressure so high as is necessary in other designs to obtain autoignition of the fuel. On the other hand, combustion is somewhat out of control, so that the pressure rise after ignition of the fuel may be high. In the General Motors direct-injection, two-cycle Diesels the maximum pressure is as much as 1,100 lb. per square inch.

To offset this, the direct-injection engine will usually develop a high rating and, in addition, will start cold without the need of a heating coil. It is necessary to use a multi-orifice spray valve to insure diffusion of the oil throughout the air mass, although a pintle nozzle is often satisfactory.

The turbulence chamber (Fig. 89 *B*) has the advantage that when the oil is sprayed into the chamber, the entire cylinder charge of air is massed in the spherical or flat-cylindrical chamber This air is rotating, or is at least turbulent, so that mixing of oil and air is immediate and thorough.

The objections offered at times are that in the flow through the throat the air loses considerable heat, so the engine is hard to start when cold, and that the throat offers resistance to the return flow of these gases, thereby reducing the engine power. This objection has been overcome by some designers who are able to obtain as high as 130 lb. brake m.e.p. To prevent heat loss, designers arrange to have the throat poorly cooled or, in some instances, employ a lining which is separated from the head casting by a minute air space.

The precombustion engine (Fig. 89 *C*) has the advantage that it insures ignition of the charge and holds down the maximum cylinder pressures. This is, in some respects, a handicap, as the engine rating is low per cubic inch of cylinder volume.

Since mixing of fuel and air depends upon the air flow through the throat, the oil spray is directed toward the throat and may be relatively coarse. Consequently, the oil pressure can be low—from 1,000 to 1,500 lb.

The precombustion chamber is found in more American Diesels than any other.

Air-cell System.—When fuel is injected in the engine cylinder, the oil sprays do not reach all parts of the air. There are, in fact, regions of "lean" and "rich" mixtures. To obtain a complete mixing, insuring perfect combustion, Lang invented the Acro air-cell system. This adds a small auxiliary air chamber to the cylinder. On the compression stroke, air is forced into the cell; and when combustion occurs, the high cylinder pressure forces more air into the cell. As soon as the piston starts in its power stroke, the cylinder pressure drops sharply. The high-pressure

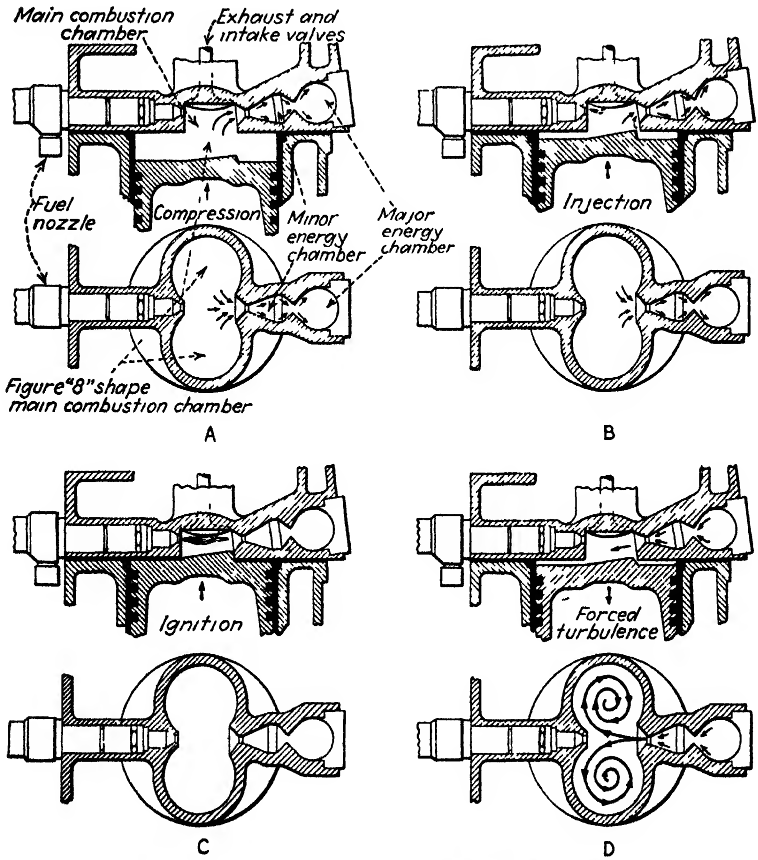


FIG. 91.—Action in a Lanova energy-chamber Diesel.

air in the cell then rushes out, creating a turbulent action of the cylinder-oil vapor and air.

Lanova System.—Lang developed the Lanova system by arranging to have a small part of the fuel enter an energy cell.

The action of the Lanova system is as follows: With reference to Figs. 91 *A* to *D*, it will be seen that the energy cell of the

Lanova combustion system is of relatively small volume. As a matter of fact, it is less than 20 per cent of the total compression volume, as compared to the main combustion chamber. In this respect it differs basically from the auxiliary air-storage chambers of the air-cell-type engines, such as the Acro, in which the air-storage chamber contains approximately 70 per cent of the total compression volume, air being utilized to support combustion on the outward stroke of the piston. The second fundamental difference of the Lanova energy cell from an air chamber is that it is designed with two or more restricted openings, by means of which the time and rate of injection of the turbulence-creating blast are from the energy cell into the main combustion space so controlled as to give the desired results.

During the compression stroke, air is forced into the energy chamber (Fig 91 *A*); and when fuel is injected across the main two-lobe clearance space (Fig. 91 *B*) toward the throat leading to the energy chamber, a small oil charge is carried into the energy cell, as shown in Fig. 91 *C*. The pressure here is lower than in the cylinder as the result of wiredrawing in the throat.

Fuel injection starts at about 17 deg. before top dead center, and at 6 deg. before dead center (Fig. 91 *C*) oil particles along the outer surface of the oil spray ignite. The flame travels into the energy chamber, as shown in Fig. 91 *C*, where the oil-air mixture in the chamber ignites. The pressure rises sharply to about 1,400 lb., and the flaming gas contents of the energy cell back-flows into the main combustion chamber, where the oil and air are completely mixed by the turbulence indicated in Fig. 91 *D*.

CHAPTER VIII

OIL-ENGINE INSTALLATION

General.—The increasing number of oil engines being installed in industrial plants indicates that engineers will more and more encounter erection work involving the building of engine foundations. While this work merely calls for the same degree of care and skill that is required in the erection of any machine, there are a few points which, if observed, will make the work fairly easy. Most Diesels are installed to drive generators, but there are many instances where it is necessary to line up the Diesel with a line-shaft. The following pages may be of assistance.

Establishing Center Line.—The first step is the locating of the desired engine and driving-pulley center lines in relation to the shaft and pulley to be driven. Each installation has an individual problem as to permissible belt centers. As a general rule an engine from 50 to 100 hp. should be set at least 18 ft. from the driven shaft; with larger units 25 ft. should be the minimum. Two plumb bobs should be dropped from one side of the shaft, immediately against each side of the pulley to be driven. Posts should be set below each bob with the top a foot or so above the ground. Center tacks should then be put in marking the bob points. The shaft should next be revolved a half revolution to see if the pulley is square on the shaft. A board is nailed across the two posts mentioned, and a tack, in alignment with the first tacks and at a point half the distance between two, is then put into the board. This gives a point that is on the direct pulley center line. The bob is next dropped over the shaft at a distance of 16 ft. from the center tack just mentioned; a tack located here now gives two points on the line of the driven shaft, as shown in Fig. 92, one of which is also on the pulley center line.

A wire is now stretched from this point *A* at approximate right angles to the shaft center line. A mark *C* is made on this wire 12 ft. from the tack, as shown, and a steel tape stretched from the tack *B* to the mark *C*. The wire is shifted until the tape from

B measures 20 ft. to the point *C*. This is the 3–4–5 method outlined in Fig. 93, wherein if 3 ft. is laid out on one line from *A* and 4 ft. on a second line measured from the intersection *A*, then the two lines are at right angles if the diagonal *BC* measures 5 ft. Multiples of 3–4–5 may be taken, such as 9–12–15 or 12–16–20, as used in the aligning (Fig. 92). The piano wire, on being set at

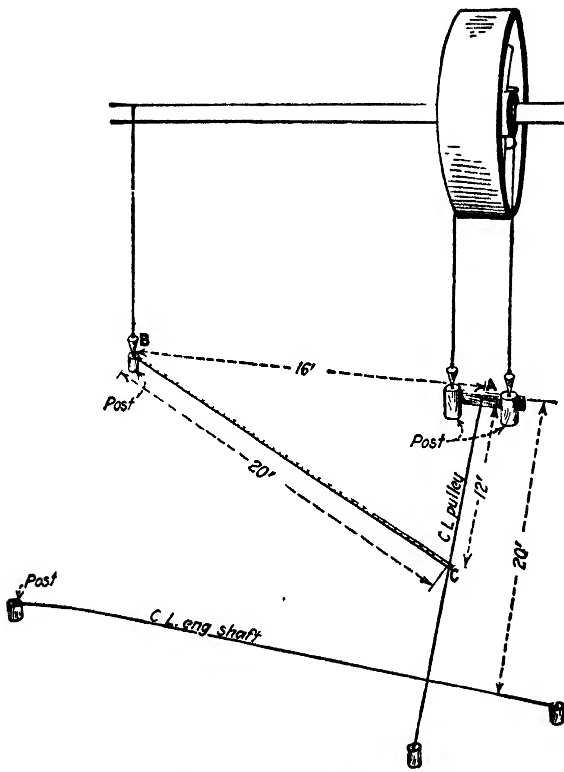


FIG. 92.—One method of locating engine-shaft center line.

right angles to the driven shaft, should be anchored to a post on the opposite side of the foundation site. The desired distance between the engine shaft and driven shaft is next measured, say 20 ft., and a stake or post set in with a center tack. The point thus determined is the intersection of the engine shaft and driving-pulley center lines.

The engine-shaft center line is next obtained by using the 3–4–5 rule. In making the measurements the tape should be held as near horizontal as conditions will allow. Posts or stakes should be set in at some distance from the end of the foundation with center marks so that the line may be fixed for all time. An excellent plan, if the building permits, is to place four center tacks on the ceiling above the shaft and pulley center lines. This enables the engineer to relocate the lines at any time. It is not necessary to attempt to establish an exactly horizontal center line, as this is best done after the engine is placed on the foundation.

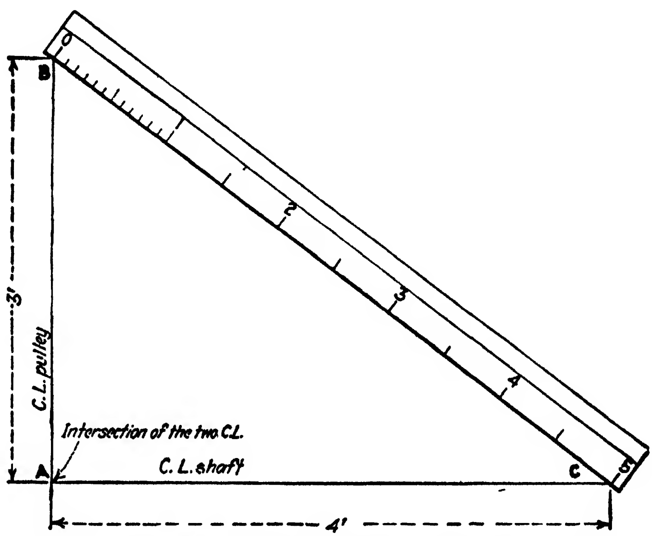


FIG. 93.—Locating lines at right angles by the 3-4-5 method.

Method on Large Engines.—With large engines the proper method is to use an engineer's transit in locating the engine's center line. The transit is set under the existing shaft and sighted along the shaft. Dropping two plumb lines from the shaft makes the locating of the line an easy problem. After the line of sight is established and the datum mark made, the transit is turned 90 deg., and a stake set along the desired engine shaft line. Moving the transit to this latter stake, it is sighted on the datum stake or mark. This establishes a line of sight at right angles to the lineshaft. Turning the transit 90 deg. and driving a stake along the new line of sight gives two points in a line parallel to the datum shaft. After these two points are located, a strong piano wire run through them will indicate the desired engine-shaft center line. The anchorage for the wire ends should be substantial, since workmen in moving material quite often strike it.

Foundation Form.—The foundation drawing showing the crankshaft and pulley-wheel center line is sufficient to enable one to lay out the foundation pit. If the engine is small, say under 50 hp., the dirt walls of the pit are quite suitable for the foundation form. If the dirt has a tendency to cave, rough planking may be placed in the pit, making a fairly good rectangu-

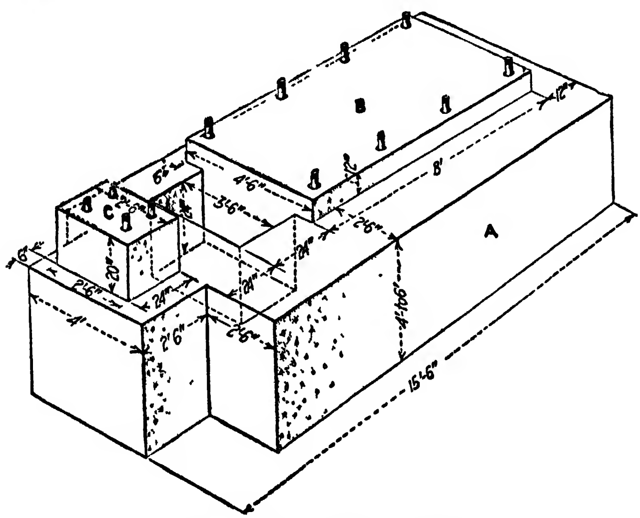


FIG. 94.—Typical engine foundation.

lar form. With larger engines it is advisable to construct a wooden form, for in most cases the foundation is not rectangular but embodies an extension to carry the outboard bearing. In making a foundation it is a good plan to provide a concrete trench, or pipe chase, along one side of the foundation.

This allows all piping to be placed under the floor and yet be accessible. This is much better than the all too common plan of installing the pipe in the floor, permanently embedded in the concrete.

A typical foundation is shown in Fig. 94. At first glance the construction of the form may seem to be difficult. Upon inspection it is found to consist of a rectangular lower portion upon which is set a portion that rises above the floor to carry the engine and the block supporting the outboard bearing.

Frequently, a foundation plan embodies a great number of steps and angles, especially in cases where the draftsman is allowed to indulge his fancies. Some claim that the earth pressure on the steps tends to eliminate vibration. However, the effect is slight, and more is to be gained by employing generous dimensions and a better balancing of the engine. The engineer is advised to make the foundation as nearly rectangular as possible. Often this increases the concrete yardage, but concrete is cheaper than carpenter wages in building complicated forms.

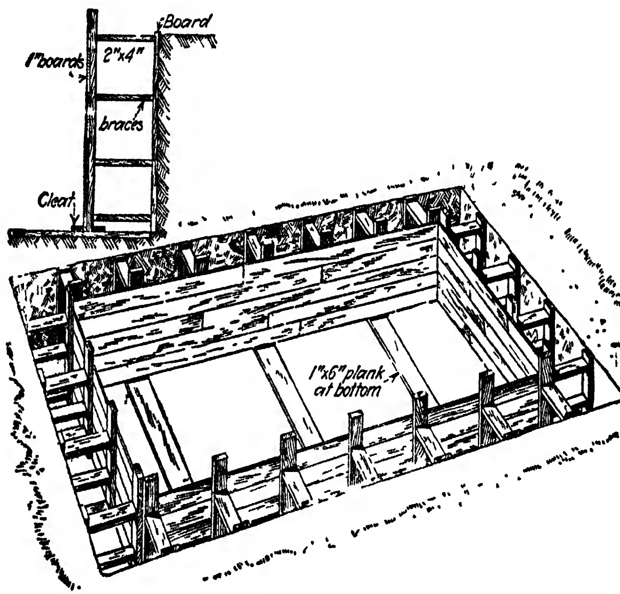


FIG. 95.—Foundation form in place.

The first step is to excavate the pit that is to carry the lower block, this being from 4 to 8 ft. deep, as the size of the engine and the character of the soil may determine. The easiest method is to make the pit rectangular; as an example, take the dimensions given in Fig. 94, 15 ft. 6 in. long by 7 ft. 6 in. wide, to which should be added 18 in. on each side to allow room for the forms and the removal of the planking. This gives over-all dimensions of 18 ft. 6 in. by 10 ft. 6 in. If there is no objection to leaving the form on the foundation after pouring, the allowance at the sides need be but the thickness of the plank plus that of the vertical supports.

Proceeding with the form, 2 by 4 studding long enough to project above the floor level a distance equal to the height of the upper base should be placed against the pit walls at intervals of 30 in., and 1-in. boards, preferably tongued and grooved and finished on the foundation side, nailed to the 4-in. side of the studding. The boards should be cut to extend the entire length of each side of the pit and should be simply tacked with light nails just enough to hold them in place. With the sides made, a 1-in. plank should be laid across the bottom of the pit in line with each 2 by 4, and two cleats nailed at each end. When the sides are straightened, the base of each 2 by 4 rests between a set of cleats holding the bottom of the frame. Planking is next placed against the dirt sides in line with each 2 by 4, and short strips are joined to the plank and 2 by 4 timber. The ends of the form are next made in the same way. The foundation now has the appearance of Fig. 95; at the top of the illustration is a sketch of the method of supporting the form. In addition a few braces should be nailed across the form at the top to brace the frame and to serve as a foundation for the upper base form. These braces must be no thicker than the amount of grouting to be used. The offsets in the lower form are next made by adding a box at each corner.

Engine Templet.—Before the upper foundation form is constructed, the templet for the foundation bolts should be made and placed over the excavation. It is usual for the engine builder to furnish this; but if the contract does not so specify, the engineer can easily make one from the dimensions shown in the foundation plan. For a small unit 6-in. finishing lumber 1 in. thick is acceptable. For larger units there is not enough rigidity in such light material, and 1½ by 8-in. lumber should be used.

Assume that the blueprints show the foundation bolts, engine shaft, and flywheel center lines to be as shown in Fig. 96. Two planks about 12 ft. long should have the engine-frame bolt holes laid out and bored. Cross strips and a center strip at least 16 ft. long should then be laid on a smooth floor, and a line drawn down its center. The longitudinal planks carry the bolt holes and are then nailed to the cross planking, giving a templet as shown in Fig. 96 *A*. It is no great matter to place the planks carrying the engine-bed bolt holes parallel to the shaft center line. In locating the flywheel and pulley-wheel center lines, a steel square may be laid along the shaft center line, and the fly-wheel line drawn at right angles. To check the accuracy the 3–4–5 method may be used. The outboard-bearing bolts are located in the same manner. The templet should be thoroughly

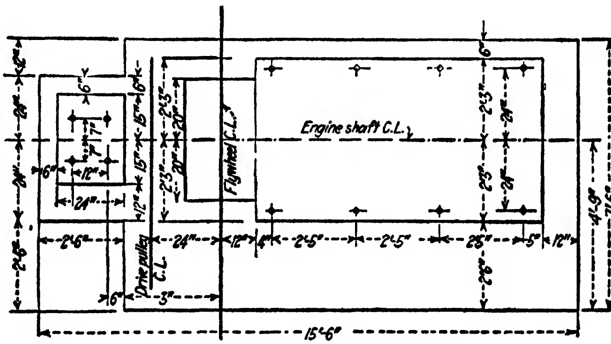


FIG. 96.—Engine foundation plan.

reinforced and, after being placed on the foundation form, should be rechecked for alignment of the center lines.

The templet is often swung from rods or heavy cable suspended from the roof. In this way, if the templet is shifted by accident, it will return to its true position. This method cannot be used in all cases, and the templet may be set on the form. A good

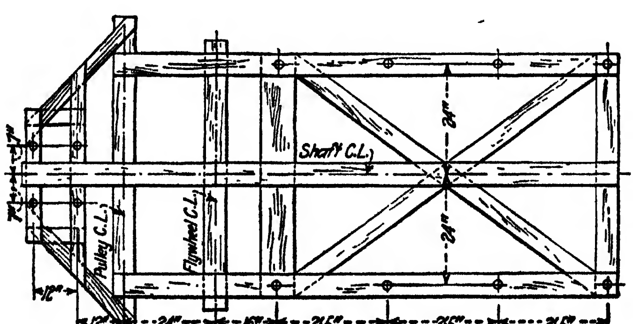


FIG. 96 A.—Making foundation templet.

plan is to drive in heavy posts some distance away from the foundation and suspend the templet from timbering nailed to these posts. This eliminates all danger of misalignment due to shifting of the foundation form. This method is largely followed but is not the best. The one method that is free from criticism is the use of two heavy timbers, about 12 by 12 in., to which are bolted cross stringers of 6 by 6 in. The templet is suspended from this framing by bolts, and, if the ends of the 12 by 12 timbers rest on solid footings, the entire framing is rigid. If the engine is of medium size, 8 by 8 in. will serve for the bearing timbers. Such a structure will not move in the event a barrow of concrete is thrown against it.

With the templet in place the foundation bolts should be hung from it, and it should be lowered or raised until its bottom is on the level of the top of the upper form. The templet should now be fastened rigidly to the supporting posts previously mentioned. It is also necessary to level the templet. This is not difficult, for

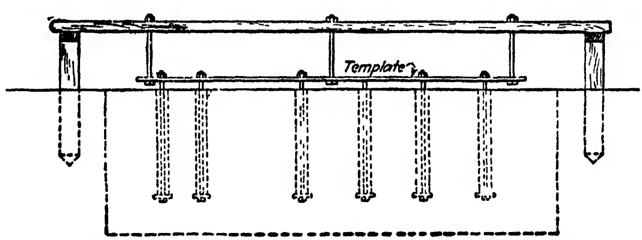


FIG. 97.—Suspending templet over foundation excavation.

by using a spirit level a close approximation of the true level is secured. A variation of ½ in. is not serious, as this can be corrected when leveling the engine bed.

Next in the process is the construction of the upper foundation form and the form for the outboard-bearing support. All that is necessary is to built up an open-bottom, open-top box at the proper place. The cross timbers holding the top of the lower form are used as a basis of the framing for the upper form. No timbering should project into these open boxes, for, if it does, they will be covered with the concrete.

In the matter of the foundation bolts practice varies. It is necessary to have some play in the bolts, for the engine-bed bolt holes are cored and may be slightly out. A convenient tube to place around the bolts in order to have the bolts free to move is 2½- to 3-in. water spouting made of galvanized sheet steel. These are light and less costly than iron pipe. In large engines it is quite common to use such tubes together with recesses at the bottom, access to which is from the side of the foundation base.

This has the advantage of allowing the engine to be moved on to the foundation without the necessity of being lifted to clear the protruding bolts. The bolts are then dropped in, and a nut and washer are placed at the bottom of the bolt as well as at the top. The objection to this method is that it is hard to screw on the lower nuts, since one must work in a very confined position. Preferably, the bolts should be set into the foundation before the latter is poured. Washers, say 6 to 19 in. square, should be placed at the lower end of the bolt.

Foundation Material.—The proportioning of the concrete ingredients varies over a wide range. It depends to a considerable extent upon the caracter of the sand and gravel used. It is obvious that, if the gravel is not washed, the proportion of sand would be less than where a clean gravel was obtainable. For heavy foundations such as the one under discussion, a ratio of 1 part cement, 2 parts sand, and 5 parts gravel or crushed rock is advisable. This mixture possesses ample binding strength and is free from the danger of cracking which develops when a leaner mixture is adopted. In order to lessen the foundation expense, many replace the gravel or crushed stone by broken brick. The objection to the brickbat lies in the danger of serious fractures; furthermore, unless the brick is clean, a good bond with the cement cannot be secured. Upon pouring the conglomerate into the excavation it should be thoroughly rammed, especially at the corners. Enough water should be added to make the mixture "quaky" but not enough to cause the cement to be lost by leakage.

In most installations the process of pouring the foundation extends over a period of several days. Each night the surface should be given a thorough wetting down to prevent any premature setting during the night. If the weather is cold, an old carpet or other covering can be placed over the foundation to prevent freezing. It is seldom necessary to place a foundation in freezing weather; but when such conditions do exist, the water should be warm, and a liberal covering of straw and old canvas placed over the concrete each night. After bringing the foundation to the desired level, the surface should be left in an unfinished condition and dampened each night. This keeps the surface concrete green and allows a good finish coating to be applied after the erection is complete.

Often the foundation print shows no reinforcing steel. With any oil engine there is need of tie bars inserted in the concrete. It is not necessary that special bars be purchased. Old steel rails, discarded I-beams, and the like are just as serviceable. A row of rods or bars laid longitudinally with crossbars at frequent intervals for a reinforcing matting will bind the entire structure. The steel should be laid about halfway from the base.

Vibration.—One of the objections voiced against the installation of a Diesel engine in an office building is the vibration so often present in the internal-combustion engine. There is no adequate defense against this charge, for, as often installed, an oil engine sets up vibrations that can be felt even in large buildings.

In preparing the foundation for these installations a layer of cork at least 3 in. thick should be placed over the entire bottom of the foundation excavation. Special cork blankets are now available for this application. A concrete retaining wall 6 in. thick should be built about the foundation. This serves to keep the earth from touching the foundation. A wooden form for the foundation is then placed within this retaining wall. The form can be made of 2 by 4-in. studding and 1 by 12-in. rough boards. The 2 by 4's should not touch the retaining walls but should be supported by wedges. After the concrete is in, the wedges can be removed; this will allow the wooden forms to be dismantled. The distance between foundation and retaining wall ought not to be less than 8 in. If a narrow space is left, it should be filled with asphaltum. With this construction the foundation is not bound in any way to the building, and the layer of felt will absorb all the shocks incident to the engine's operation.

Since the horizontal engines are, as a rule, more inclined to set up longitudinal vibrations, due to the horizontal thrust, more massive foundations are required than for a vertical engine of the same number of cylinders.

With any concrete foundation, after the engine is erected a heavy coating of waterproofing cement mixture makes an ideal finish. This coating will serve to keep any oil from seeping into the concrete.

Springs under the foundation, known as "vibro-dampners," are in general use as vibration isolation. The frequency of the springs is different from the enforced frequency of the engine, so the engine vibrations are killed. Such a spring system appears in Fig. 98.

Unloading Engine.—An engine is usually shipped on a flatcar, although at times flat-bottom coal cars are called into service. If the latter, one end must be removed in order to get the engine off the car.

Various methods are followed by erecting engineers, the particular one used depending upon the plant location and other considerations. It is usually feasible to have the car set so that one end is accessible. A good plan of unloading is to construct a runway leading from one end of the car to the ground, at an angle of not over 20 deg.; a steeper incline causes the engine to become unmanageable.

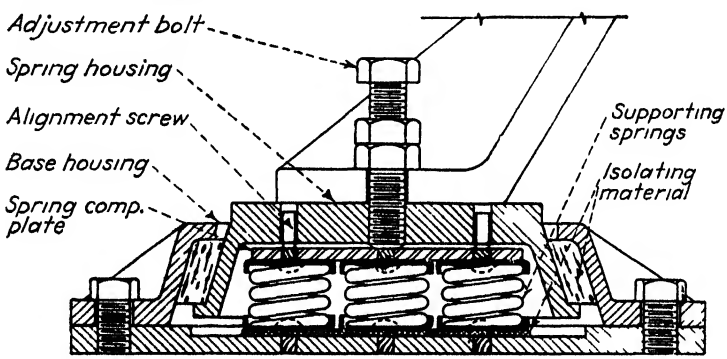


Fig. 98.—Springs used to dampen engine vibration

The engine frame is lifted with ratchet or gear jacks and 6-in. maple rollers placed between the skids and car floor. Since the car flooring is laid crosswise, 2 by 12 planking should be placed lengthwise of the car for the rollers to rest on. A 1½-in. rope-tackle block should next be fastened to the frame at some convenient place, such as around the bearing housings, and the second block chained to the railroad track or drawbar. The rollers support the frame with ease, and men can roll the latter to the end of the car with little trouble after getting it started by using a couple of jacks tilted against the skids at the back end. When the engine frame projects over the car floor enough to tip, the tackle block, already mentioned, should be tightened so that the front of the frame will come down on a roller without jar. By easing off on the tackle the engine will roll down the incline without hitch, rollers being placed under the skids at intervals of about 3 ft. If maple rollers are not at hand, 3- or 4-in. pipe will do.

To move the engine onto the foundation, a runway of planking will make a good path for the rollers. Upon arrival at the foundation, the frame may be jacked up until the rollers can be moved onto the foundation, or the runway inclined so that the rollers carry the engine onto the foundation.

Lowering Engine on Foundation.—As soon as the frame is in the proper position, the skids are removed. By using jacks at three or four points the frame may be lowered on to a set of wedges. These wedges should not exceed 1 in. in thickness at the heavy end and should be 3 in. wide by about 8 in. long. A jack will seldom lower closer to its footing than 2 in., and so it is often necessary to make a recess in the corners of the foundation top into which the jack is set. The hole need not be over 2 in. deep if step, or, as they are usually called, "Barrett railroad," jacks are used. These will usually raise 5 tons without danger of breaking.

For heavier frames, geared or hydraulic jacks should be used. These will lower to about 2 in. of the bottom. The geared type of jack is to be preferred, since a man can lift 10 tons with ease. Unless the frame weighs more than 10 tons, three jacks should be used, as this allows the leveling to be accomplished more easily than with four or more jacks.

In many of the vertical engines the frame rests on a base that is by no means difficult to handle. This bed, or subbase, must be placed on the foundation and leveled before the engine frame proper is set.

Leveling Engine Bed.—Many oil engines are provided with adjusting screws along the lower edge of the base. In leveling up, iron plates about 4 by 4 in. are set under these screws. It is well to level the plates, grout them in, and allow the cement to harden before the screws are brought down against the plates. If soda is mixed with the cement, the grouting will harden in 10 min. By manipulation of the screws the frame is easily made level.

If the engine is not large, it is often leveled by placing a spirit level across the planed tops of the bearing housing and lengthwise on the housing. This will give a very close approximation of the true position.

If the engine is to be belted to a machine or lineshaft, the erector must be careful that the engine-shaft center line lies along the center line established at the time of putting in the foundation, when posts with center tacks were set in to establish both the shaft and the pulley center lines.

In all probability these posts are too low to allow a line to pass over the frame level with the top of the posts. Under these conditions center-line posts should be set in as close behind the first posts as possible without disturbing the latter. These posts should be high enough to permit the stretching of the wire above the engine, clearing the frame by, say, 20 in. The approximate location is easily determined. A wire should now be stretched between the two posts in a level position. This wire should be in the same vertical plane as the wire originally set up. The dropping of a plumb bob will make this easy.

The leveling of this wire can be done with a spirit level. A good method of attaching the two wire ends for leveling is to place a smooth piece of tin over the top of one post so that the wire can be shifted sideways or vertically. On the other post erect two substantial strips. To this end of the wire is attached a circular piece of tin; to the other is fastened a 30-lb. weight, which is hung over the first post. The weight gives a uniform tension to the wire and brings the wire back to its position if anything displaces it. The tin washer on the second end, by being placed behind the two strips, may be raised or lowered or shifted slightly sidewise. This will allow the wire to be placed in the exact vertical plane. In addition the tin washer may be raised or lowered to bring the wire level. The leveling of the center-line wire is done by placing the spirit level along the wire. One cannot hold the tool level without some support, such as a frame of planking or blocks. The level may be brought up by wedging, the wire's position checked, and the tin disk tacked into place.

Establishing Center Line.—After this wire is put up, two wires should be laid at right angles to the lineshaft which is to be driven by the engine by the well-known 3–4–5 rule, and the distance from the driven shaft to the shaft center-line wire measured for the purpose of checking the position. A steel tape should be used, as a linen tape is unsafe, stretching as much as 1 in. in 25 ft. The pulley center-line wire should be stretched if it has not already been established. Often the position of the pulley is marked on the shaft; if not, the position is indicated by the keyway.

Dropping a bob to the pulley center line will show how much the engine must be shifted sidewise to come into the correct position. To bring the engine shaft exactly under the shaft center-line wire, erected as noted above, putty should be hammered into the shaft-line centers, and a small mark made in the exact centers. This is found by calipering the shaft. If plumb bobs are dropped from the center wire, the erecter will be able to shift the engine the slight amount needed to bring these centers square with the bobs. Since the weight will always bring the wire back to its level position, the bobs may for convenience be hung from the wire. A good erector, with the foundation

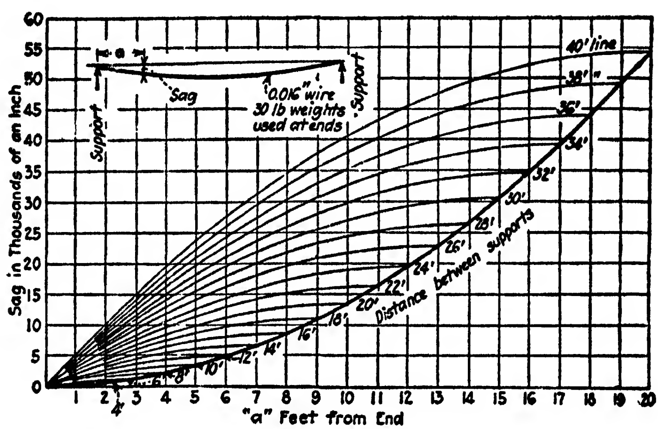


Fig. 99.—Deflection of steel wire.

bolts to guide him, should have the engine in a position that will require but little shifting. The next step is the leveling of the frame. Since the shaft is bedded in the bearings in its proper place, all that is necessary is to level the two ends of the shaft and then level the frame lengthwise. By measuring from the wire to the shaft with a pin gage, the latter can be brought level with the wire. If the engine is of any size, it is well to remember that the long wire will not be exactly level, having a small sag in it. Figure 99 is a chart showing the allowance that should be made for a 0.16 wire with different lengths between posts.

If an engineer's level is at hand, it may be set up at a point somewhere back of the engine. The instrument is then sighted at a rod placed vertically on one end of the engine shaft. A mark, cut by the hairline of the engineer's level, is made on the rod.

Next, the engineer's level is swung so that it will be in line with the mark when the rod is held on the opposite end of the shaft. One end of the frame must be raised or lowered until the mark again cuts the crosshair. The leveling should be rechecked at least once. It is seldom, though, that such an instrument is available, although for engines over 500 hp. it should be used.

Leveling Small Engines.—A customary method, which, although apparently crude, is often followed successfully on small engines, is the leveling by means of a spirit level with a groove in one surface. This is placed on the engine shaft and, by the cross bubble, brought up in the proper position. Longitudinal alignment of the shaft is then indicated by the second bubble.

Leveling Large Engines.—If an engine is relatively large, say over 16 ft. long, the best plan of leveling the base is by the crankshaft alignment method.

When an engine is received from the builder it may be accepted as a truism that, when the subbase is correctly installed, the shaft will be in alignment. Notice the qualification "correctly installed."

To bring the base to approximate level, scrape the paint or slush from the machined top surfaces of the bearing housings. Place the spirit level on these surfaces, both lengthwise and crosswise. If the base is not level, drive in wedges at the low section until the spirit-level bubble "registers" when placed at all positions.

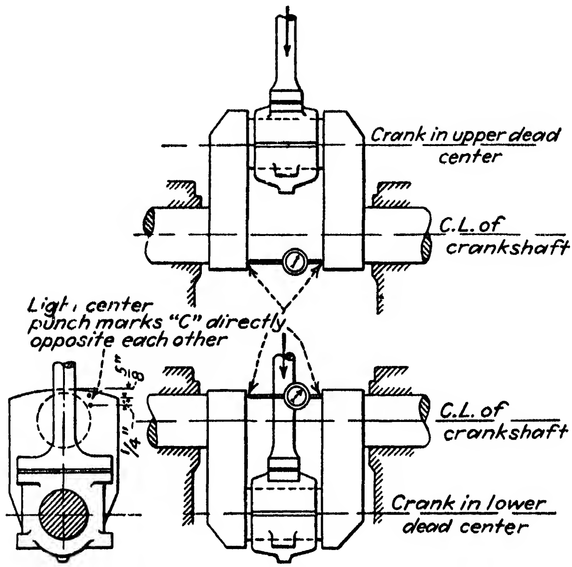
While doing this, the flywheel should not be on the crankshaft, for its weight may influence the later alignment of the shaft.

On an engine with a relatively short subbase, placing a spirit level lengthwise and crosswise on its machined housing tops is a satisfactory system because the boring bar has bored all the bearing housings in alignment. But if the engine is long, it is not enough merely to use a spirit level.

As will be pointed out in the chapter on crankshafts, alignment of bearings was obtained originally at the builder's shop by scraping. Consequently, the machined flat surfaces may not quite check.

In addition, it is possible for a long base to be high at one point, yet the spirit level might show "level" at all bearings. It is far better to employ a gap gage.

The correct way to use a gap gage is as follows: Move the engine subbase upon the foundation, and insert wedges along the edge. Remove the bearing caps; and by placing a spirit level on the machined top surfaces of the bearing housings, level the base as accurately as possible. Then place the shaft in position;



MEASUREMENTS BETWEEN CRANK WEBS CYLINDERS								
Number of Cylinder	1	2	3	4	5	6	7	8
Crank in Top Position A								
Crank in Bottom Position B								
Difference Top and Bottom								
Crank in Front Position C.L. of Crank Horiz. C								
Crank in Rear Position C.L. of Crank Horiz. D								
Difference Front and Rear								

FIG. 99 *A*.—Measuring crank-gap clearances.

and placing the caps back into place, tighten the bolts slightly, to hold the shaft against the lower bearing shell.

Using a center punch, place two small punch marks on the faces of these crank throws, as indicated in Fig. 99 *A*.

Turn the shaft until one crankpin is vertical, as shown in Fig. 99 *A*. Place the gap gage between the throws, with the pointed ends resting in the two punch marks mentioned. Jot down the dial reading. Next loosen the bearing-cap bolts sufficiently to allow the shaft to be rotated 90 deg. Again tighten the bearing caps, and jot down the gage reading in the line marked *C*, crank at 90 deg., in the column for No. 1 crank. Proceed to turn shaft until the crank is vertical below the shaft, as shown in broken lines in Fig. 99 *A*. Again read gage, jotting down the figure in the line marked *B*, crank at 180 deg. Continue to turn shaft until it is at right angles on the opposite side, marking the gage reading in line *D*, crank at 270 deg.

Follow the same process for all cranks.

By subtracting value *A* from *B*, the difference in gage reading at top and bottom dead center of a crank is obtained. One should expect to find the top-position reading slightly greater than the bottom-position reading, but a difference of over, say, 0.003 in. should prompt the erector or engineer to adjust the wedges under the frame until the readings are satisfactory.

Difference in the 90- and 270-deg. readings (the horizontal positions of the crank) will prompt the erector to shift one end of the base horizontally until the gage readings check.

With the base and shaft aligned as outlined above, grouting and tightening the foundation bolts will hold the frame securely in place.

Some engine builders advocate loading the piston of the crank being checked, by admitting starting air into the cylinder. It is recommended that, starting with No. 1 cylinder (next to the flywheel), all the valve levers be removed from the cylinder head, to prevent valve action.

Install a pipe tee between the indicator cock and the cylinder. Connect a pressure gage to a side outlet of the tee, and connect the air hose to the indicator valve. Locate No. 1 crank on top dead center. Chuck or block flywheel so that it cannot turn. Admit air to cylinder under a pressure of 75 lb. during the entire period of measurement determinations. Delegate one operator to maintain this pressure. When air is admitted to the cylinder under these conditions, the crankshaft is forced to a firm seat in the lower half of main bearings adjacent to this cylinder.

Many engine erectors feel that the air-pressure method entails a lot of time and so prefer jacking the shaft to the bearings. With this arrangement, the bearing caps are removed, and a screw jack is placed between the frame crossweb and the bearing. Preferably a block of wood should be placed on the shaft, so that there will be no danger of marring the journal surface.

A particular crank is placed on top dead center, and jacks placed on the two adjacent journals force the shaft against the lower shells of these two bearings. The gap gage reading is recorded as already outlined. The jacks are then loosened, and the shaft is turned to bottom dead center. After again tightening the jacks, the new gage reading is set down on the tabulation.

Some may not see the necessity for the jacks, feeling that the shaft will seat in all bearings even if one is low. The fact is, however, that modern shafts are made so large in diameter, to avoid torsional vibration, that the shaft will not by its own weight bend sufficiently to bed itself in low bearings.

Grouting the Engine.—Before running the grouting, boards should be placed around the foundation, extending above the surface the desired depth of the grouting. A mixture of one part sand and one part cement should be thoroughly mixed with enough water to make it flow readily. This is poured in around the base and troweled into the center of the foundation. The space between the bolt and iron piping should also be filled. Each bolt hole should be filled completely, and the ordinary laborer should be watched to prevent him from covering the top of the foundation before fully filling the holes.

Enough grouting should be poured in to fill the space up to the top of the boards already mentioned. As the cement sets, the boards may be removed, and the grouting troweled down at a bevel. After this has set for several days, the foundation bolt nuts should be drawn up tight, and the iron wedges knocked loose, or, in case of screws, these should be backed off a little.

Putting on the Cylinders.—Continuing on the erection, if the engine is large, the cylinders may be shipped separately from the frame. In such cases the first step is that the cylinders are placed on the base. If an I-beam trolley and a suitable chain hoist are in the plant, the handling of the cylinders is easy. Since the cylinders must be centered over the studs on the top of the base, they must be held exactly vertical by the hoist to prevent jamming on the studs. The author recommends that a 6 by 6 timber be cut long enough to extend over the cylinder top.

Holes large enough to accommodate the cylinder-head studs are bored in this timber, the latter slipped over two opposite studs, and the nuts screwed on. By wrapping a chain around the center of the timber and engaging the chain in the hoist hook, the cylinder can be lowered over the studs without trouble. This is shown in Fig. 99 *B*. Every plant should be equipped with an I-beam trolley and differential hoist. This hoist should be of at least 5-ton capacity. The I-beams can rest on brick pediments that are incorporated in the building walls. In case such equipment is not available, it becomes necessary to construct a wooden frame. The top beam should clear the frame by at least 10 ft. to give ample room for the blocks. If timbers of sufficient length are not obtainable, the frame can be made of built-up timbers constructed of six planks 2 by 2 in. in size, lag-screwed together. Stout planking should be placed along the side of the engine to prevent the cylinder from damaging the frame as it is hoisted to the top; 6 by 6 cross timbers can be laid across the top of the frame to act as a landing for the cylinder. These timbers guard the studs against damage. The tackle blocks, or chain hoists, which should have at least 5-ton lifting capacity, are fastened to the cylinder top, and it is raised until it can be lowered onto the 6 by 6 platform. It is always advisable to use two blocks. This is a safety measure in case one breaks. After the cylinder is placed on the platform it should be pinched into place over the studs. All the timbers, save two outside the bolt circle, can be removed. By lifting the cylinder with the blocks, which have been centered above the cylinder, the two timbers are withdrawn, and the cylinder lowered onto the frame.

After the cylinder is in place, a plumb bob is dropped through it. To drop it, a metal or wood strip is fastened to one of the cylinder-head studs. A washer is attached to the bob line after the line has been passed through the slot in the strip. The strip is placed over the approximate cylinder center, and the slot allows the bob line to be moved until it is exactly center. To center the plumb line at the top of the cylinder, a pair of inside calipers are used and this centering calls for the greatest patience on the part of the erector. After centering the line at the top, measurements must be taken at the bottom of the cylinder to ascertain if the bob line is central at this place. Since both the cylinder flange and the frame top are machined true, it is seldom that the cylinder does prove out of plumb, provided the frame has been properly set. If this should occur, the frame must be wedged up until the cylinder center line is plumb. If the engine is multi-cylindered, a plumb is dropped through each cylinder and kept in place until the engine shaft is lined up (Fig. 100). The

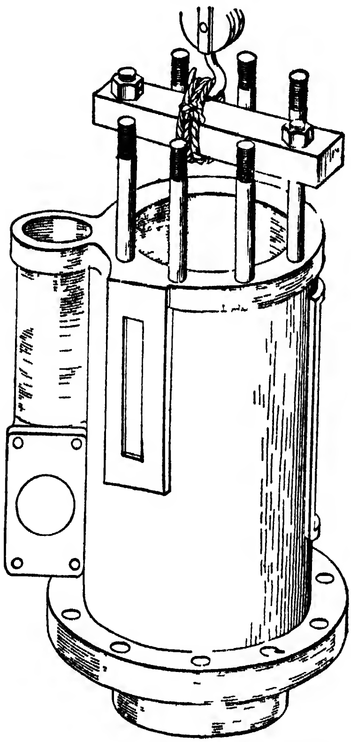


FIG 99 *B* —Method of handling cylinders

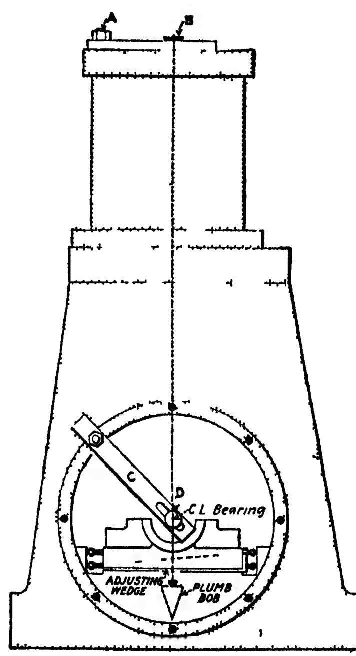


FIG 100.—Aligning cylinder and shaft.

bob lines must square with the shaft center line. Practically the same method can be followed with the A-frame engine. Many engineers do not check the cylinder alignment, feeling that the factory has already done this and has properly doweled the castings in place.

Centering the Shaft.—It is necessary to check the crankshaft bearings to insure that the shaft will be square with the cylinder center lines, especially in case of used or reconditioned units. Two metal or wood strips should be fastened to the ends of the engine frame, as in Fig. 100, with a piano wire stretched between. The disks should be moved until the horizontal wire just touches the three cylinder plumb lines; the wire should be leveled at the same time with the spirit level. The bearings, which have been placed in their housings, should be calipered to see if the line is central with each bearing. If a bearing is out, then it must be shifted until it registers central. With the wedge-type bearing this alteration can be obtained by proper movement of the wedges. If the bearings are of the non-adjustable shell type, it becomes necessary to scrape the high bearing and shim up the low ones. In most engines the bearing housings are bored in alignment, and it is necessary only to see that the bearing shells are of the same thickness. This makes each bearing in alignment, and the cylinder lines must check if the cylinders have been properly fitted.

Bedding the Shaft.—Before the shaft is lowered onto the bottom bearing shells, small lead wires, about $\frac{1}{32}$ in. in diameter, should be placed along the bottom of each bearing. When the shaft is placed in position these wires flatten out. By raising the shaft these wires can be removed, and their thickness measured by a micrometer. If the lead is of the same thickness throughout its length, the shaft evidently bears evenly in a longitudinal direction. If any unevenness is present, the shell must be scraped to a fit. To check the area of the shell that is supporting the shaft, the latter can be coated with Prussian blue and rotated. Upon lifting the shaft the bearing will show which points are high or low. Scraping the babbitt will bring the surface to a perfect contact.

Many erectors have the habit of using a chain sling on the shaft. This is absolutely out of place with a Diesel engine. Sacking wrapped about the shaft and the use of rope slings will eliminate all danger of scoring the shaft. It requires only a small dent or cut on the journal to ruin a bearing. In handling the shaft it is advisable to suspend it by slings from each end; if possible, the slings should be tied on the shaft at other than journal joints.

Aligning Outboard Bearing.—With an engine employing an outboard bearing the alignment of the extension shaft merely necessitates the continuation of the shaft center line. In erection, the main point that demands the exercise of extreme care is the bolting of the extension shaft to the main shaft or to the flywheel, as the design may be. The flange bolts should be drawn up uniformly, a part turn of each nut in succession. Some erectors are in the habit of bolting the two parts together with the outer end unsupported by the outboard bearing. They then turn the engine over and adjust the outer bearing against the shaft end. The danger here lies in the liability of the unsupported shaft weight throwing it out of level. Many hot outer bearings can be attributed to this carelessness.

In leveling the extension shaft and outboard bearing, erectors follow various methods. A good plan is to raise the outboard bearing until the shaft no longer touches the main bearing next to the flywheel. This journal is coated with Prussian blue, and the outer bearing is then lowered until the blue is rubbed off by the lower shell when the shaft is rotated a half turn or so.

Putting on the Flywheel.—The work of putting the flywheel in position is often difficult, because of the building arrangement. On the basis, first, that the unit is a belted machine, the extended shaft is set in place and the out bearing aligned, after which the shaft is removed. One-half of the flywheel should next be skidded to the side of the foundation and in front of the flywheel pit. The half is then upended, and by attaching rope slings to the arms the casting is lifted and swung over the pit. Cross blocking is placed in the pit, and the wheel half is lowered until the hub is approximately in position.

The shaft extension is next swung into place. If room is available, it may be shifted lengthwise into its position. Unfortunately, the usual plant has little room at the end of the engine, and the shaft must be rolled along the engine until it is opposite the flywheel. It may then be lifted enough to clear the flywheel resting in the pit, and after cross timbers are placed under the shaft the latter may be rolled into position. With the shaft supported by the hoist, the cribbing is removed until the shaft half of the coupling fits into the engine coupling half, and the shaft end rests in its bearing. Before tightening the coupling, both the alignments of the shaft should be checked. Some engineers merely shift the bearing until the extended shaft shows no sign of wobbling, but this is by no means good practice.

Installing the Generator.—With a unit direct connected to a generator the procedure follows the same general lines. The shaft, however, must be placed high enough above its true position to enable the generator frame to be moved onto the base or rails. This requires the blocking of the shaft and rotor in such a way that the generator can be shifted over it. As soon as the shaft end is exposed, blocking should be placed under it. When the generator is moved up to the position of the rotor, the shaft should be shifted until the generator will pass over the rotor without cramping. To do this, blocking should be placed in the generator pit until the generator can be moved over the pit. Of course, if the generator sets level with the floor or above it, there is no difficulty, for the shaft may be placed in exact position before the generator is moved. When the generator sets in a pit, the lowering of the frame, shaft, and rotor at the same time is somewhat difficult, but there is enough air gap between the rotor and frame to allow a little leeway.

The best plan is to run down the leveling screws on the generator feet until they support the generator. Part of the blocking under the generator is then removed. The shaft is supported by four screw jacks placed on the cribbing, each pair of jacks, supporting a block upon which the shaft rests, being kept from rolling by cleats. The whole may now be lowered until the generator rests on its sole plates, the shaft coming down into place. The shaft must be reblocked to allow the end jacks to be moved, and the outboard bearing slipped into place. The shaft coupling should be connected, and the alignment of the shaft and outboard bearing checked.

The next step after bolting up the coupling, fastening the out bearing, and centering the generator is to put on the top half of the flywheel.

The lower half is lifted into contact with the shaft at the hub and blocked into place. The top half is moved opposite the wheel pit, upended, and a slide placed under it. The casting is next lifted by jacks, and a cribbing built under it. It is then moved into place over the shaft and lowered until the hub rests upon the shaft in line with the bottom half of the wheel. The hub bolts should now be drawn up as tight as possible.

Putting in Wheel Links.—The rim links should next be heated in a forge or like place. When at a dull red heat, the links are slipped into the recesses in the rim and driven home. As soon as they cool, the rim halves are drawn together. It will be found that the hub bolts are now slack, and it is often impossible to draw them tight. The best scheme is to remove the bolts, and after warming the bolts and nuts the nut may be drawn up sufficiently to make the bolts tight when the latter shrink upon cooling. Both bolts should be drawn up at the same time to prevent tightening the hub on one side only.

Installing Parts.—After the engine frame and generator are in place the pistons and connecting rods are bolted up. If the engine is direct from the factory, the correct amount of shims for running conditions are already between the bearing halves. Likewise the babbitt shells have been scraped and bedded in. Old engines or engines being overhauled must have the bearings examined and fitted to their respective pins or journals. After some years of service, bearings, especially the main shaft bearings, take on a hard skin or surface, which is removed with difficulty. This is more often found in engines where a splash system of lubrication is used The fine carbon apparently gets into the bearings and settles, cutting the journals in some instances. The proper running clearances, etc., are taken up in the chapter on bearings.

The installation of the several valves and cages is comparatively easy. The chief point is to see that the ground or gasketed joints are in good condition and that the cages are not distorted by sledging up the holding down bolts.

The various parts and accessories are discussed in later chapters.

CHAPTER IX

DIESEL ENGINE FRAMES

Horizontal Frames.—At present no Diesel engines are built in America with horizontal frames, with the exception of a few relatively small two-cycle designs. Early in the Diesel development, from 1912 to 1917, several builders made use of the horizontal frame. Among these early horizontal engines were the Natural Transit, the Snow, the McEwen, the Standard, and the De La Vergne. At present most of the horizontal oil engines are of the low-pressure, or semi-Diesel, design, and these builders

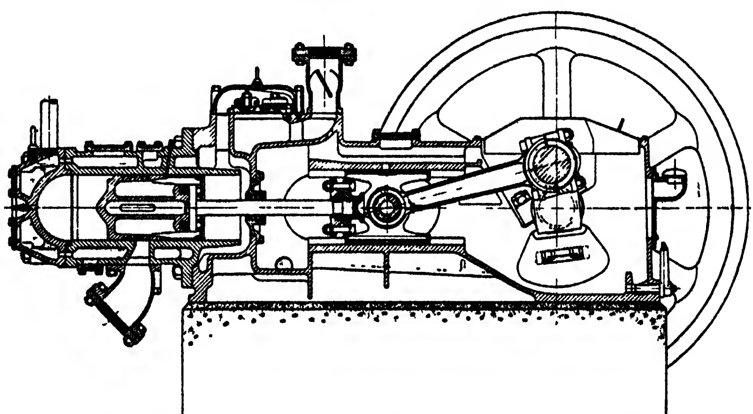


Fig. 101.—Section through Reid horizontal two-cycle Diesel frame.

usually adopt the vertical frame when increasing the cylinder compression to the cold-starting point.

Reid Horizontal Frame.—The Diesel built by the Joseph Reid Gas Engine Company (Fig. 101) employs a horizontal frame which embodies guides for the crosshead and which extends forward with an enlarged section. The cylinder end slips into this section. A partition separates the frame from the inner part of the cylinder, and the space thus formed serves as the compressor for the scavenging air.

Vertical Frames.—Present-day vertical Diesels use one of three frames: the en-bloc frame; the box frame; and the AA, or what might be termed the "girder," frame. The latter type of engine has two cast-iron vertical sections carrying a cross box at the top, which supports each cylinder or, in some designs, forms the cylinder jacket and receives the liner.

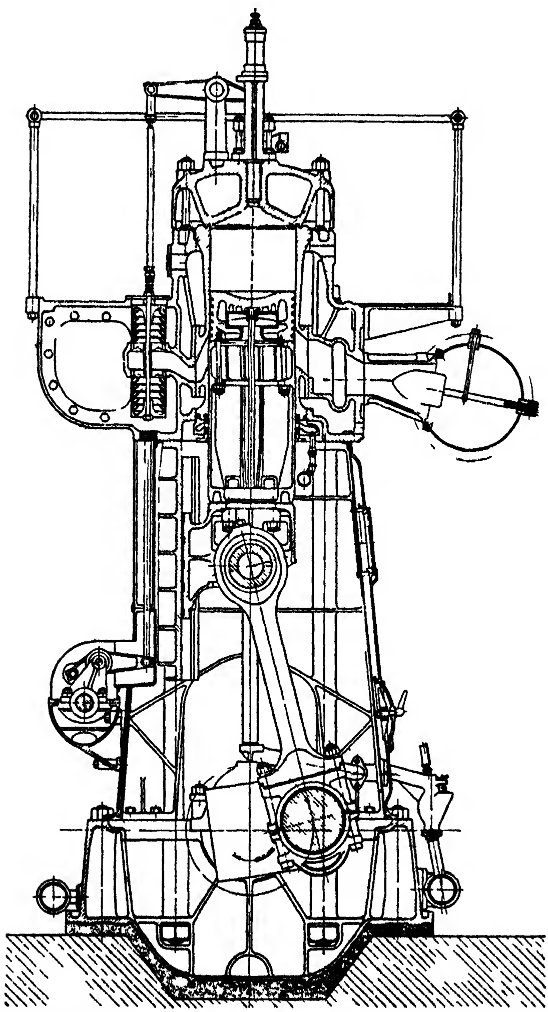


FIG. 102.—Section through a Nordberg crosshead-type, two-cycle frame

The box frame is the most popular in units from 200 up to 1,500-hp. rating, as it is easier to cast and machine, easier to assemble, and very accessible.

The en-bloc frame is generally the design adopted for high-speed engines. Such frames are shown in Figs. 72 to 86.

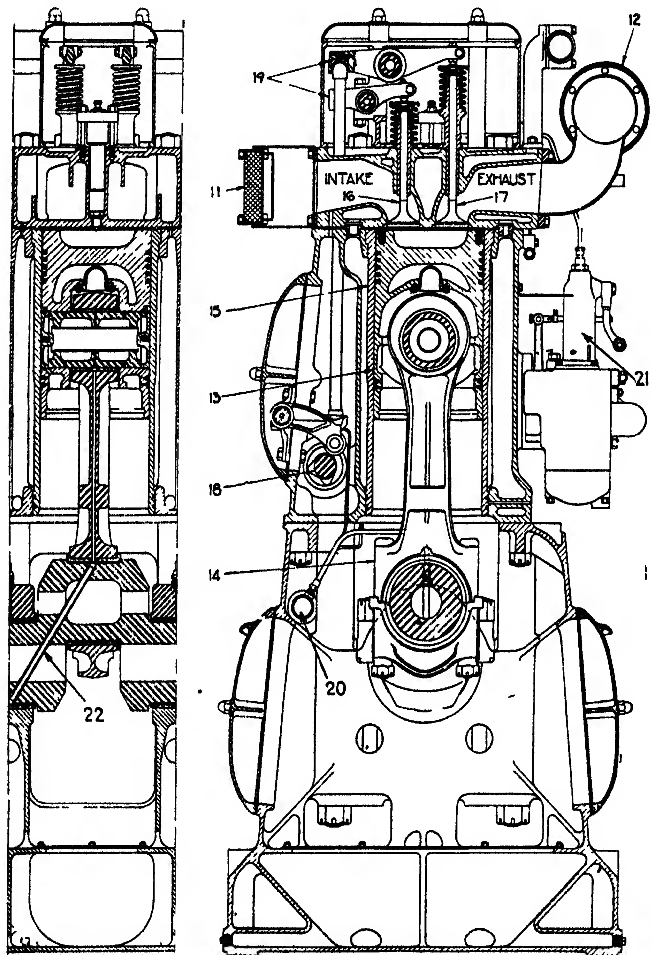


FIG. 103.—Cross sections through the Alco four-cycle Diesel. 11 = air intake filter; 12 = exhaust manifold; 13 = piston; 14 = connecting rod; 15 = liner: 16 = intake valves; 17 = exhaust valves; 18 = camshaft; 19 = valve rockers; 20 = lubricating-oil header; 21 = Bosch fuel pump; 22 = lubricating tube in shaft.

For medium-speed Diesels, the same en-bloc frame is generally accepted, although in such cases the frame does not carry the crankshaft underslung; the shaft rests on a separate subbase.

A typical AA-frame is that employed in the Nordberg two-cycle crosshead Diesel (Fig. 102). The frame is made up of a series of standards resting upon the subbase. Each cylinder rests upon two of these standards, long through rods holding the parts together.

Alco Four-cycle Frame.—The American Locomotive Company employs a two-piece construction in the frame of its four-cycle, mechanical-injection Diesel. As shown in Fig. 103, the crank-case includes the subbase. On this crankcase is mounted the cylinder block. This forms the cooling-water jacket, and removable liners are slipped into this box, as shown. The crankshaft rests on cross members of the crankcase and must be slipped longitudinally out the end of the frame in event of renewal.

Venn-Severin Frame.—The Venn-Severin two-cycle, crank-case-scavenging Diesel has a subbase, or lower crankcase; a crankcase; and separate cylinder castings, (Fig. 104).

Atlas-Imperial Frame.—For its industrial-type Diesel, the Atlas-Imperial Company employs a modified en-bloc frame. This rests on a base and extends up to form the cylinder jackets. Removable cylinder liners are inserted in recesses in this upper section. The design appears in Fig. 55.

Cummins Diesel Frame.—The Cummins Diesel was designed basically for truck applications, and it is not surprising that the framing follows automotive gasoline-engine practice. The frame supports an underslung crankshaft, and the open bottom is closed by a steel oil pan.

Caterpillar Frame.—The Caterpillar Diesel also is fitted with an en-bloc frame, as a study of Fig. 89 *C* will reveal. The bottom of the crankcase is closed by a cast-iron oil pan, or sump.

Witte Diesel Frame.—The small vertical, four-cycle Witte Diesel (5 to 10 hp.) is of single-cylinder construction, and the crankcase extends up to form the jacket of the cylinder. A removable cylinder liner is used. The crankcase rests on a sub-base, which houses the fuel and lubricating oil tanks.

Worthington Frame.—A cross section of a four-cycle mechanical-injection Worthington Diesel is shown in Fig. 104 *A*. The frame is of a single casting, from the base to the cylinder heads. Into this block replaceable cylinder liners are slipped. In the largest units the cylinder block is separated from the crankcase.

Buckeye Frame.—As Fig. 105 shows, the Buckeye, four-cycle, mechanical-injection Diesel employs a one-piece, or en-bloc, frame. This is held to the subbase by long through bolts.

Cooper-Bessemer Frame.—Although details differ in the various Cooper-Bessemer frame constructions, the Type EN Cooper-Bessemer, four-cycle Diesel has a crankcase (or center frame) and a cylinder block (Fig. 106)

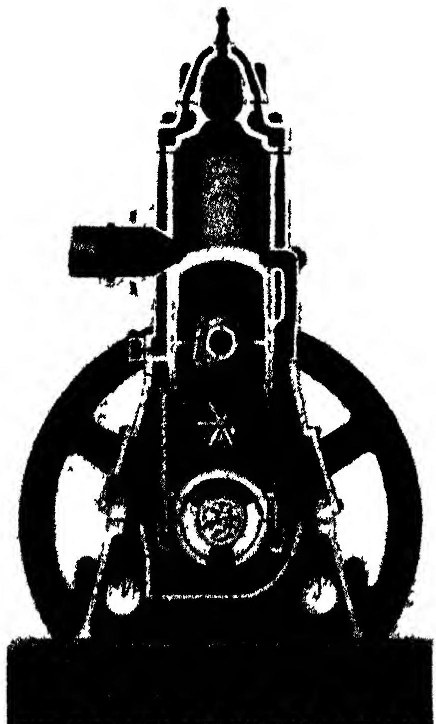


FIG. 104 – Section through Venn-Severin two-cycle Diesel

The base and center frame are cast in one piece of either semi-steel or aluminum, depending upon the service and weight requirements. Each main bearing is supported by a trussed cross section. An oil trough cast full length of the base collects all lubricating oil where it can be quickly drained or pumped to an outside sump tank.

The cylinder block is a symmetrical, well-ribbed, separate casting designed to give abundant strength with the least weight.

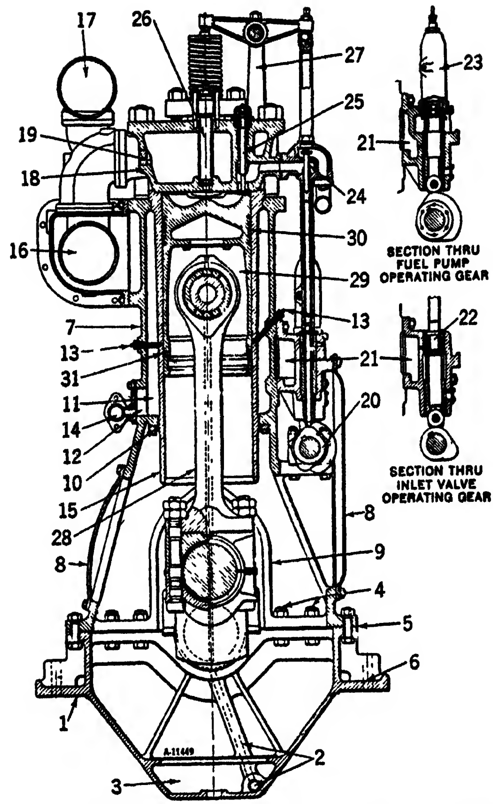


Fig. 104 A.—Section through the frame of a Worthington Diesel. 1 = base arranged for concrete, steel, or timber foundation; 2 = cast-in lubricating-oil passage insures against damage by oil-pipe failures; 3 = lubricating oil sump separated from crankcase by screens 4 = bolts holding frame to base; 5 = bolts holding frame to base; 6 = raised lip to keep oil from foundation; 7 = frame in one rigid block from base to cylinder heads; 8 = doors in front and rear; 9 = cross webs of frame; 10 = individually controlled water circulation around each cylinder liner; 11 = cylinder jacket; 12 = cleaning openings, allowing access to liner and jacket; 13 = cylinder lube-oil feed; 14 = water header; 15 = removable cylinder liner with metal-to-metal joint, secured on top of frame only; 16 = water-cooled exhaust header; 17 = air intake manifold for outside connection or air filter; 18 = cylinder head; 19 = plug; 20 = camshaft in horizontally adjustable bearings; 21 = pump-filled, lube-oil reservoir for tappet lubrication; 22 = spring-supported pushrod; 23 = individual fuel pumps operating without pushrods or levers; 24 = camshaft-controlled, air-starting pilot valve for each cylinder; 25 = air check valve; 26 = fuel spray valve; 27 = rocker-arm shaft; 28 = connecting rod; 29 = piston; 30 = piston rings.

It carries the removable power-cylinder liners, with top and bottom joints sealed watertight by round rubber grommets held in turned recesses. This construction permits the liners to follow the cylinder heads, allows for expansion due to increased temperatures, yet insures watertight joints between cylinder liners and block.

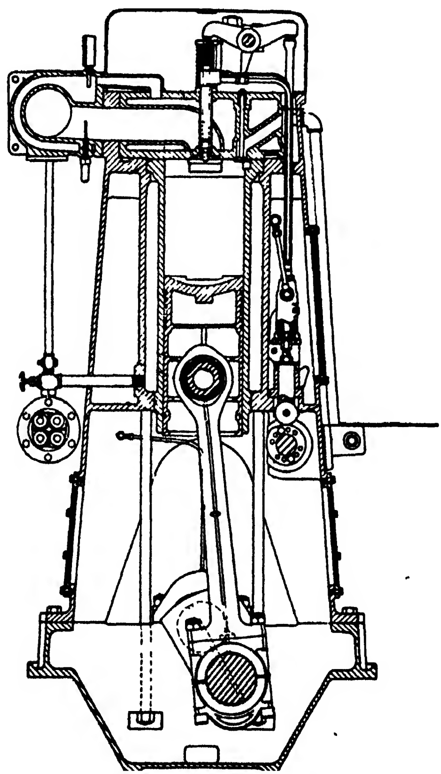


FIG. 105.—Frame of the Buckeye Diesel.

All the main parts of the engine, that is, base and center frame, cylinder block and cylinder heads, are held together by four long, alloy-steel through-bolts. These extend from the truss sections beneath each main bearing to the top of the cylinder heads.

De La Vergne Frame.—The horizontal frame design used for many years by the De La Vergne Machine Company was abandoned some years ago, and a vertical box frame (Fig. 107) adapted. The frame, or upper section of the crankcase, is of box type, heavily ribbed, and is bolted to the bedplate. Large doors give access to all the bearings and connecting rods.

The model VA and VB engines are provided with separate cylinder housings, bolted to the frame The model VG engine

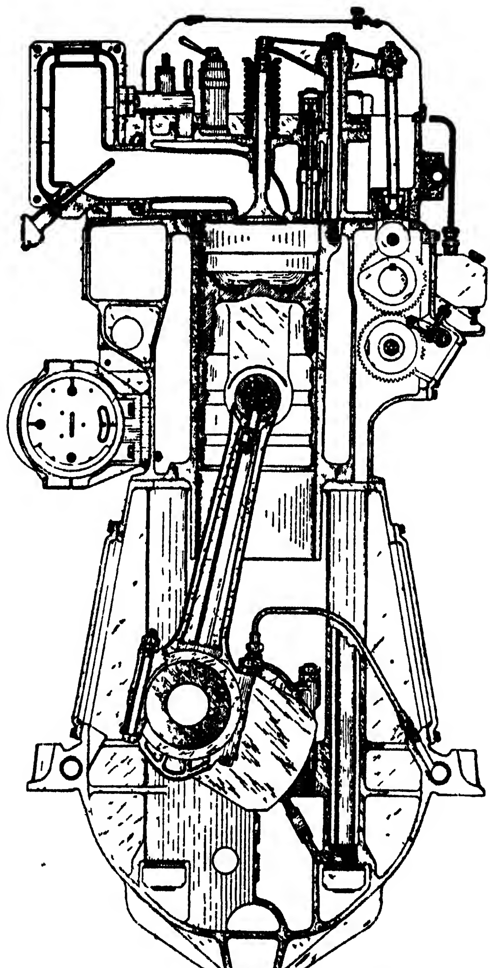


FIG. 106.—Section through a Cooper-Bessemer frame.

is of the en-bloc frame design, with the cylinder housings forming an integral part of the frame.

All engines are provided with removable cylinder liners. The liners are made of close-grain, heat-treated, nickel-alloy cast iron.

To obtain uniform wall thickness, the outside, or water side, of each liner is machined, whereas the interior circumference, after boring, is ground to close tolerance.

The upper end of the liner is provided with a flange which seats in a bored recess with a $\frac{1}{32}$-in. copper gasket to insure tightness. At the lower end, grooves are machined in the outer wall of the

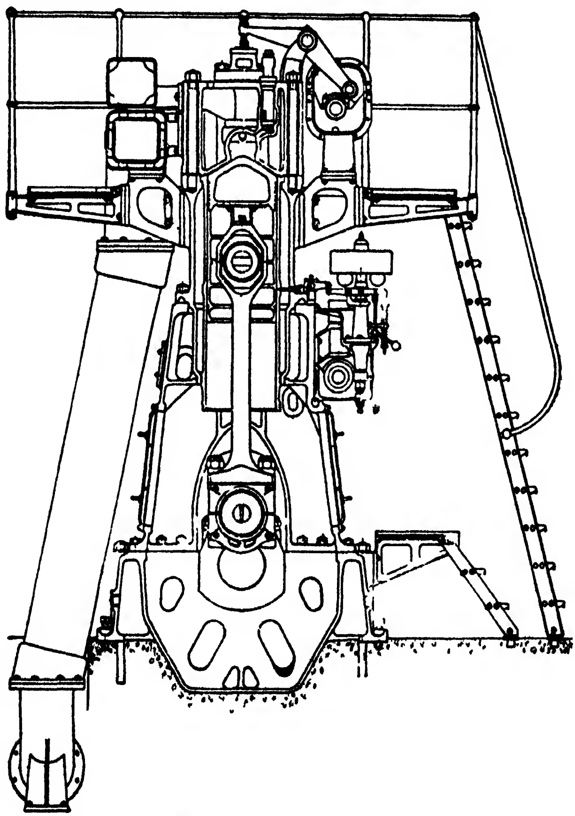


FIG. 107.—Cross section of De La Vergne vertical engine.

liner to accommodate two rubber rings. These rings seal the water-jacket joint.

Conclusion.—Obviously it is impossible to show the frames of all makes of Diesels. Those mentioned in this chapter have been chosen as being representative, and the various illustrations depict the general lines of all frames.

CHAPTER X

MAIN BEARINGS AND CRANKSHAFTS

General.—Aside from valve grinding and retiming of the fuel valve, adjustment of the main, or crankshaft, bearings is the maintenance work the Diesel operator performs most frequently. Still, the amount of this bearing wear is not serious so long as the lubrication is good. Diesel-engine bearings may be divided broadly into two classes, namely, adjustable and non-adjustable. The former, of course, embodies the addition of a wedge and, as applied to the vertical-type engine, is outlined in Figs. 10 and 66.

Adjustable Bearings for Vertical Engines.—This bearing design is no longer found on Diesels and followed gas-engine practice of 25 years ago. At that time the rapid wear of the main bearings seemed to demand some form of adjustment. It is now apparent that the trouble lay in excessive bearing pressures, non-homogeneous babbitt, and very flexible shafts. With more liberal bearing area this difficulty of rapid wear has largely disappeared.

If the engine is in the hands of an experienced and skilled operator, the adjustable bearing is not dangerous. On the other hand, a less intelligent mechanic will quite likely damage the engine when adjusting the bearing. It is well-nigh impossible to have three or four bearings with babbitt liners of uniform characteristics. The babbitt first run out of the ladle is of a density considerably higher than the bearing that is poured last. The natural result of this non-uniform density is the variation in the bearing wear. To the engineer who is versed in the finer adjustments of a Diesel engine the realignment of the bearings is not a difficult task.

Modern Bearings.—The majority of present-day Diesels have main bearings of two-piece construction, both identical in design. Each piece is made up of a steel backing, or shell, upon whose inner surface a lining of babbitt or other antifriction metal has been deposited. At present, one manufacturing method consists of upsetting a section of steel tubing until the approximate shell thickness is obtained. The shell is then placed in a machine, such as a special lathe, and while the shell is rotated, hot babbitt is allowed to flow through a tube upon the inner surface. The centrifugal force due to the rotation of this babbitt before it is cold causes the lining to adhere to the shell and to be uniform in thickness and density. To assist the babbitt to adhere to the shell, the latter may be provided with dovetail grooves.

On large Diesels the shell is usually of cast steel or cast bronze, as shown in Fig. 108.

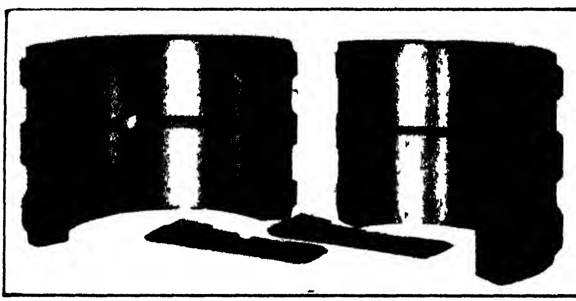


FIG. 108.—Main bearing of the Fairbanks, Morse Diesel.

Precision Bearings.—Up until a short time ago all Diesel engines were fitted with white-metal or babbitt-lined bearings, either spun into place or made in the form of removable shells. Higher rotative speeds and increased bearing loads have resulted in frequent bearing failures, usually starting with cracking of the babbitt surface and finally resulting in complete breakdown. Increased lubricating-oil flow, increased pressure of oil, improved filtering of oil are of no particular avail in reducing the possibilities of ultimate bearing failures. The fundamental fault lies in the reduced physical strength of babbitt, particularly at higher operating temperatures.

The solution of the problem has come through the development of the copper-lead bearing. The bearing material is an alloy made up of approximately 30 per cent lead and 70 per cent copper. The making of this mixture into a fine-grained homogeneous structure has been an outstanding development in the bearing art. The thickness of the shell is about 0.075 in. for a 3-in. shaft, and the bearing metal itself is approximately 0.020 in.

thick. These bearings are made with such precision that it is not necessary to bore or ream them after installation or even to scrape them to a fit. The running clearance for the copper-lead bearing is about 0.001 in. per 1 in. of shaft diameter.

It is necessary in applying this new bearing to employ crankshafts of greater hardness, for it has been found that when "running in" a new set of bearings there was danger of cutting the shaft. This has been entirely overcome through insuring ample and clean lubricating oil at all loaded points and with the use of a high-viscosity oil for running in. The increased oil-film thickness prevents intimate metal-to-metal contact and, over a period of running of a few hours, leaves the surface of the shaft in perfect condition.

Crankshaft Alignment.—A crankshaft for a medium- or slow-speed Diesel costs from $5 to $6 per rated horsepower of the engine. It follows, then, that crankshaft and main-bearing maintenance deserves the attention of operator, owner, and service agencies.

The difficulty confronting many operators is the lack of familiarity with practical methods of handling the bearings and shaft. This should not be, for it is quite easy for a man familiar with micrometers and other tools to check shaft alignment and bearing performance.

Crankshafts do get out of alignment on an occasional engine, and it is by no means impossible for misalignment to exist for months or even years, revealing itself ultimately by shaft fracture.

In the main, four factors may produce misalignment. These are: (1) engine subbase out of alignment by reason of improper installation or foundation settlement; (2) bearings bedded out of alignment; (3) wear or defect in bearing; (4) bent crankshaft.

The crankshaft-bearing housings in the cross-members of the subbase are machined by a horizontal boring bar. If the subbase is relatively short, and the shaft of reasonable diameter, the boring bar can be sufficiently sturdy to bore all the housing in alignment. When the subbase is long, and especially when it is in two pieces, bolted together, the boring bar may have so much deflection that the housings are bored slightly out of alignment.

It is necessary under the latter circumstances to scrape the housings to a mandrel. This mandrel is a piece of shaft of the same diameter as the bearing housings.

As shown in Fig. 109, the mandrel, long enough to cover three bearing housings, is coated with Prussian blue at the sections where it rests in the housings. Rotation of the mandrel causes the high places on a bearing to be coated with the blue; if an entire housing is "high," the condition will be revealed by a coating on the surface.

The initial step is to check roughly the alignment of the bearings, so that the "high" ones can be determined before any are scraped.

Fig. 109.—Fitting main bearing shells to seats in subbase.

After three housings have been checked, the mandrel is shifted so that it extends across another housing. This is checked and if necessary scraped. The process is continued until all bearings are in exact alignment.

Many builders scrape the outer surface of the bearing shells so that the entire surface firmly seats in the housing.

If the total thickness of each bearing is made identical, the shaft must be in alignment when it is bedded in the bearings. It is good practice, however, to check the alignment of the shaft when seated in the bearing shells. This may be done by employing a mandrel of the correct diameter. The better method, however, is the use of a gap gage or inside micrometer. The micrometer is somewhat difficult to handle if one is not expert; the gap gage, which is essentially a combination of a dial gage and extension rods, is easily read, and its measurement indication is not dependent upon the "feel" of the mechanic.

The details of how to use a gap gage have been discussed in the chapter on installation.

After the engine has been in service it is possible that the crankshaft may become bent or out of alignment. To check the shaft's straightness, when the bearing caps are all removed (and the flywheel off the shaft), place a surface gage on the machined surface of each bearing housing in turn (Fig. 110), and read the dial as the shaft is slowly rotated. If there is no variation in the reading, the shaft is straight. This proceeding checks shaft straightness but does not reveal the condition of alignment. The alignment should be rechecked periodically by the gap-gage method.

On engines with a relatively short subbase, misalignment of the bearings seldom exists, since all bearing housings are bored at

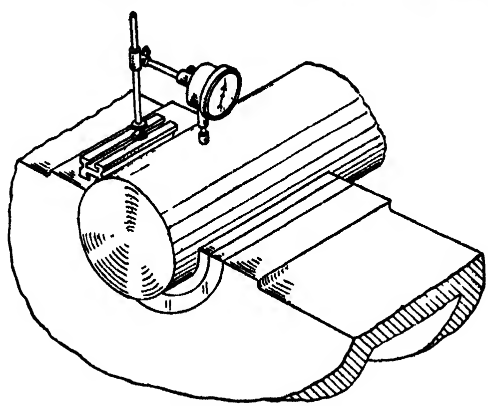


Fig. 110 —Checking shaft straightness.

one setting of the boring bar which is not long enough to deflect perceptibly. This permits the use of "precision-type" bearings. Such a bearing is made of a tubular-steel forging upon whose inner surface a thin lining of antifriction metal is centrifugally cast. The thickness of this lining is small, so that scraping cannot be followed. The housings must be bored true; and since the bearings are reamed to exact thickness, shaft alignment exists.

In case of bearing wear, the antifriction lining is so thin that serious misalignment will not occur unless the lining is entirely worn through.

On most engines having underslung shafts, the bearings are of the precision type, so misalignment seldom occurs. To check wear of the caps, which take all the wear, certain makes of engines are provided with a depth gage. After the lubricating-oil pipe is disconnected, the gage bayonet is inserted in the bearing opening, and the distance to the shaft is read on the gage; this should check with the original dimension, which is usually stamped somewhere on the bearing.

After an engine has been operating for some period of time, alignment of the shaft should be rechecked. This time period varies with the engine; certainly on all save high-speed Diesels 8,000 hr. should represent the minimum between such inspections. Frequently 16,000 hr. is not too long a time.

The first step in shaft inspection is to check the alignment, following the most suitable of the methods already outlined. If misalignment is revealed, the lower bearing halves must be removed, and the high ones scraped.

As with any instrument of the sort, the skillful use of the gap gage and the interpretation of its readings are matters of experience. Not only does an understanding of gage readings help to prevent accidents by locating hazardous conditions, but the useful life of the engine may actually be increased by eliminating a condition of shaft distortion before it has reduced the metal to a condition of fatigue with subsequent cracking.

The maximum bending of the crank webs as cantilever beams is then the greatest difference found between any two of the four readings taken. From this the fiber stress in the crank webs can be calculated by means of a simple formula, which is based on Hooke's law. This is as follows:

$$\text{Apparent fiber stress, pounds per square inch} = \frac{43{,}500 \times a \times t}{L^2}$$

in which

a = maximum difference between gage readings in two positions, mils (a mil is $\frac{1}{1000}$ in).

t = thickness of crank, web, inches.

L = length of crank, or half engine stroke, inches.

In the use of this formula it is assumed that the entire distortion takes place in one of the two crank webs, which is, of course, the worst condition that can exist. Actually the distortion of the crank as indicated by the gage may be divided between the two crank webs, so that the formula is intentionally on the safe side.

When the apparent fiber stress in the crank webs exceeds 20,000 lb. per square inch by this method of investigation, the shaft should be taken out of service immediately for expert inspection, for it is reasonable to expect that dangerous progressive cracks will have already developed in the crank webs or in the fillets of the crankpin or in the fillets of the journals.

Checking for Fractures.—Whenever the crankshaft is checked for alignment, it should also be inspected for incipient cracks. This occurs, if at all, in the crank throw, or web, close to the crankpin. The shaft should be thoroughly wiped clean of oil. The surface is then coated with a mixture of chalk and alcohol. The alcohol evaporates, leaving a thin coat of chalk. If a crack exists, the oil in it will mix with the chalk coating, showing up in a fine thread of dark color, which outlines the extent of the crack.

Inspection of Bearings.—Just as there should be a regular schedule in grinding valves and pulling pistons, the engineer should test the bearings at stated intervals. It is difficult to give a rule as to the length of time elapsing between inspections; the operator is not unduly active if he arranges to examine the bearings every 6 months. To remove the bottom half of the bearing of a small engine, it is necessary only to lift off the top half and turn the flywheel. The revolving shaft will draw the lower bearing from the housing.

On large engines the most common method is to insert a brass pin in the oil hole in the shaft and by rotating the shaft cause this pin to engage the edge of the lower bearing shell and pull the shell around, until it is free from the lower housing.

The thickness of the shell should be measured at both ends and at the center; several measurements should be taken at various points. If these measurements are the same, it is evident that the wear on the particular bearing has been uniform. If the measurements check with readings taken on the other bearings, the main bearings are in alignment. If one bearing shows that it is low, the engineer must decide his course of action. If the variation is less than 0.005 in., the best scheme is to scrape the other bearings, removing an amount of babbitt to compensate for the variation between the highest and lowest bearings. In cases where all the bearings vary, scraping is the only remedy.

That more attention should be given to bearings is evidenced by a crankshaft failure, photograph of which is here reproduced in Fig. 111. On examination the shaft showed marked crystallization. When the alignment of the bearings was checked, one was found to be $\frac{1}{32}$ in. lower than the others. The shaft had been unsupported between two of the cranks, and at each power stroke of these two cylinders it had bent. This bending, continuing for months at the rate of 350 times per minute, resulted in crystallization and final failure. In this breakage a loss of some $7,000, plus loss of operating time, is traceable to misalignment of the main bearings

FIG 111.—Welding of fractured crankshaft

Measuring Bearing Wear.—A ready means of measuring the amount the shaft is below its original position is available if the builder has supplied a bridge gage Some builders at the time the engine leaves the shop stamp the bridge-gage readings on the engine bed near each bearing and supply the engine buyer with the gage with which these measurements were made. This gage is similar to the one shown in Fig. 112, and the measurements recorded are the distances between the top of the shaft at each bearing and the point on the gage. Any subsequent measurements with the same gage, when compared with the original measurements, show how much the shaft is down at each bearing.

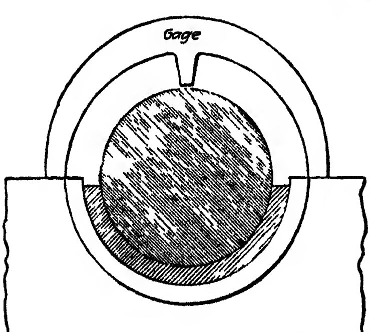


FIG. 112.—Bridge gage to check shaft clearance.

Variation in Bearing Wear.—The variation in the bearing levels should not exceed 0.001 in., for the housings are bored with a tolerance equal to or less than this amount. If more than this, the shaft will likely spring in operation. On engines with the flywheel mounted on an extension shaft, the wear is more rapid on the main bearing adjacent to the flywheel. The thoroughly versed erector always sets this bearing from 0.003 to 0.004 in higher than the other bearings. This allows a wear of 0.006 to 0.008 in. before the bearing becomes too low. Because of the initial misalignment this particular main bearing may run warmer than the other bearings. Many operators experience a great deal of worry over the temperature condition of this bearing when, in fact, the higher temperature actually indicates that the bearing is in proper shape. When the temperature becomes lower than with the other bearings, it can be taken as evidence that this bearing has worn low and must be raised.

Measuring Bearing Clearance.—For measuring the clearance in main bearings, where no gage is provided, the top cap of the bearing being worked on is lifted off with chain falls. The leads or soft fuse wires, say $\frac{1}{32}$ in. thick, are laid circumferentially around the top of the shaft, after which the cap is replaced, and the nuts sledged up to their former position. On engines with shafts 9 in. or over, three leads, one at each end and one in the center, should be used. The cap is removed a second time, and the thickness of the leads measured. The leads taken indicate the total amount of clearance between the shaft and the babbitt lining of the top half of the bearing, but they give no information as to how much of this clearance is due to wear of the bottom half and how much to wear of the top half. For this reason it is necessary that the lower bearings be measured for wear as indicated above. The clearance should be about 0.0007 in. per inch of shaft diameter.

Refitting a Bearing.—There are many instances where a bearing becomes hot, causing the babbitt to drag and cut; yet in the majority of cases it is not necessary to rebabbitt this bearing, for the actual reduction in bearing thickness may be less than 0.002 in. The proper procedure is to use scrapers made of old files along the lines of Fig. 113, although there are various scrapers on the market, which, if available, are probably handier. The babbitt should be smoothed down, and all lose and damaged metal cut out. This may cause part of the liner to be so low that it fails to contact with the shaft. This is not objectionable, since the ordinary shell has an excess of bearing surface. Where the damage is considerable, a new babbitt liner is the only solution.

With very large engines, short mandrels are usually supplied, one each of the same size as the main bearings, crankpins, and crosshead pins. When refitting a box that has been badly worn, overheated, or rebabbitted, the work can be done at any convenient place in the engine room, using the mandrel for spotting in. This makes it unnecessary to do the spotting on the bearing itself and does away with a lot of laborious handling of the box within the restricted space in the housing.

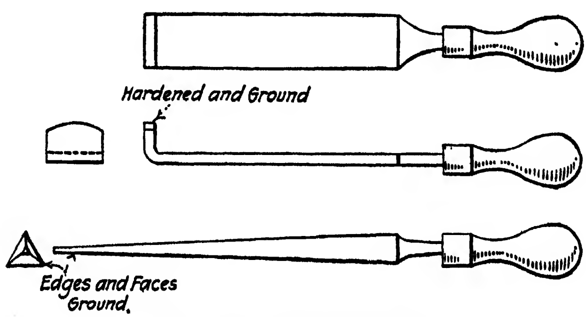


FIG. 113.—Bearing scrapers.

If the engine is fitted with precision bearings these cannot be scraped or rebabbitted; they must be replaced by new ones.

Rebabbitting a Bearing.—When the situation arises, the engineer must determine the best way of rebabbitting and getting the unit back into service. In too many cases the old babbitt is melted out, and a new lining run with a mandrel of the same diameter as the bearing. Often but not always the babbitt is scraped to a fair fit to the journal.

It has been generally found that, when handled in the manner outlined, the babbitt fails to maintain an anchor to the bearing shell; it has a decided tendency to loosen and finally crack when in service. Usually, the particular make of babbitt is blamed, when almost invariably the methods followed in the babbitting process are at fault.

In preparing to run a new babbitt liner, the old metal should be carefully removed. As a rule it is necessary to heat the shell in order to remove the babbitt from the dovetail grooves. After the metal is removed, the shell should be thoroughly cleansed of any oil deposits by washing first with gasoline and then with lye water and wiping dry.

Muriatic acid is next placed in a porcelain or pottery vessel, such as a small crock. Zinc scraps should be dropped into the acid until it is thoroughly "cut," or neutralized. The bearing shell is then brushed rather heavily with the cut acid. A blow-torch is played against the side of the shell while the engineer rubs a solder stick all along the inner surface of the shell, as shown in

FIG. 114.—Tinning bearing shell before babbitting.

Fig. 114. To insure a thorough tinning of the shell, sal ammoniac should be sprinkled over the surface as the coating of tin solder is rubbed into all the grooves. The tinning process insures a positive anchorage of the babbitt when run to the bearing shell. Without the preliminary tinning the babbitt, on cooling, appears to shrink away from the shell.

Before pouring the babbitt, it is necessary to procure a mandrel, if the engine builder has not supplied one. If a more improved device is not at hand, a quite serviceable one can be made of a piece of pipe and a companion flange, as outlined in Fig. 115 at *A* The pipe should be at least 1/8 in. smaller in diameter than the shaft journal. A piece of asbestos should be placed on the flange, and the two bearing shells clamped together with paper or brass shims between the halves and slipped over the mandrel. To insure against leakage, wet fire clay may be placed around the bottom edge of the shells.

Only high-grade babbitt metal should be used on engine bearings. In heating the babbitt in a ladle, two blowtorches impinging against the ladle are better than a coke or coal fire. The metal should not be hot enough to burn a pine splinter when the latter is thrust into the molten babbitt. If the stick chars but does not blaze up, the metal is about right for pouring.

After the bearing has been poured, it is never advisable to peen the surface of the babbitt. Although this practice is quite

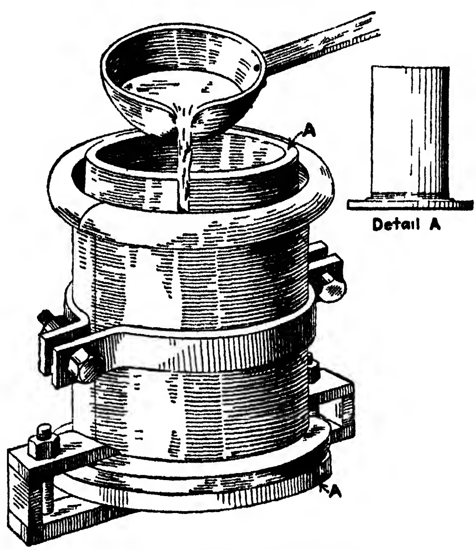


Fig. 115.—Babbitting bearing.

usual, the peening does not make the babbitt anchor any better, and often the hammering loosens the bond between the babbitt and shell.

Boring Bearing.—Any bearing that is under high pressures, such as occur with an engine bearing, should be bored to exact diameter. This is by no means difficult if a lathe or boring mill is at hand. Figure 116 illustrates one method of boring the bearing. Here the shells are chucked on the lathe, and the boring tool held in the tool post. The lathe speed must be low, or chattering will occur. Almost all bearings have a flat surface at the ends, and the bearing must be bored square with this edge. Care must be used in chucking the shells to get them in the correct position.

If the bearing is too heavy to be chucked, it can be clamped to the lathe carriage and bored by a bar made of a piece of steel shafting with a cutting tool passed through the shaft and held fast by a setscrew. Often a drill press is called into service. In this case the shells are clamped on the press table and bored by a tool having two cutting edges to prevent side slippage of the spindle.

After boring, the shells should be separated, and at the edges the babbitt should be cut away in order to provide a clearance which will enable the oil to enter between the bearing and shaft.

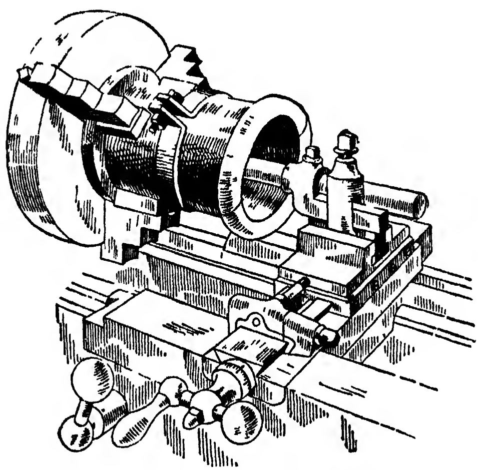


FIG. 116.—Boring rebabbitted bearing.

All bearings on engines should be scraped to an exact fit to the journal. This is best done by coating the journal surface with Prussian blue or lampblack. In a pinch, red lead thinned down with gasoline will do. If possible, a mandrel exactly the diameter of the shaft should be secured. The spotting of the bearing can be done with the mandrel, thus avoiding the difficult handling of the bearing in and out of the engine crankcase. A very light coating is applied, and the shell placed on the journal and lightly rotated. When removed, the shell will be coated with the lampblack at those points where the babbitt is high. Scraping after repeated trials will finally cause the babbitt to bear evenly at all points. In cutting oilways in the babbitt, the grooves should not extend to the ends of the shell or to the chamfer at the parting edges, this is to prevent a too rapid flow of oil out of the bearing. In fact, all that is needed are two short grooves intersecting at the middle of the bearing and about 4 in. in length.

With engines where the lower shell rests on a bored seat in the housing, the shell, including the babbitt, must be of the same thickness as the first shell when new. Consequently, in boring this must be measured and in scraping the babbitt must be cut away to give this exact value. If the procedure outlined is followed, a bearing equal to the original one will result. A small amount of care and patience in doing the work will bring large returns in long bearing service.

Hot Bearings.—A hot bearing is a trouble that every engineer some day will experience. Generally this occurs when the load is heavy and when a shutdown is an impossibility. The engineer must maintain his presence of mind even though the bearing smokes. It is an all too common practice for the operator excitedly to douche the bearing with a bucket of water. This is of no avail. Water is at best a poor lubricant, and it will merely make matters worse by washing off the little oil that does cling to the shaft. The water strikes the shell and the shaft; this results in the contraction of the shell, thereby loosening the babbitt. Another bad practice is the use of an air hose in a vain endeavor to cool the bearing. The only correct procedure is to run an oil pipe or hose to the bearing and feed a heavy stream of cool oil through the inspection hole in the cap directly on to the shaft. A hot bearing is generally traceable to a failure of the lubricating-oil supply. It is best to run the engine light until the bearing cools off. If the engine is arranged to allow a cylinder to be cut out, the two cylinders adjacent to the hot bearing should be operated idle, with the exhaust valves blocked open. This relieves the damaged bearing of part of the pressure due to the cylinder explosion. After shutting down, the bearing liners should be examined, and any necessary repairs made.

Clearance between Bearing and Crank Cheek.—Many engines after being in service a few years develop considerable side play in the crankshaft. This is attributable to excessive clearance between the end of the main bearings and the crank cheeks or throws. Engines of different makes vary as to the allowable clearance, but 0.007 in. is a representative value. If the clearance is too great, the remedy is to tin the ends of the bearing and run a ridge of babbit around these ends. Using a file and scraper, all surplus metal can be removed, and this ring of babbitt reduced to the desired thickness. This will prevent the side play and should last for at least a year before requiring renewal.

In direct-connected engine-dynamo units much of the side play is due to the magnetic pull. If the field and armature are not centered, there is a tendency for the shaft to be pulled lengthwise until the two electrical elements are center. The thrust of the connecting rods then causes the shaft to be gradually worked back, whereupon the magnetic pull again starts the lengthwise motion.

A second cause is the existence of a low bearing which will set up a lengthwise travel of the shaft. A third cause lies in the existence of a crankpin box which is incorrectly fitted to the pin, throwing the plane of travel of the connecting rod out of square with the shaft. The rod thus exerts a lengthwise thrust upon the pin and shaft. The engineer should see if the cause can be eliminated before using the thrust-ring mentioned.

Bearing Pressure.—The dimension of the main or shaft bearings is almost entirely dictated by the shaft size, which in turn depends upon the cylinder dimension, number of cylinders, etc. As a general statement, it may be said that the bearing pressures should not exceed 1,000 lb. per square inch of projected area.

Comparison of Two-cycle and Four-cycle Bearing Wear.—In a four-cycle engine the downward pressure upon the lower bearing shell is exerted during only part of the cycle. Upon the suction stroke and for the later part of the exhaust stroke the unopposed inertia of the piston is sufficient to eliminate the downward thrust on the bearing.

The film of lubrication has, then, on the four-cycle engine a chance to reform, and, consequently, bearing wear due to poor lubrication is not prevalent.

In the two-cycle, single-acting engine usually the downward thrust exists during the entire cycle. If the lubrication is not positive, there is danger of the film of oil breaking. In this event metal-to-metal contact would occur, resulting in rapid bearing wear. Where pressure or stream lubrication is carried out, no more difficulties are experienced on the two-cycle than on the four-cycle engine. The problem of bearing wear in the two-cycle engine was solved when a good system of lubrication was adopted.

Crankshafts.—The crankshaft is rightly one of the most important parts of the engine. The life of the Diesel depends in a great measure upon the proper design and maintenance of the shaft.

In the small- and medium-powered engine, say, up to 22 in. diameter, the shaft is a one-piece forging of open-hearth steel

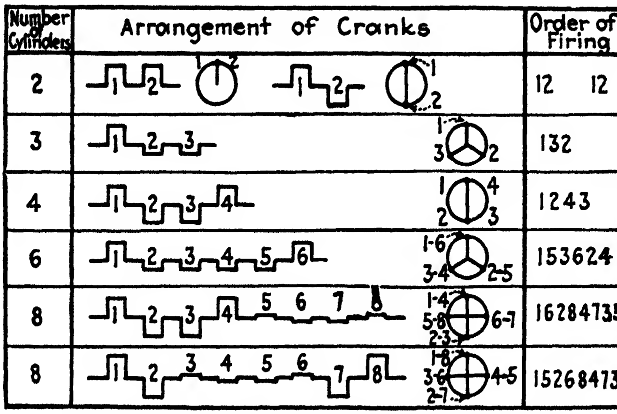


FIG. 117.—Order of firing, four-stroke-cycle engines.

of around 64,000 lb. tensile strength. In larger cylinder units the shaft is necessarily built up, since it is well-nigh impossible to secure such large forgings.

The arrangement of the cranks and order of firing of the cylinders are shown in Fig. 117 for four-stroke-cycle engines and in Fig. 118 for two-stroke-cycle units.

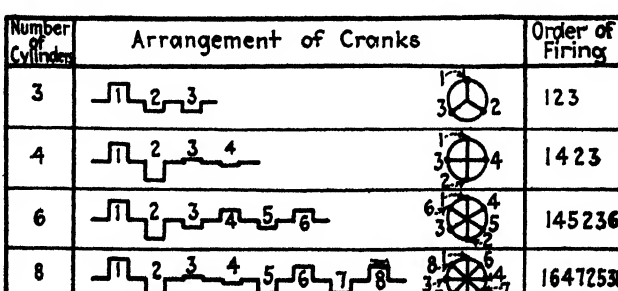


FIG. 118.—Order of firing, two-stroke-cycle engines.

The theory entering into the design of a Diesel crankshaft is no different from that of steam engine, gas engine, or air compressor shafts and is given in all treatises on pure design. Those who are interested are referred to such volumes, for the complete treatment is beyond the scope of this work and is a matter of design rather than of purchase and operation.

Lloyd's Rule for Crankshafts.—Lloyd's survey demands that the crankshaft diameter be as determined by the formula

$$D = \sqrt[3]{D_c^2(AS + BL)}$$

when

D = shaft diameter, inches.
D_c = cylinder diameter, inches.
L = distance below bearing centers, inches.
S = piston stroke, inches.
A = constant } as per table below
B = constant }

Number, cylinders	Compressor driven by overhung crank		Without compressor	
	A	*B*	*A*	*B*
1, 2, 3	0 086	0 038	0 089	0 037
4	0 093	0 037	0.099	0 036
5, 6	0 103	0 035	0.111	0 035
8	0 120	0.034	0.131	0 033

The table is based on a four-stroke-cycle, single-acting engine. If the engine is two-cycle, the constants for a given number of cylinders are those shown in the table for twice the given number of cylinders.

Crankshaft Proportions.—Some designers proportion crank webs by the formula

$$a = \sqrt{\frac{3LD}{4bF}}$$

where

a = thickness of web in direction of pin and shaft length.
L = load on crankpin.
D = half distance between web centers.
b = web breadth.
F = allowable fiber stress = 7,500 lb. per square inch.

In a discussion before the English Diesel Engine Users Association it was recommended that, using the cylinder bore as unity, the crank dimensions should be approximately:

Diameter of pin and shaft	0.525 to 0.54
Length of bearings	0.75 to 0.80
Length of crankpin	0.525 to 0.54
Thickness of web	0.32
Width of web	0.8

Fractured Crankshafts.—The bogie of the Diesels when first introduced in this country was broken crankshafts. Undoubtedly the one thing that proved an obstacle to the introduction of the oil engine was this question. The early Diesel operator knew but little about the engine and believed in letting well enough alone. No attempt was made toward the adjusting of the various parts. The fuel valve frequently got out of order and opened too early, causing pre-ignition. These excessive pressures had to be relieved by some means. Often the head gave way, but at times the head proved stronger than the shaft, and so a fractured shaft was the result. The second cause was the failure to take up the wear in the main bearings. Frequently the inside bearings became worn, allowing the shaft to be supported by the two outside bearings only. This produced a deflection in the shaft which was repeated and reversed each revolution. Ultimately the shaft gave way. As a rule, the break occurs between the pin and web (or throw). More liberal fillets at this point along with more knowledge acquired by the engineer will eliminate this danger. The third cause was the fact that torsional vibration set up owing to lack of stiffness in the shaft.

Welding of Crankshafts.—It has been quite customary in the past to discard a shaft upon fracturing. When it is considered that a 9-in. shaft for a four-cylinder 500-hp. engine costs around $7,000, it is apparent that it should not be junked if it can be saved.

Of late years electric, acetylene, and thermit welding have been used. Of the three, apparently electric and thermit are uniformly successful. In Fig. 111 is shown a 500-hp. engine crankshaft which broke because of a low bearing. Thermit welding was resorted to with success. Such work should be left to experts and if done right is economical. In the case in question the cost was $1,000, a saving of $6,000 over a new shaft.

Scored Shafts.—There are occasions when an engine crankshaft becomes scored to a serious extent or worn unevenly at the journals. The average engineer is prone to think that a shaft in either of these conditions is worthless and should be removed at once. When the damage is severe, of course, this is necessary. More often a little work will allow the engine still to pull its load. In cases of scored shafts an emery stone will serve to smooth up the shafts, finishing with a scraper and a final polish by lapping. Figure 119 outlines a lapping sleeve used by one large Diesel user. The scores do not cause trouble provided the edges of the cuts are smoothed off, preventing the babbitt from being picked up. When the wear is uneven, the bearing

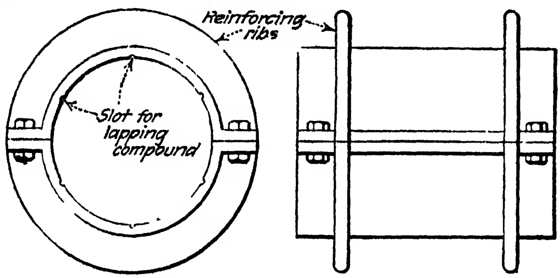


FIG. 119.—Lapping sleeve.

shell should be scraped to a fit, even though the diameters at the two ends vary to a marked degree.

Bearing Wear.—The flexure of the shaft will gage the amount of bearing wear or rather the amount of inequality in the bearing wear. In case of a four-cylinder, four-cycle engine reported to the Diesel Engine Users Association, the following measurements were recorded after 18,000 hr. of operation:

Cranks	1 Inches	2 Inches	3 Inches	4 Inches
Top center	7.118	7.099	7.1050	7.114
Bottom center	7.116	7.0985	7.1045	7.117
Differences	0.002	0.0005	0.0005	0.003

One point, however, should be remembered: The presence of shaft flexure, as indicated by a difference in the top and bottom measurements of a crankpin, does not locate the low bearing. This must be found by measuring the bearing thickness.

Busch-Sulzer Brothers found that with 100 engines reported the bearing wear averaged 0.0015 in. per year.

Crankshaft Wear.—The amount of wear on pins and journals depends upon the character of the steel, the type of bearing, method of lubrication, and general care. All cranks and journals tend to wear oval, since the high pressure on the piston occurs during a small portion of the crank circle. One 6½-in. shaft after 26,000 hr. service showed the following amount of wear, using the journals as numbered in Fig. 117:

Journal No.	Oval, In.
2	0.0015
3	0.0033
4	0.0011
5	0.0007
6	0.0018

Crankshaft Torsional Vibration.—Those who desire to delve deeply into crankshaft torsional vibration will find in various technical papers presented before the several engineering societies very complete studies and calculations. No attempt will be made here to go so deeply into the subject, but instead a simple outline of what torsional vibration is, how it is set up, and how to avoid it will be given. It is intended to meet the needs of the operating engineer.

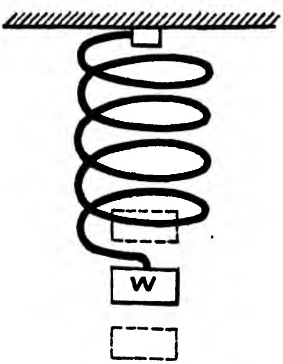


Fig. 120.—Action of spring when weight is forced downward.

If a weight is suspended by a spring, as shown in Fig. 120, it will be found that when the weight is given a downward pull the weight and spring will start to bob up and down with a decreasing displacement, or movement, until the weight finally comes to rest. The observer will notice that even though the movement, or downward displacement, gradually diminishes, the time interval occurring while the weight completes a cycle of downward and upward motion remains constant. If the weight is small compared to the mass of the spring, so that its inertia has little influence, this cycle interval, or period, depends upon the length and coil diameter of the spring as well as upon the thickness of the spring wire and its structure. If the spring is weak and long, the period will be long; if the spring is short and thick, its period is short, and the cycle occurs many times more often per minute.

It will be found that if the weight is given an angular displacement, or twist, it will not only return to its original position but also go beyond it a distance almost equal to the original twist. A reversal of angular travel then occurs, and the alternate winding and unwinding of the spring will continue until it finally comes to rest. This is essentially a *torsional vibration* of the spring, and the time taken for the cycle of twist and untwist depends upon the spring form and character.

Torsional Vibration of a Fixed Shaft.—If a straight piece of shafting is bolted to a wall, as in Fig. 121, a force exerted on a lever bolted at the outer, or free, end will cause the shaft to twist and untwist or otherwise to experience torsional vibration. The amount of twist the shaft experiences will depend upon the force applied.

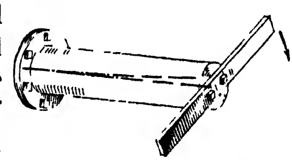


FIG. 121.—Twist of fixed shaft.

If now, after applying the force once to set up a vibration, the force is applied again at the proper time, the twist will be greater than before; and if a series of impulses is given to the lever, the amount of twist or angular displacement of the shaft will be increased to a marked extent. In fact if the force is great enough and applied enough times, the shaft may be twisted off.

Vibration of a Free Shaft.—Ordinarily, a shaft is not fixed as discussed heretofore but revolves in bearings and carries a flywheel. The action, however, is the same as with a fixed end. Suppose the shaft is mounted on two bearings, at the points indicated by arrows, with a flywheel at one end and a lever at the other, as in Fig. 122.

The flywheel weighs so much in comparison with the shaft that an ordinary force applied to the lever will merely twist the shaft, as when one end is bolted to a wall. If a line is scratched along the shaft, one will discover that where the shaft is twisted the angular displacement of any point on the line decreases as the point approaches the flywheel and that at the flywheel no twist is seen. In other words, the shaft inside the flywheel hub is a "node." However, if the flywheel is light in weight, the node, which is always located at the center of gravity of the combined shaft and wheel, might be at some point between the flywheel and free end of the shaft.

If the flywheel is at the center of the shaft length, and if a force is applied to both ends of the shaft in opposite directions, the two halves of the shaft will be twisted, or displaced, in opposite directions, with the part inside the hub stationary; it is still a node.

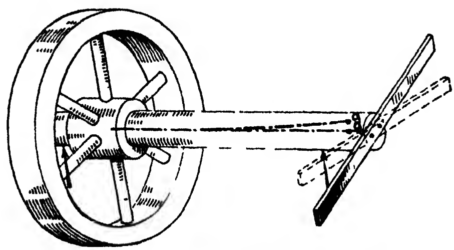


FIG. 122.—Effect of flywheel.

Vibration of a Rotating Shaft.—If the shaft and flywheel already discussed are rotated, and a force is applied to the lever, the twisting of the shaft will occur as in the case where the flywheel was at rest.

The longitudinal scratch placed on the shaft will, of course, be in rotation but will be a straight line as long as the force is not applied. A thrust on the lever will cause a shaft twist, evidenced by the scratch becoming a helix; but at the flywheel the twist becomes zero, and the shaft maintains its original position in respect to the flywheel; in other words the *node*, or no-twist point, still exists at the flywheel even though the shaft is turning. If a single impulse is given to the lever, the twist of the shaft is small; but if an impulse is given the lever each time the shaft untwists and travels to the other end of its angular displacement, the net displacement, or twist, will be greater. If the impulses are continued in synchronism with the periodic movement, the displacement may become very great, and under extreme conditions the shaft may be sheared or broken.

If the impulses are not in synchronism with the period of the shaft's twisting, they will have no effect on the shaft's twist, save when one of the impulses does occur at the proper time, but will tend to dampen the vibration, just as when one is in a swing, the latter can be brought to rest by a person on the ground giving a push at the wrong time.

Torsional Vibration of a Crankshaft.—To extend the torsion study to engine crankshafts, let it be assumed that the lever on the shaft end is replaced by a crank (Fig. 123) connected to a connecting rod and piston of an oil engine. Let it be assumed that the shaft and flywheel are rotating. At the first explosion, or firing, in the cylinder, the shaft twists; and if the firing impulses coincide with the natural vibration of the shaft set up by the original firing, the shaft will twist and untwist violently; that is, extreme torsional vibration will be set up, leading to ultimate fracture of the shaft.

For example, suppose that when given an initial twist the shaft vibrates at the rate of 60 times a second. If the cylinder turns

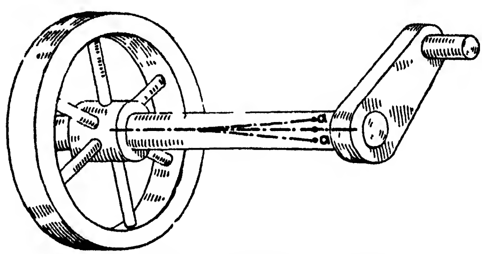


FIG. 123.—Result of addition of crank.

the shaft at 300 r.p.m., or 5 times a second, and fires at each revolution, only five impulses will be given per second, whereas the shaft naturally vibrates 60 times per second. The influence of the firing on the shaft's torsional vibration will not be of any moment, and the friction in the bearings and in the shaft fibers would probably destroy the effect. As the relation of the number of the firing impulses to the shaft's natural frequency of vibration is 12, we can say that the condition is a critical speed of the "twelfth order."

But if the engine had four cylinders firing at each revolution, we should have 20 impulses per second, or there would be a critical speed of the "third order." Some shaft vibration might be encountered, which would be dampened out if the speed were changed to, say, 260 r.p.m.

On the other hand, if the engine had six cylinders which fired on each revolution, and the engine ran at 300 r.p.m., or 5 per second, the firing impulses would then be 30 per second, or the shaft would receive a forced impulse at every other period, and the torsional vibration would be violent. Thus a six-cylinder, two-cycle engine at 300 r.p.m. would, if its shaft had the natural period assumed, have a critical speed of the second order.

If the engine had 12 cylinders, the firing impulses would equal the shaft's natural period, and the vibration would be so violent as to wreck the engine.

It will be noticed that with any given shaft the tendency for torsional vibration to occur increases with an increase in the number of cylinders. In addition, an increase in the number of cylinders requires lengthening of the shaft to accommodate the crank, which results in a more flexible shaft and a decrease in its natural period. This means that the engine's speed must be reduced unless the shaft's diameter is enlarged to make it stiffer so it will have a high vibration period.

For this reason crankshaft breakage due to torsional vibration seldom occurs on an engine having less than six cylinders. Vibration in engines of a lesser number of cylinders is generally due to unbalance of the reciprocating and rotating parts.

The subject of shaft vibration is extremely mathematical in approach, and the reader, if interested, is referred to several books on vibration.

CHAPTER XI

PISTON AND PISTON PINS

General.—Pistons and piston pins have given the engine designer and operator more anxious moments than all the other mechanical parts of the engine. The reason is that the piston crown must receive the brunt of the high pressures and high temperatures of the combustion gases. It must transmit the pressure force, through the connecting rod, to the crank and must transfer to the water-jacketed cylinder walls all the heat that the piston crown absorbs.

Since the piston reciprocates in the cylinder, there must be a coating of lubricant to reduce friction. This oil film is an insulator against heat transfer, consequently it interferes with the flow of heat from the piston to the cylinder wall.

Since there must be some clearance between piston and cylinder to prevent metal-to-metal adherence, this clearance further interferes with heat flow, with the result that about 80 per cent of the heat passing to the cylinder jacket flows through from four to six relatively narrow piston rings. It might be well to point out that the piston absorbs about 18 per cent of the heat of the burning gases.

The operator is there confronted with the problem of having enough side clearance to prevent the piston's sticking in the cylinder, a sufficient oil film to eliminate excessive friction, the proper kind of oil to avoid the deposit of gum and oil sludge, and rings that will prevent blow-by, with excessive ring wear.

Piston Types.—The pistons of all engines properly fall into two general classes, crosshead and trunk. The crosshead piston is usually shorter than the trunk piston and is provided with a crosshead which receives the side thrust due to the angularity of the connecting rod. Figure 124 illustrates this type; other examples may be found in a review of illustrations in previous chapters.

At the beginning of the stroke when the cylinder pressure is greatest, the angle made by the connecting rod is so small that side thrust is negligible. In consequence, cylinder or piston wear does not occur at the top of the stroke by reason of side thrust but is due to the gas pressure getting behind the rings.

With the trunk piston the upper end of the connecting rod is supported by the piston pin, which is fastened in the piston.

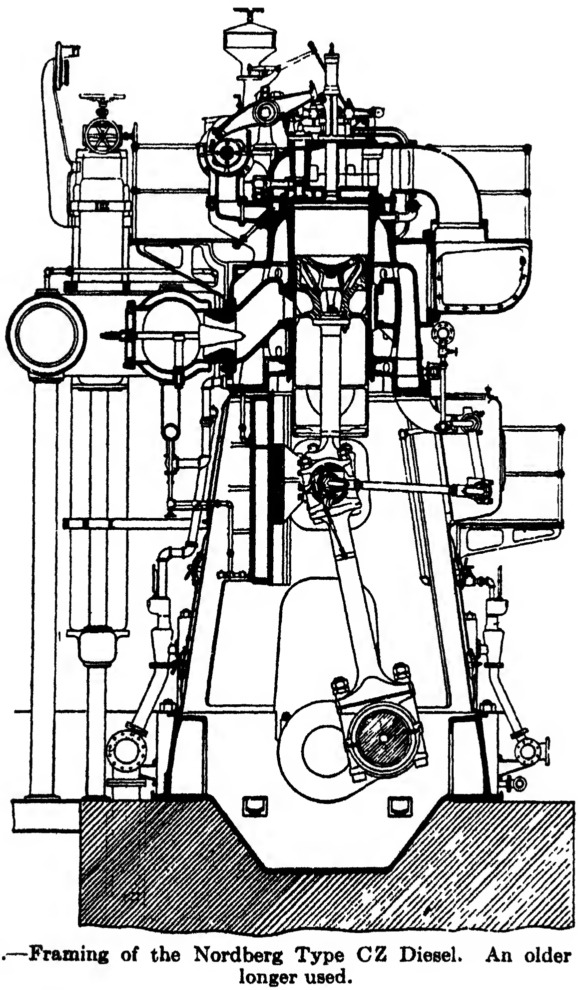


FIG. 124.—Framing of the Nordberg Type CZ Diesel. An older design no longer used.

Consequently, the piston receives the side thrust which is taken up by the crosshead in the former type. The engine shown in Fig. 125 employs the trunk design of piston, as do most of the engines discussed in this volume.

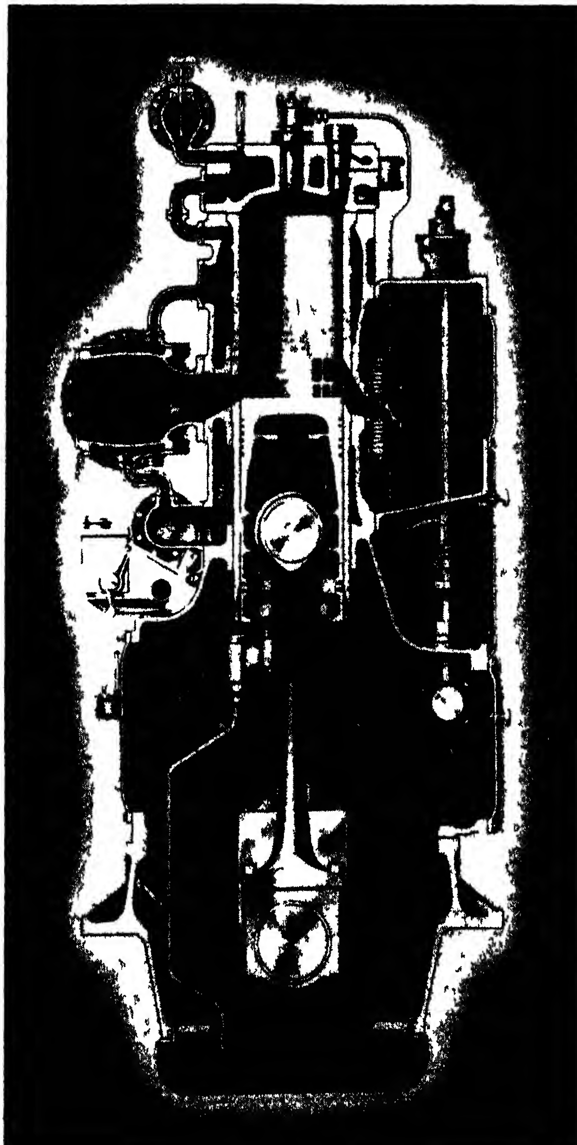


FIG 125 —The Alco-Sulzer two-cycle employs a trunk-type piston.

The American manufacturers of small- and medium-powered engines, up to 200 hp. per cylinder, have with few exceptions designed their engines with trunk pistons. Owing to the high cylinder pressures of the Diesel engine, the side thrust of the piston is of some consequence, although in a 200-hp. cylinder the piston is usually constructed of a length sufficient to bring the side pressure within reasonable limits.


FIG. 126.—A typical trunk piston for a high-speed Diesel.

On units with a rating beyond 200 hp. per cylinder, practically all builders use a crosshead design of piston; in fact, the first Diesel built, even though of 25 hp., employed a crosshead.

Trunk Pistons.—The trunk piston possesses certain features that make it attractive to the average operator. Since the side pressure is taken by the piston, there is no crosshead shoe to adjust. This adjustment on a Diesel must be made with a degree or knowledge possessed by none save experienced engineers. A guide clearance that would be quite satisfactory on a high-grade steam engine will prove entirely too liberal with the oil engine. The operator should understand that, beyond the pound that it occasions, a loose crosshead will allow the piston to bind in the cylinder, producing heavy scoring.

The trunk piston presents a problem in lubrication that does not exist with the crosshead type. The side thrust is borne by the trunk piston along its entire length, but this bearing surface extends over only a portion of the circumference. This rubbing area must be positively and copiously lubricated, and always the problem of lubricating a surface periodically exposed to hot cylinder gases is difficult. If the cylinder and piston are not oiled, either the piston or the cylinder liner will cut.

Since the transverse pressure throws the piston against the bearing side of the cylinder, the clearance between the piston and the cylinder must be less on the trunk than on the crosshead design. This is evident since, with the trunk piston, the entire clearance exists on the piston opposite the wearing side. Consequently, a clearance between piston and cylinder of 0.007 in. is

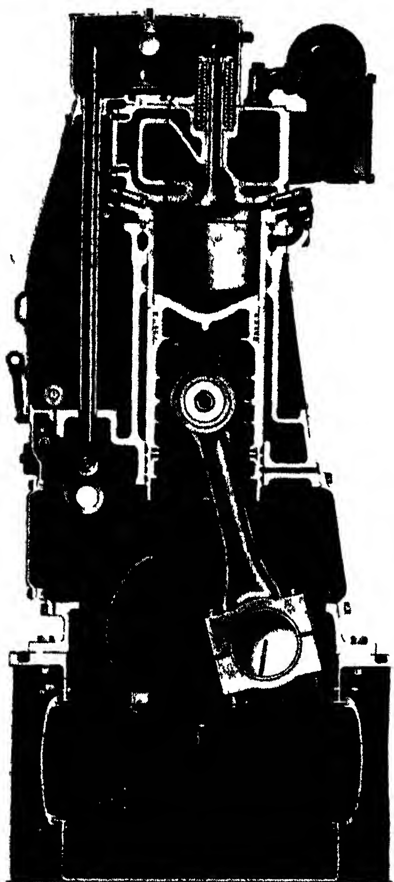
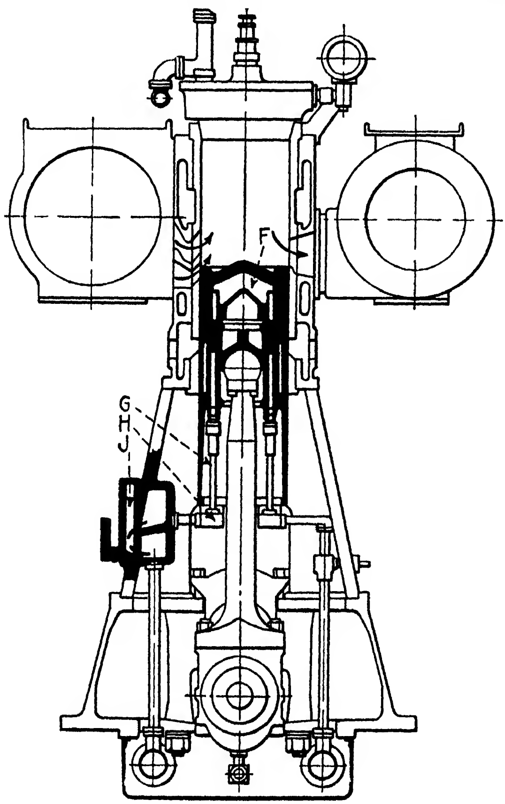


FIG. 127.—Piston of the Ingersoll-Rand Diesel.

actually a clearance of 0.015 when the engine is firing. On the crosshead type the clearance is fairly well distributed around the piston.

Waukesha Piston.—The piston of the Waukesha-Ricardo Diesel shown in Fig. 126 is of the trunk type. Three plain piston rings are used, along with an oil-control ring above the piston-pin bosses and one at the lower end of the skirt. The cup-shaped depression in the piston crown is to provide space for the burning gases entering the cylinder from the turbulence combustion chamber, thereby preventing overheating of the piston crown.



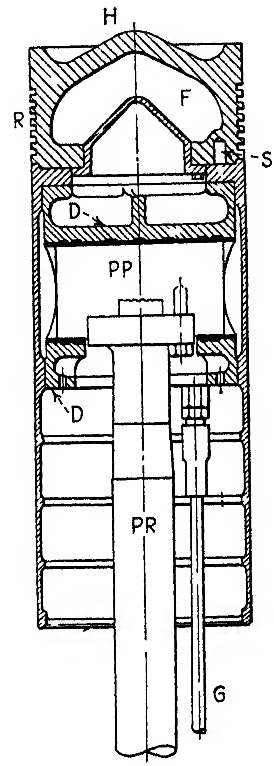


FIG. 128. FIG. 129.

FIG. 128.—Cooling arrangement of the Busch-Sulzer two-cycle Diesel.

FIG. 129.—Busch-Sulzer piston. *F* = oil-cooling compartment in piston head; *G* = oil tube to *F*; *H* = steel piston; *S* = cap screws, *D* = case supporting piston pin *PP*; *R* = ring grooves, *PR* = connecting rod

Ingersoll-Rand Piston.—In Fig. 127 is shown a cross section through an Ingersoll-Rand four-cycle Diesel. This engine employs two oil sprays which jet the fuel oil inward and downward into the space between the cylinder head and the concave piston crown. The piston crown, as with all engines, operates at a high temperature; to prevent lubricating oil thrown upward by the rotating crankpin from striking the inner surface of the piston crown, this is shut off by a light metal cover, shown in Fig. 127 just above the piston pin.

Busch-Sulzer Piston.—A novel piston design (Figs. 128 and 129) is found on the Busch-Sulzer Brothers Diesel. As will be seen from Fig. 129, the piston pin *PP* is bolted to the end of the connecting rod *PR*. The pin seats in the bronze-lined cylindrical cage *D*, which, in turn, is slipped into the piston. Since this engine is two-cycle, the pressure between bearing and piston pin is at the top of the pin; this design affords a long bearing surface and, consequently, a low unit pressure.

The piston skirt is not confined by the ends of the piston pin, so it does not distort. The piston crown, which is water-cooled on the larger engines, is a steel casting and carries the piston-ring grooves as shown.

On the Busch-Sulzer locomotive V-Diesel, the pistons are made of aluminum.

Crosshead Piston.—The crosshead design eliminates the heat difficulties of the piston-pin bearing which are so often present with the trunk piston. Opportunity is also afforded for a heavier reinforced piston head. There is also less likelihood of the piston's fracturing, since it is not confined at the pin bosses. The crosshead and rod design admit of an oil guard at the lower end of the cylinder, thereby preventing the throwing of lubricating oil into the cylinder with the consequent carbonization. The dripping of dirty cylinder oil or unconsumed fuel oil into the crankcase, where it renders unusable the bearing oil that is held there, is also eliminated by this design. To this can be ascribed the lower lubricating-oil consumption of the crosshead design of engine. These manifest advantages are, it is the feeling of the majority of engine builders, offset by its greater complication of parts and the greater necessity for intelligent adjustments. As a consequence, the small units are built with trunk pistons, going to the crosshead type when the cylinder sizes become so large that a trunk piston would require an exceedingly long cylinder. It is understood that the cylinder length depends upon the piston length which in turn depends upon the total cylinder pressure.

Nordberg Crosshead Piston.—Figure 130 *A* is a cross section of a piston for a Nordberg crosshead-type, two-cycle Diesel. The piston is in two parts, the crown *A*, which carries the piston rings, and the skirt *B*. The latter is bolted to the crosshead *C*.

The piston crown is oil cooled. Part of the lubricating oil going to the crosshead pin flows up through tubes to the piston crown and returns through another tube to the oil header. The piston crown is provided with a light metal core which forces the oil to flow along the inside of the crown.

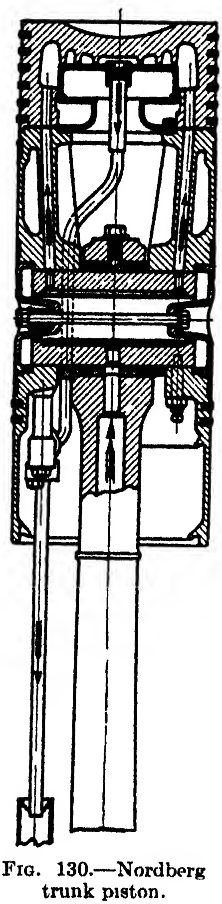


FIG. 130.—Nordberg trunk piston.

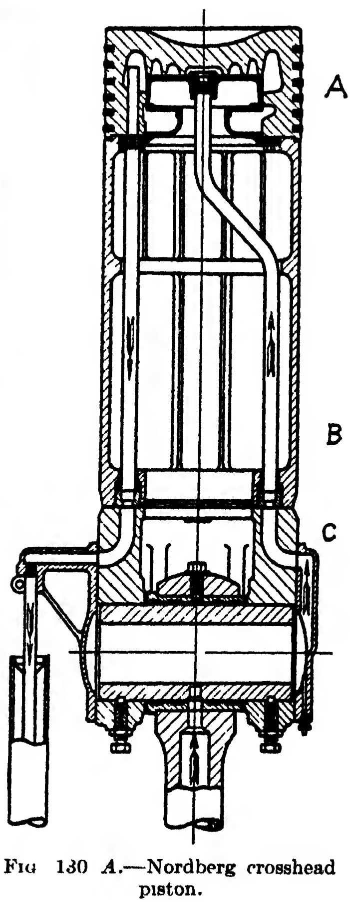


FIG. 130 A.—Nordberg crosshead piston.

The piston in Fig. 130 is a Nordberg trunk-piston design with provision for oil cooling.

Cooper-Bessemer Piston.—The trunk piston of the Cooper-Bessemer Type EN Diesel is of design somewhat like that of the Busch-Sulzer. As shown in Fig. 131, the piston pin *O* is bolted to the end of the rod *I*. The pin rests in the bearing bushing *D* which, in turn, is supported in part by the bushing *A*, in the piston bosses. Both aluminum and cast iron are used for the piston material; the choice depends upon engine speed.

Fairbanks, Morse Piston.—The piston on the 14 by 17-in., two-cycle Fairbanks, Morse Diesel has no unusual points.

The piston of the 16 by 20-in. engine is unusual, as will be seen from Figs. 132 and 89 *A*. To permit cooling of the piston

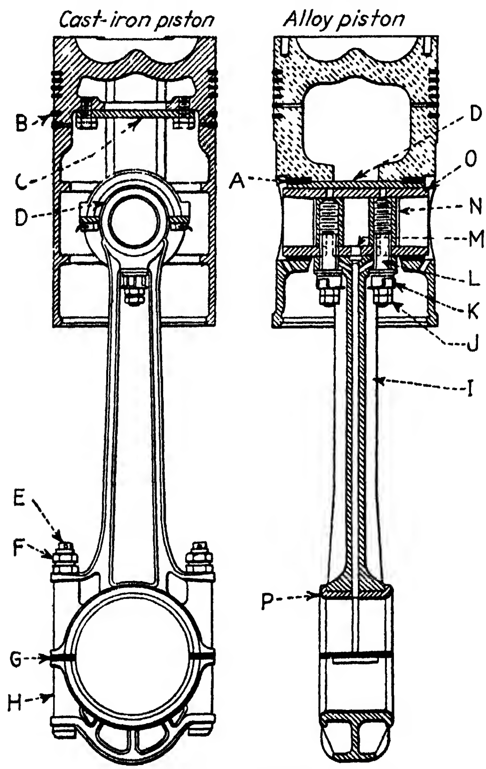


Fig. 131.—Cooper-Bessemer pistons.

head, which is carried out with the lubricating oil, the piston is made up of an inner core, which carries the piston-pin supports, and an outer casting, to form the head and skirt. One advantage of this design is that the manufacturing cost is not high, and, in addition, replacement, due to wear, involves only the outer casting.

The oil passes up through the hollow connecting rod, through the pin, and up a tube to the head compartment. The return is down a tube which passes through one of the piston-pin boss caps.

Other Pistons.—The pistons used on other Diesels are of general and well-known designs, as may be seen on inspection of various cross-sectional drawings shown in various chapters.

Stuck Pistons.—It is by no means uncommon to find that on a Diesel engine one or more pistons stick or "freeze" to the cylinder walls. (Stuck piston rings are not considered here.) .

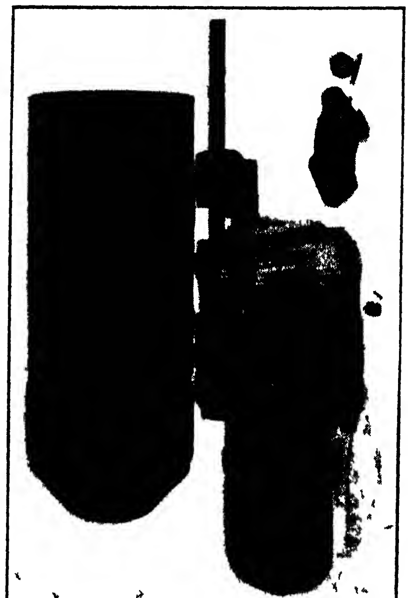


Fig. 132—Piston of Fairbanks, Morse 16 by 20-in. Diesel.

In the first place some cast irons have a marked tendency to "grow," or increase in diameter. As a result, the piston in time increases in diameter until close metallic contact is made with the cylinder liner. The engine loses power and finally stops. The freezing becomes so intense that the piston can be removed with only the greatest effort.

This growth apparently is due to the iron's having a low elastic limit under the high cylinder temperatures. The pressure then distorts the piston, which takes a permanent set, since it has been stressed beyond its elastic limit.

To correct this the piston should be tapered from the top, extending down to the first ring. The clearance at the edge of the top may well be as much as $\frac{1}{16}$ in. Some builders always taper the piston and have its straight portion extend only over the middle third of the piston length.

The second and more common cause of seizing, or freezing, of the piston is lack of cooling water. This may occur both while the engine is in operation and after it is shut down. The former is evidenced by the loss of power and decrease in the engine speed and usually happens on a change from light to heavy load. The average operator understands that on full load the quantity of cooling water required is greater than on the lower loads. As the load comes on, the usual practice is to increase the flow of the cooling water. This chills the cylinder liner and causes it to contract before the hot piston experiences any effect from this additional cooling medium. This contraction lessens the working clearance; the piston becomes hotter and ultimately grips the liner walls. The remedy is to keep the amount of cooling water uniform as the heavy load comes on; then, after the cylinder liner warms up, the flow of water can be gradually increased while the discharge temperature is kept fairly constant.

The best practice is to hold the cooling-water temperature uniform by thermostatic control.

There are also occasions when the seizing can be attributed to a hot piston pin, and ordinarily this is noticeable on starting the engine. After an engine has been in operation for some minutes, the heat absorbed by the piston is equaled by the heat given off by the piston to the cylinder walls, etc. The heat contained in the piston when the engine is shut down is large. If the flow of cooling water is discontinued at once, this heat is slowly radiated, and a great part is absorbed by the piston pin. The pin elongates as a result of this heat absorption. The end thrust of the pin produces a change in the shape of the piston walls; especially is this true when the walls are not strongly ribbed. The deformation of the piston occurs along the thinnest sections, which commonly are at the junction of the pin bosses with the piston walls. These ridges, or deformations, produce severe cutting of the cylinder walls, and many liners are ruined by this action. When this misfortune is experienced, the sole relief other than piston replacement is the filing of the high spots until the piston is again cylindrical. The surface can then be dressed by emery cloth. Filing will also smoothen the scored cylinder walls. A quick and efficacious repair can be accomplished by first dressing down the rough places with a fine emery wheel, say 80J grade, held in the hand, finishing with a file and emery cloth. To some this probably seems a radical treatment, but extensive experience on many scored pistons and cylinders tends to prove that this is an effective way to rectify the damage. If the engineer is careful in using the emery cloth, the cylinder and piston can be made as smooth as when new. Some engine builders flatten, or relieve, the piston skirt around the piston-pin bosses.

Piston seizing is at times in evidence when turning the engine over at the beginning of a run. If, at the close of the last run, the cooling water was shut off too early, the heat in the piston head may evaporate all the lubricant on the piston pin. As the dry bronze bearing absorbs the heat, it cannot expand outward because of the greater mass of the connecting-rod end. The bearing closes in on the pin, which is also expanding; this action results in a gripping of the pin, which is not completely loosened even after the parts have cooled. On starting, this wedged bearing restrains the motion of the connecting rod, and the engineer calls it a seized piston, though it actually is a case of a seized piston pin.

Still another cause of freezing is the carrying of too low a compression pressure. This, in turn, lowers the temperature at the end of the compression stroke. This temperature is not high enough to ignite the heavier part of the fuel. The heavy portion settles on the piston and cylinder walls. The rings gum, and eventually the piston sticks. The remedy is to raise the compression pressure. Ring sticking is considered later.

Cooling of Pistons.—One of the main problems of Diesel design is that of getting rid of the heat that is absorbed by those parts about the combustion chamber. The jacket cooling water as well as the head cooling water removes the heat absorbed by these parts. On the other hand, the fuel is injected downward toward the piston head, and the temperature at the head is much higher than at the cylinder wall or cylinder head. The removal of this heat is necessary if piston fracture is to be avoided.

The difficulty is to provide enough metal below the crown surface to enable the heat absorbed by the head to flow to the walls. If the heat flow is hindered, the heat, so to speak, accumulates in the crown, and the temperature approaches that of the cylinder gases. Fracture will result. For this reason aluminum is desirable, aside from the advantage of its lightness, for its thermal conductivity is four times that of cast iron, which means that on the basis of thermal conductivity the cast-iron piston must be four times as thick as the aluminum.

On small engines, with pistons of 18 in. and less, the natural air circulation in and out of the piston usually is all that is needed. On large cylinders some form of forced cooling is required. It was formerly the practice to use pure water only, even in the case of marine engines. This has been changed, and now sea water is used without damage.

In the case of engines in which sea water is used for cooling the pistons, a small header branches off from the main supply pipe, and it in turn has branches leading into each piston. The return water from each piston flows into a header, and this header discharges into the engine-room bilge. The reason this system discharges into the bilge is that the discharge from each piston is arranged to flow into a funnel on the discharge header so that the operator may observe the flow from each piston and test the temperature by feeling. Consequently, only atmospheric pressure can be carried on this line.

On marine engines whose pistons are cooled with fresh water or oil, the system is a closed one, and positive circulation is secured by means of a separate pump. A cooler is placed in the circuit so that the water or oil that is heated inside the pistons may be cooled defore it is again pumped through the system. A variation in the case of oil-cooled pistons is to discharge the hot oil from the pistons into the crankpits, where it mingles with the lubricating oil and passes into the sump tank.

Lubricating Oil for Piston Cooling.—Formerly, oil was the only cooling agent used in pistons, the reason for this being the difficulties experienced in preventing leakage of the cooling fluid into the crankpins and into the lubricating oil. Practically all lubricating oils available were compounded oils, and water leaking into the lubricating system formed an emulsion from which it was very difficult to separate the water. This emulsion quickly clogged the lubricating system. This led to the simple expedient of using a portion of the lubricating oil to cool the pistons. This method is in use on some trunk-piston engines using forced lubrication, a portion of the oil forced to the piston pin being led through a passage into the piston and, after passing through the piston, being discharged into the crankpit. The objections to this system are that the heat conductivity of the oil is rather poor, the heat of the piston sometimes causes the oil to carbonize, the carbon impairs the lubricating qualities of the oil and clogs the passages in the system, and the supply of oil to the pistons may become insufficient, owing to loss of oil through the ends of bearings that have too much clearance.

The operator of engines using this method of cooling should be constantly on guard against trouble from these last two conditions. The author has experienced persistent bearing trouble with a certain make of engine, and this trouble was finally traced to the clogging of oil passages by flakes of carbon formed on the inner side of the piston top and flushed into the lubricating system. In the case of a group of engines of another make, constant trouble was experienced from cracked piston heads. The pistons were dished on top, and the cracks invariably occurred in the center of this dished top, the metal cracking in much the same manner as muddy ground cracks when dried by the sum. Examination of the pistons revealed that the outlet pipes extended only a short distance into the piston so that the oil flowed out without filling the piston, allowing the portion of the piston head directly under the spray valve to become overheated. The outlet pipes in all the pistons were changed so that they extended into the pistons to a point close to the under side of the piston top. No oil could flow out until the piston was full and contact between the oil and piston top was assured. After this modification no more pistons cracked. In passing, it may be remarked that these cracked pistons were made serviceable by boring out the cracked center of each of the piston tops and closing the hole by screwing in a plug and welding it in place.

Cooling by Oil Spray.—On a few engines, notably the General Motors two-cycle truck Diesel, the piston crown is cooled by allowing part of the lubricating oil reaching the piston pin to jet out through a set of nozzles in the rod end and impinge on the inner surface of the piston crown. The oil then drips back down the piston and into the crankcase.

The only objection to this system is that the oil may overheat and deposit sludge in the lubrication system. The arrangement is shown in Fig. 133.

Water for Piston Cooling.—At one time water was used for the cooling of large land-engine pistons. Since the specific heat of water is twice that of oil, the amount to be pumped is one-half the equivalent amount of oil. If the operator keeps the water lines in shape, there should be no leaks into the crankcase.

Although water is used on marine Diesel pistons, even with a "straight" mineral-oil lubrication, difficulties are experienced if salt water is allowed to mix with the oil. The oil will form an emulsion with fresh water, but water can be easily settled out. For this reason some marine engines are equipped with fresh-water piston-cooling systems. Usually, one of the double-bottom tanks is used as a fresh-water reservoir; a special pump forces the water through the pistons; and, after leaving the pistons, it is passed through a cooler, being used again. This cooler is usually constructed on the surface-condenser principle, and salt sea water is circulated through it to cool the fresh water. For this reason the water in the piston-cooling reservoir should be tested periodically for chlorine. The presence of salt water, as indicated by an abnormal chlorine content, calls for an immediate examination and test of the piston-water cooler for leaks.

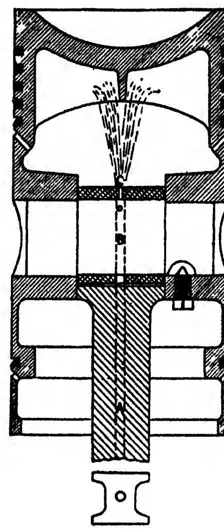


FIG. 133.—Cooling piston by oil spray.

Piston Rings.—Piston rings are, almost without exception, simple cast-iron snap rings. Special forms of eccentric-turned, spring-controlled, and double rings have been tried but usually have been abandoned in favor of the ordinary rings.

Where the service is extreme, the gap should be about as shown in Table IV, the reader must remember that individual cases differ.

Special Rings.—There are a score of special rings on the market, each of which is claimed to solve all ring problems. Some undoubtedly are better than the usual snap ring; especially does this apply to a two-piece ring made by the D. D. Cook Company, to one made by the American Hammered Piston Ring Company, and to a third ring having a long sealing tongue at the gap, made by Double Seal Ring Company.

TABLE IV.—RING GAPS IN THOUSANDTHS OF AN INCH

Diam., in.	No. rings from top				
	1	2	3	4	All others
Ends beveled 30 deg.					
12	25	19	16	12	6
17	34	25	21	17	8
21	42	31	26	21	10
Ends square					
12	51	38	32	25	12
17	68	50	42	34	17
21	84	63	53	42	21

Types of Rings.—In Fig. 134 are shown a number of the more common types of rings on the market. Rings 1 and 2 are plain

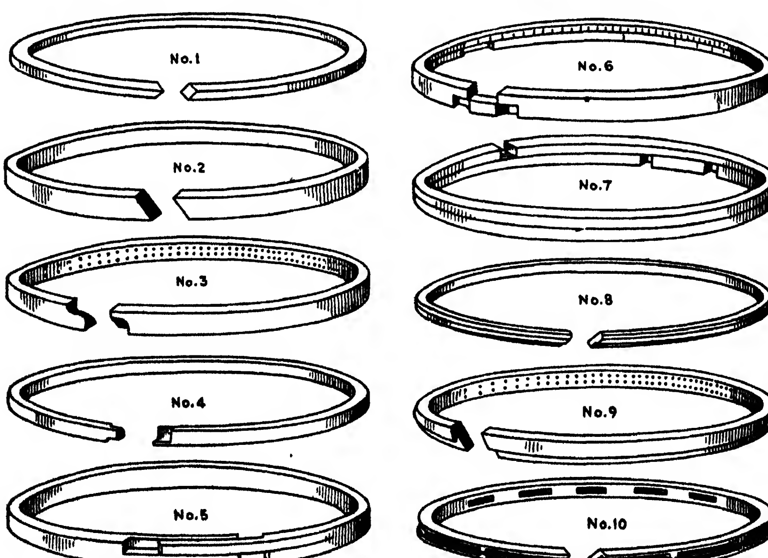


FIG. 134.—Types of piston rings.

"snap" rings. Numbers 3 and 4 are "seal" rings, on which the ring ends are so machined as to prevent gas leakage, to some extent. In No. 5 one sees a ring with a long finger slipping into a recess in the other end of the ring. This is termed a "double-seal" ring. In 6 and 7 the ring is made of two sections, to prevent gas leakage. Rings 8 and 9 are "oil-control" rings; each has a recess on the edge to scrape off excess oil from the cylinder wall. Ring 10 is a "ventilated" ring. Oil is scraped off the wall and flows through the slots to the back of the ring groove, whence it passes through drilled holes into the inside of the piston, finally dripping into the crankcase.

If two-piece rings are used, they should not be placed in the top groove, for the high gas pressures may jam the two parts of the ring and so prevent ring action.

Oil Control Rings.—On high-speed and medium-speed Diesels the cylinder walls are lubricated by the oil splashing out the ends of the crankpin bearings. There is no way to control the amount of this oil deposit; consequently, it is quite possible that a cylinder may receive an excessive oil supply. This oil is gradually worked up into the combustion space where it decomposes and settles on the cylinder wall and piston crown in the form of carbon. Part of this excess oil supply fills up the ring grooves and, under high temperature condition, will form gum which will stick the rings.

Oil-control rings, if properly applied, will scrape off the excess oil. In case the cylinder receives too little oil, enveiling an oil-control ring enables it to feed oil up to the dry part of the cylinder.

Stuck Rings.—Modern engine-operating temperatures cause lubricating oil to break down. Sludge formation occurs; to avoid this, many lubricating oils are given a "dope," or additive. This additive increases the ability of the oil to adhere to the metal surfaces and to improve lubrication, but in some cases the additive causes a lacquer to settle on the rings, sticking them. In most instances, however, ring sticking is a sign of an improper fuel oil. One should not ignore the fact that today's refining of fuel oil is such that gum may settle in the rings.

If an engine is of the slow-speed, heavy-duty type, gum may appear when the engine is underloaded. The reason is that the engine gets too cold for combustion to be good. The gum comes from the unburned fuel oil.

On the other hand, gum may appear in a high-speed Diesel when overloaded. It is here due to the breaking down of the lubricating oil.

One remedy for gum is the addition of Lubal to the fuel oil. This additive carries napthenic acid which acts as a gum-freeing agent.

Ring Grooves.—Ring grooves wear, as a result of the sliding action of the ring. In time, the wear is so much that gas enters behind the ring too easily. It is then necessary to re-machine the groove. This increases the width, and either a wider ring or two narrow ones must be used.

With aluminum pistons, the lands between the grooves may soften and break away because of high temperatures. An aluminum piston crown absorbs more heat than a cast-iron piston, and this heat tries to flow to the top rings. To avoid this condition, the aluminum crown is made thicker so that the heat flows to all the rings. The best solution is to use a bronze sleeve around which the aluminum is cast. This carries the ring grooves.

Life of Pistons and Rings.—The life of these parts depends upon the service as well as upon the design. Busch-Sulzer Brothers obtained reports from 100 engines, ranging in age from 3 to 10 years. The average replacement was one piston per engine per 8 years and five rings per engine per year. The average life of a cylinder liner was 6.88 years. Of the 100 engines 92 had never had a piston-pin bearing replaced.

Alloy Pistons.—Of late, much attention has been given to the subject of alloy pistons, which have proved successful in automobile engines of high speeds and powers. The Aluminum Company of America has done extensive research on the problems of proper dimensions and alloys. As pointed out by B. Isidin of that company, when designing large-size Diesel-engine pistons, the overcoming of the thermal difficulties becomes the most important consideration. In a cast-iron piston the head thickness reaches dimensions that become cumbersome and out of proportion, so that oil or water cooling is resorted to as a means for eliminating heat.

The amount of heat flowing through the piston can be determined by the formula

$$F = \text{horsepower} \times \text{pounds oil per horsepower-hour} \times \text{oil heat} \times \text{per cent jacket losses} \times \text{per cent heat to piston}$$

In this formula the jacket heat going through the head can be taken as 19 per cent.

The head thickness for an uncooled 23 by 26-in. cast-iron piston was determined as

$$t = 0000805FD^2 = 0.0000805 \times 207 \times 23^2 = 9 \text{ in.}$$

whereas for Lynite

$$t = 0.0000485 \times 208 \times 23^2 = 5.34 \text{ in.}$$

The metal thickness of the head determined from the strength viewpoint will be for cast iron 2.53 in. and for aluminum 2.75 in. This indicates that the thickness determined from the thermal-stress viewpoint would be emormous for a cast-iron piston 9 in. with stresses of 15,000 lb. per square inch, which is much more than that calculated to withstand explosion pressures, namely, 2.5 in.

These conditions make it necessary to resort to a means of cooling the piston by water or to use a metal with a better heat conductivity. Lynite as a material would permit an uncooled piston with 5⅜ in. thickness of head (a possible dimension taking into consideration the weight of Lynite in proportion to the size of the core) with thermal stresses of about 8,000 lb. per square inch, which are practical stresses for heat-treated No. 142 Lynite.

To calculate approximate maximum temperatures of the center of a piston, assumptions can be made such that the temperatures of the ring belt below the head will be the same for Lynite as for cast iron, which will be about 450°F. Considering 350° as a temperature gradient for a cast-iron piston, the temperature in the center of an uncooled cast-iron piston will be 800°F.; and if a temperature difference of 130°F. is assumed for Lynite, the maximum temperature in the center of a 23-in. uncooled Lynite piston will be approximately 580°F., which is a safe temperature for the material in question.

It is apparent that the limits which restrict the application of uncooled, cast-iron pistons may be raised considerably for aluminum. The cost of an uncooled aluminum piston to replace a cooled cast-iron piston will probably be below that of the cooled iron piston in view of the fact that all complicated and troublesome parts of the cooling mechanism are eliminated.

From the foregoing example the necessity of heavy proportions in large pistons can be inferred. Part of the heat absorbed by the head will be transmitted through the rings to the cylinder wall, and the sections behind the rings must necessarily be heavy to facilitate conveying that portion of the heat not transmitted through the rings due to their limited capacity. The heat not eliminated by the rings passes through the skirt and from here to the cooler part of the cylinder wall. Heavy sections bring the head temperatures to a lower value, which in its turn increases the absorbing capacity of the top of the piston.

Because of the difference in the coefficient of expansion of aluminum and cast iron, 0.0000123 as against 0.00000556, the clearances allowed on the top of an aluminum piston will be greater:

	Per In.
Top of piston	0.008
First land	0.007
All succeeding lands	0.006

The skirt is usually ground to a taper such that the top of it just below the last ring has a clearance of 0.0025 in. per inch of diameter, and the lower end has 0.0015 in.

Temperatures in a Diesel Cylinder.—Sulzer Brothers, some years ago, conducted a series of tests on the temperatures existing in a two-cycle Diesel piston and cylinder walls, in centigrade degrees. As shown by the data in Fig. 135, only on the walls surrounding the combustion chamber are the temperatures high enough to burn the film of lubricating oils.

Fractured Piston Heads.—The cracks that develop in the piston head can be placed in two classes, namely, those which take a circular form and appear around the base of the concave portion of the head, and those which are radial in direction. The former are caused by heat strains with a consequent breathing action at the base of the cone. These fractures seldom prove serious, and it is possible to continue to operate the engine without piston replacement. At times these fractures appear in conjunction with cracks on the interior side of the piston head, across the reinforcing ribs. In such cases the entire head may give way.

The really serious fractures are those that develop radially in the head. Often these cracks extend across the head some 6 to 10 in. (Fig. 136). The danger lies in the now non-rigid head allowing the piston to distort, scoring the cylinder walls. When the fracture is only a few inches in length it can be repaired by "sewing." A hole should be drilled at each end of the crack

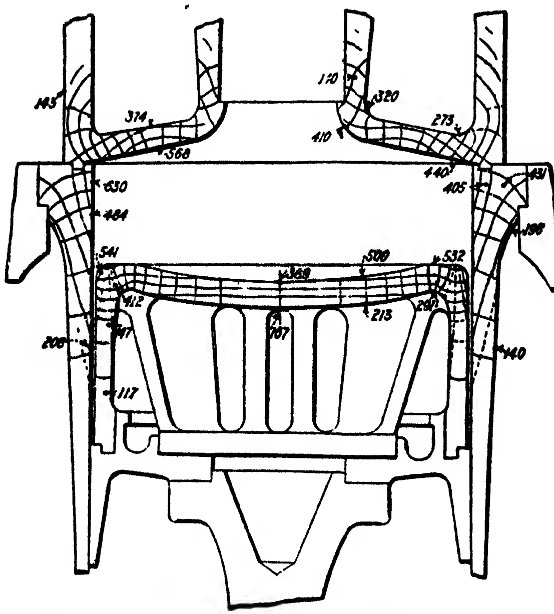


FIG. 135. Cylinder temperatures found in a Sulzer engine, degrees centigrade.

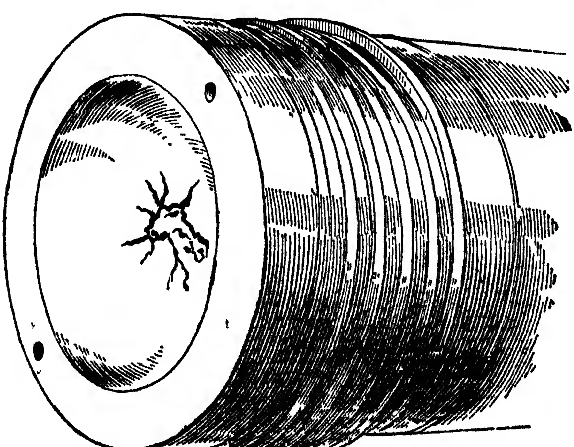


FIG. 136.—Fractured piston crown.

to prevent any further development. A series of $\frac{3}{8}$-in. holes is drilled and tapped along the line of the fracture. Into these

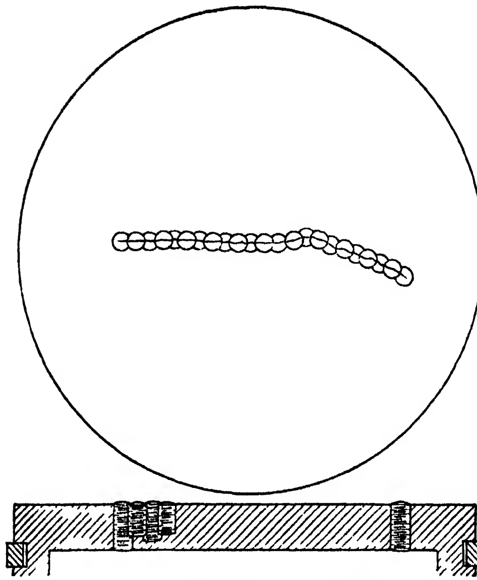


FIG 137.—Sewing fractured piston head

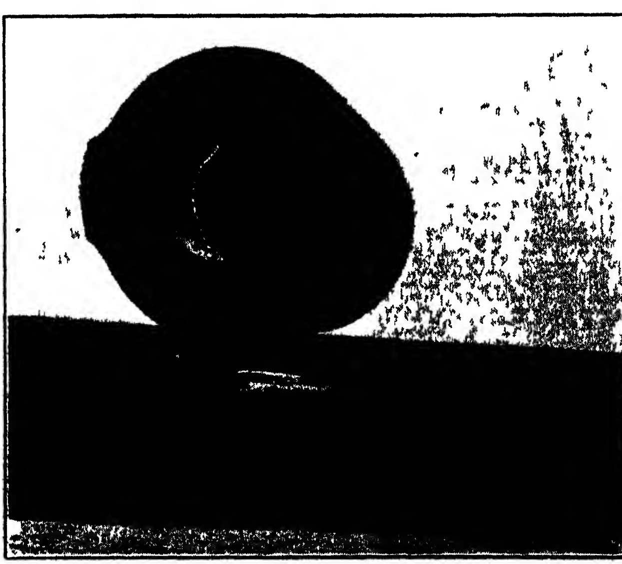


FIG. 138.—Sewed piston after 6 months' use.

holes threaded brass plugs should be inserted and cut smooth with the surface. Between these plugs a second row is inserted, lapping over the first plugs. This entire line of plugs is then hammered smooth (Fig. 137). This sewing has been practiced with success on pistons as large as 18 in. in diameter. Figure 138 shows a piston 6 months after sewing. Welding, if done correctly, is a success.

Piston Clearance.—No hard and fast rule can be given as to the allowable clearance between the piston and liner. It may be stated that generally this should be from 0.006 to 0.012 in. on pistons of 10 to 20 in. Much depends upon the design of piston and upon the cast iron of which it is made. Many show a tendency to grow at the crown, and for this reason the upper end above the top ring should be tapered, giving a side clearance at the top of about 0.02 in.

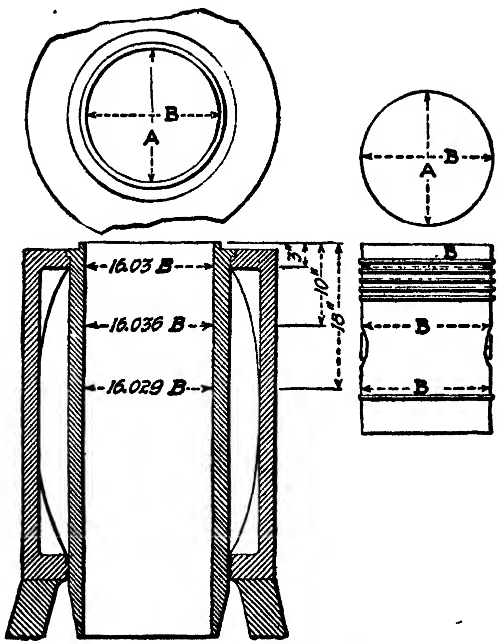
In Table V are given piston clearances of various engines. This clearance is measured at the largest diameter of the piston. Although the clearance on a new piston should be as stated, a piston can wear considerably before renewal and reboring or replacement of the liner become necessary. Engines have been run with clearances as great as ⅛ in. Under such conditions there is a good deal of piston slap, and the engine runs noisily. It may be taken as a general rule that for pistons up to 18 in. in diameter a clearance of 0.08 in., and for large pistons a clearance of 0.10 in., is about as much as should exist before renewal. Greater wear will increase the fuel consumption and lower the capacity of the engine.

Wear.—The rate of piston and linear wear varies; the figures below give the wear on one engine.

Length Service, Hr.	Rate of Wear per 1,000 Hr., Thousandths In.
4,000	2 8
16,000	2.3
21,000	2.8
25,000	3.5

The operator should check the wear at least every 6 months. If a drawing similar to Fig. 139 is kept on hand, the wear may be recorded at each inspection.

To maintain the proper compression pressure as the wear increases, the clearance between cylinder lead and piston may be decreased by shims under the rod's big end. It is likewise possible to secure correct combustion at low pressure by making the fuel admission earlier. There is danger in this, as excessive pressures may be developed, causing intense pounding.



CLEARANCE MEASUREMENTS

Distance from Top. Inches	Cylinder Measurements		Piston Measurements	
	A	B	A	B
3	16.025	16.020	15.975	15.980
10	16.030	16.029	15.980	15.982
18	16.029	16.019	15.975	15.982

FIG. 139.—Log of piston and liner wear.

In a discussion before the English Diesel Engine Users Association it was stated that piston and liner wear depends upon the following factors:

Ash.—Tar, creosote oils, and petroleum fuels, oils having an asphaltic basis, form ash, some of which may be, and generally is, of a highly abrasive character. The permissible ash content should not exceed 0.08 per cent, of which proportion not more than one-half should be of gritty nature.

TABLE V.—PISTON SIDE CLEARANCE PER INCH OF PISTON DIAMETER

Engine	Cycle	Bore, in.	Piston material	R.p.m.	Clearance per in.
Atlas-Imperial	4	4.75	Alloy / Cast iron	1,500	0.001
Buda	4	3.62	Aluminum alloy		0.002
Buckeye	4	9	Cast iron		0.0007
Busch-Sulzer	2	19.57	Cast iron	240	0.0014
Caterpillar	4	$5\frac{1}{8}$	Aluminum alloy	800	0.001
Chicago Pneumatic Tool	4	10.75	Cast iron	360	0.001
Cooper-Bessemer	4	11.5	Cast iron	400	0.001
De La Vergne	4	22	Cast iron	200	0.001
	4	9	Aluminum alloy	850	0.0014
Electric Boat	4	15	Cast iron	400	0.001
Fairbanks, Morse	4 / 2	$4\frac{1}{2}$ / 14	Cast iron	1,200	0.001
Hercules	4	5	Aluminum alloy	1,600	0.0014
I. H. C.	4	$4\frac{3}{4}$	Aluminum alloy	1,000	0.0014
Ingersoll-Rand	4	$17\frac{3}{4}$	Cast iron	300	0.001
Ingersoll-Rand (H)	4	$12\frac{3}{4}$	Cast iron	250	0.001
Ingersoll-Rand	4	10	Cast iron	600	0.001
McIntosh & Seymour	4	20	Cast iron	300	0.001
	4	$9\frac{1}{2}$	Cast iron	720	0.0009
	4	$12\frac{1}{2}$	Aluminum alloy	720	0.0014
Nordberg	2*a*	17	Cast iron	200	0.001
	4	16	Cast iron	240	0.001
	2	17	Cast iron	240	0.0036
	2	17	Cast iron	240	0.0012
Superior	4	$4\frac{1}{2}$	Cast iron	1,600	0.00075
Reid	2*h*	$13\frac{1}{2}$	Nickel, c.i.	325	0.001
Stover	4	5	Cast iron	1,000	0.001
Union	4	10	Cast iron	400	0.001
Waukesha:					
Comet	4	$6\frac{1}{2}$	Aluminum alloy	1,600	0.0014
Hesselman	4	6.5	Cast iron	1,000	0.00117
Winton	4	10	Aluminum alloy	600	0.0014

a = air injection; otherwise engines are solid injection. *h* = horizontal frame; all others vertical.

Side Thrust Due to the Connecting Rod.—Side thrust is influenced by the length of the connecting rod and the ratio between the length of the rod and that of the crank. The wear from this cause should be in direct proportion to the bearing area (projected) of the piston in the liner. Table VI gives particulars, including side thrust, of several engines. The thrust values from indicator diagrams taken from the several engines have been calculated. These values are not strictly comparable, as the engines were not all loaded to the same fraction of their normal rated outputs when the diagrams were taken. The crank angles, at which the various maximum side thrusts occur, merit a little consideration. For maximum side thrust with a constant-pressure expansion line, the connecting rod and crank should be at right angles to each other; that is, the crank angle should be about

TABLE VI.—SIDE THRUST

Engine	Cylinder diameter, in.	Ratio stroke bore	Ratio rod crank	Piston length, in.	Mean effective pressure per sq. in.	Fraction of rated maximum load, per cent	Piston speed (ft. per min.)	Side thrust	
								Lb. per sq. in.	Crank angle, deg.
g. 1	12	1.52 :1	4:1	27¼	93.0	100.0	760	37	35*
g. 2	12	1.52 :1	4:1	27¼	96.8	90.0	760	34	37
g. 3	20	1.2 :1	4.5 :1	....	87.0	100.0	800	33	25
4	13	1.52 :1	3.9 :1	30	80.4	82.0	800	31	43
h. 5	12	1.507:1	5.31:1	26⅛	....	83.6	755	31	39
6	17¾	1.47 :1	5.27:1	37½	00.3	78.3	758	20	26

78 deg. Owing to the rapid drop of pressure in a Diesel engine during the expansion stroke, the maximum effect of the connecting rod occurs a good deal earlier in the stroke. The table shows calculated side pressures from 30 to 40 lb. per square inch. The diagrams from which the particulars were obtained were taken under ordinary workaday conditions and are not put forward as examples of the best practice under test conditions. It is desirable to ascertain if possible to what extent liner wear and side thrust are interconnected, but, as the figures of wear are conflicting, no definite decision has been made. The values of side thrust being all very similar, the point is not of much importance for comparative purposes, as far as these notes are concerned. Other factors are of greater importance.

Quality of Material.—No attempt to discuss this very important matter is made. It is one for a metallurgist to deal with, as it involves such difficult and highly technical considerations as the composition and structure of cast iron and the growth of cast iron.

Piston-ring Troubles.—Beyond an occasional broken ring the only difficulty that the operator will experience is the gumming of the rings in the grooves. This may be produced either by an excessive amount of fuel oil which remains in the cylinder in an unconsumed condition or by an overabundance of lubricating oil. The solution of the latter trouble is simple; all that need be done is the reduction of the quantity supplied or removal of part of the oil by control rings.

Piston-ring Life.—Piston-ring life in Diesels is affected, in addition to normal ring wear, by the fuel, the lubricating oil, and the type of service. The factors indicating ring troubles are few. Excessive blow-by is probably the first sign of extensive compression-ring wear. This may be evidenced by excessive fumes blown from the breather or, if the breather has a filter cap, by rapid blocking up of the cap. On larger Diesels blow-by is more audible and can be heard by listening while standing alongside the crankcase.

Blow-by removes lubricating oil from the cylinder and so increases liner and ring wear. Lubricating-oil consumption generally increases, and hence a record of oil use is most valuable in determining the condition of the piston rings.

A continuous light- to dark-blue smoke in the exhaust is one of the most common signs. Note that this is a continuous smoke, for often there is an oil smoke resulting from excess oil in the cylinders due to operating or starting conditions that will continue for a fraction of an hour and still not be serious. Loss of power and increase of fuel consumption are information that should be considered in predicting the need of ring replacement.

Measurements of rings are unreliable for determining the time of renewal, because cases will be found where rings that are only slightly worn should be renewed because they are not fitting the cylinder properly or are failing to function for some cause other than wear. The real criteria are blow-by and dry spots on the cylinder caused by blow-by carrying out lubricating oil.

Cause of Ring Wear.—Excessive ring wear and poor oil control are caused by some of the following factors:

1. Inadequate lubrication.
2. Excessive piston heat.
3. Accumulated wear of other associated parts, overloading the rings with oil.
4. Rings damaged during installation.

5. Rings not free to operate in grooves (sticking rings).
6. Improper lubricant for operating conditions.
7. Dust and dirt in intake air and absence of air filter.
8. Dirt in lubricating oil and fuel oil due to absence of oil filters.

Piston-ring Maintenance.—Preventive maintenance is important. When after a reasonable life an engine is down for general examination it is well to figure from examination of the rings whether they can go through an equal time cycle again or if they have to be replaced at an early date. Since much of the cost of replacing rings is the labor and time loss of opening up the engine, then part of the extra cost is saved by eliminating another shutdown for ring replacement only.

Under preventive maintenance should be considered means to minimize ring sticking, which is most commonly caused by lubricating- and fuel-oil decomposition products. These products deposit in the ring grooves, behind the ring and in the side clearance space, causing the ring to stick. Several methods have been employed to reduce sticking and include use of special types of rings, means for conducting heat from the top rings more rapidly, and using special lubricating oils and fuel additives.

A ring that is stuck in the groove and shows heat stains, scores, and lack of cylinder contact should be replaced, for once a ring is stuck and operates in this condition, the sealing qualities of the ring are destroyed. Rings that are sticking, that is, are not free in the grooves and are sluggish, should be carefully removed and cleaned up.

Removal of stuck rings can usually be facilitated by soaking the piston in kerosene. When the rings are burnt into place and cannot be freed by this method it is necessary to break them out. Do not use a cape chisel for this, as it is sure to damage the piston groove. Generally a small portion of the ring can be broken out with a brass implement applied to one end of the ring, and the rest of the ring can be broken out with a suitable brass wedge.

After the rings are all free and as much of the carbonaceous material cleaned off as possible, they may be removed most easily by sliding brass strips ½ to ¾ in. wide under the ring (Fig. 140). At least three such strips should be used, one under each end of the ring and one diametrically opposite. The rings can, then, be slid off without scratching the piston and without snapping into any of the other piston-ring grooves. It is not advisable to clean with emery paper so long as the piston is perfectly smooth. Each time the engine is laid up for a few hours a small quantity of kerosene should be injected into the cylinder through the air-admission valve. This kerosene will remove any carbon on the piston and rings that is in the process of formation. The exhaust valve, in such event, is best blocked open to allow the vapor to escape.

After the rings have been removed, all gum and carbon formations should be removed to restore the bottom-of-groove clearance. Piston grooves should be checked carefully. There is probably some taper wear, but the worst conditions to look for are steps in the side faces of the groove. Such conditions must be corrected by regrooving, as a new ring with its full thickness will soon be damaged.

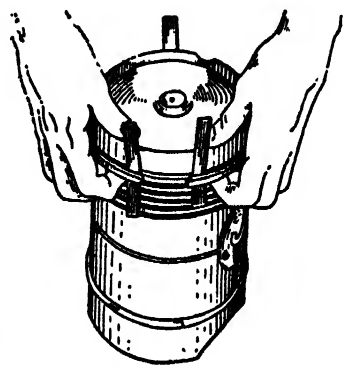


FIG. 140.—Using brass strips to remove piston rings.

After regrooving, the groove is considerably wider, and then arises the question of whether to put in a single overwidth ring or to use two narrow rings in the one groove. Authorities differ as to which will give best results. However, when the cylinder has considerable taper it would appear better to install two narrow rings in the widened groove than to use one wide ring, as less ring wear should be required to obtain perfect contact of the full ring face with the cylinder surface.

The grooves, of course, should be cleaned thoroughly under any condition, and the oil-drain holes in the oil-control groove reamed out with a drill. Ventilated oil rings may be, more or less filled with carbonaceous matter, usually soluble in kerosene. In cleaning rings and grooves be careful not to nick or damage them. The edge of both should feel smooth but sharp and without burrs. Many engineers round the top corner of compression rings to enable the ring to ride over the oil film; excellent results are reported.

For records, at least, check the piston size, and also record measurements of the bore for taper and out of round.

Even the best of rings cannot function properly unless they are fitted properly. They should fit free enough in the groove so that they will fall to the bottom of the groove when the piston is placed horizontal. Rings that have insufficient side clearance are liable to stick in their grooves after a short period of operation, and therefore it is essential that ample side clearance be provided when the rings are installed. Some engines require larger clearances than others. However, as a general rule, the clearances given in Table VII may be used as a guide. Ring side clearance can best be determined by feeler gages inserted between the side of the ring and the side of the ring groove. This clearance should be checked all the way around the ring, as sometimes a ring is warped out of shape when installed.

When the ring is on the piston ample end, or gap, clearance is as essential as ample side clearance. If a ring has insufficient end clearance, there is danger of the two ends of the joint butting and causing serious damage when the ring expands as the result of heat in operation. Unless otherwise recommended by the engine builder, it is suggested that rings be installed with the gap clearances given in Table VII. This clearance should be measured with feeler gages when the ring is placed in the smallest part of the cylinder.

Ring depth, or radial clearance, can be determined best by comparing depth of the ring groove with radial thickness of the ring. This comparison will always be the minimum clearance, as there is a certain amount of clearance between piston and cylinder wall.

In case it is necessary to reduce ring widths to fit the groove properly, the flat side of the ring toward the firing end of the piston may be rubbed down on a sheet of emery cloth laid on a flat surface.

Rings are often pinned to prevent rotation and protect the gaps where they pass over ports in two-cycle engines. Do not remove the pins, and replacement rings should be milled for the dowels, the same as the rings that were removed. When pistons with doweled rings are being placed in the cylinder, great care should be taken to see that the rings are properly located so that they will fit over the dowel without binding or fouling.

TABLE VII.—PISTON-RING DIMENSIONS

Ring width															
Standard ring widths:															
Nominal size, in.	1/8	5/32	3/16	1/4	5/16	3/8	7/16	1/2	9/16	5/8	11/16	3/4	7/8	1
Actual width, max., in.	0.124	0.155	0.1865	0.249	0.3115	0.374	0.4365	0.4960	0.5615	0.624	0.6865	0.749	0.874	0.999
min., in.	0.123	0.164	0.1855	0.248	0.3105	0.373	0.4355	0.4945	0.5605	0.623	0.6855	0.748	0.873	0.998
Ring side clearance														
Side clearances, min.:														
Upper compression, in.	0.003	0.003	0.004	0.004	0.005	0.006	0.006	0.007	0.008	0.008	0.009	0.010	0.011	0.012
Lower compression, in.	0.002	0.002	0.003	0.003	0.004	0.005	0.005	0.006	0.007	0.007	0.008	0.009	0.010	0.011
Oil control, in.	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.003	0.003	0.004	0.004	0.005	0.005	0.005
Top ring on two-cycle engines, add 50 per cent to ring side clearance. Side clearance should also be increased with ring diameter; add 0.001 in. to foregoing clearance for every 5 in. of diameter.														
Ring gap														
Ring diameter, in.	3 to 4	4 to 5	5 to 6	6 to 7	7 to 8	8 to 9	10 to 12	12 to 14	14 to 16	16 to 18	18 to 20	20 to 24	24 to 28	28 to 32
Ring gap, in.:														
Top rings	0.024	0.035	0.036	0.042	0.048	0.054	0.072	0.084	0.096	0.108	0.120	0.144	0.168	0.192
Other rings	0.016	0.020	0.024	0.028	0.032	0.036	0.048	0.056	0.064	0.072	0.080	0.096	0.112	0.128
Ring depth clearance														
Ring diameter, in.	4	5	6	7	8	9	10	11	12	13	14	15	16	17
Groove clearance behind ring:														
Compression and oil scraper	0.022	0.025	0.028	0.031	0.034	0.037	0.040	0.043	0.046	0.049	0.052	0.055	0.058	0.061
Super drainoil	0.042	0.055	0.048	0.051	0.054	0.057	0.060	0.063	0.066	0.069	0.072	0.075	0.078	0.081

Oil-control Rings.—The type of oil-control ring to be used depends upon how much oil control is necessary and upon the arrangements provided for draining oil from the ring grooves. For oil grooves where excessive lubrication is not acute and when no drainage is provided for oil in back of the grooves, a plain oil scraper ring such as shown at 9 (Fig. 134) is often used. The upper edge of this ring is beveled so that the ring will ride up over the oil film on its upstroke, and the lower edge is undercut so that it will scrape excess oil down the cylinder wall. Oil drainage with this type of ring is sometimes provided by holes drilled through the piston wall from an oil groove below the oil ring.

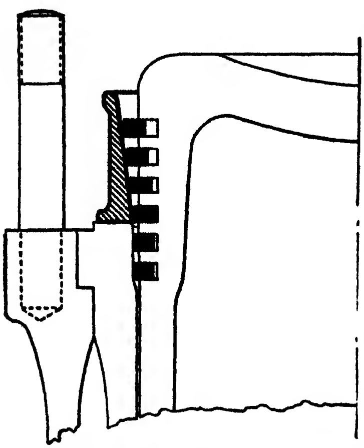


FIG. 141.—Use of taper pot to insert piston in cylinder.

When the scraper-type ring does not provide sufficient oil control, and whenever drainage is provided by holes or slots through the back of the oil-ring groove, a slotted, or ventilated, oil ring as in 10 (Fig. 134) may be used. Oil scraped from the cylinder surface flows through the slots in the ring and drains back to the crankcase through the holes at the back of the ring groove.

In replacing oil rings for better control, do not use ventilated rings where they must pass over a port. It is also advisable to keep in mind that too drastic control in the lower end of the skirt is liable to prevent sufficient oil remaining on the bearing surfaces of the piston. Where possible, the most drastic oil control should be above the piston pin, thereby eliminating all excessive lubricating-oil flow to the combustion space. Some large engines are troubled with insufficient cylinder lubrication, a condition that can be helped by turning the oil scraper ring upside down so that it will carry lubricant up into the cylinder.

Installing Piston Rings.—Before reinserting the rings on the piston, the grooves should be clean and wiped with a cloth, not waste, as only a small piece of lint or particle of dirt may make the ring stick. If the piston is put into the cylinder with rings partially stuck, the condition will become worse, not better, when the Diesel goes into operation.

Ring strips, of sheet brass should be used when replacing the rings, in the same way as when the rings were removed from the piston.

The easiest way to enter the piston and rings into the cylinder is with a pot, or sleeve, tapered 3 in. per foot and centered on the cylinder bore as in Fig. 141. With liberal lubrication the rings will be found to slide in without difficulty. If a ring pot is not available, the rings may be held against the bottom of their

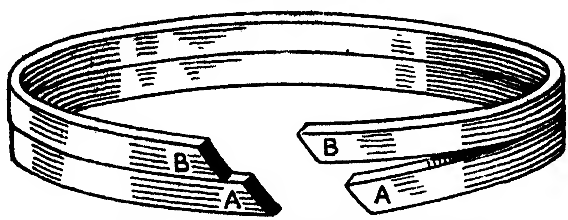


Fig. 142.—Checking flatness of piston ring.

grooves by tying them tightly with a piece of heavy string or light rope. As the edge of each ring slips into the cylinder the rope used to tie it may be removed. Sometimes the rings may be held by a thin metal band or wire; but when using metal, care should be exercised not to scratch the ring surface.

Warped Rings.—In removing a piston ring it may become warped so that the ring does not lie flat. Unless corrected, such a ring will bind in the groove when reinserted. To check a ring for flatness, place it on a second ring, as shown in Fig. 142. If warping is found, the best thing to do is to discard the ring.

CHAPTER XII

CONNECTING RODS

TYPES AND ADJUSTMENTS

General.—In the main, Diesel-engine builders have adopted the type of connecting rod long used on marine steam engines. This is the so-called "marine" type, in which the wear of the bearings is not taken up by a wedge adjustment, as is usual on steam engines, but by a reduction in the shims held between the two halves of the bearing.

At one time a few builders employed wedge adjustment, and one or two used the strap type of rod end. Nether is attractive for the typical Diesel, however, for, until the engine comes up to

FIG. 143.—Type of rod used on McIntosh & Seymour box-frame Diesel.

speed, the forces exerted on the bearing may be heavy enough to injure the wedge or bolts.

At the present time, the vast majority of designers have adopted rods with the crankpin end of the marine type and with the piston-pin end provided with an eye, in which is fitted a bushing to act as the piston-pin bearing. In this chapter are shown several departures from even this standard design.

McIntosh & Seymour Rod.—The connecting rod of the McIntosh & Seymour Diesels built prior to 1934 was of the design shown in Fig. 143. It will be observed that both rod ends have marine-type bearings. Some trouble was experienced with this design, for the bolts binding the piston-pin bearing halves had a tendency to break. This may seem surprising, since the only stress on these bolts is the inertia force of the upper half of the piston-pin bearing at the end of the exhaust stroke; this inertia force is low. The real cause of the leakage was probably the failure of the operator to draw up the nuts tightly against the shims. This permitted considerable play and increased the bolt stress.

Later, American Locomotive Company, owner of McIntosh & Seymour, adopted the rod design shown in Fig. 103, for its four-cycle engine. In this arrangement the piston pin is supported by a bushing, which, in turn, seats in the eye of the rod. The crankpin bearing is formed by a bearing shell resting in the enlarged end of the rod and in the bearing cap. On large Alco engines the crankpin end is of the two-piece design, bolted to the foot of the rod, as in Fig. 143.

Alco-Sulzer Rod.—The two-cycle Diesel built by American Locomotive Works under Sulzer Brothers design has a rod with an eye at the upper end, as will be seen by inspection of Fig. 125.

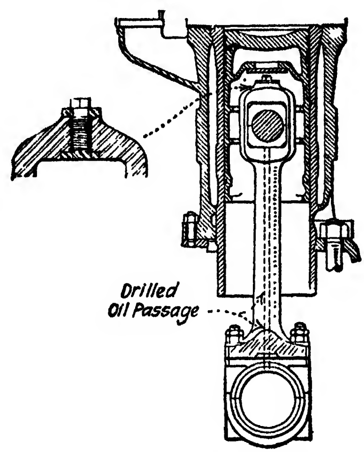


FIG. 144.—Busch-Sulzer Diesel connecting rod.

Busch-Sulzer Brothers Diesel Connecting Rod.—The rod design of the four-cycle engine has the big end of the marine type separate from the rod itself. The piston-pin end is solid, having phosphor-bronze bearing shells and an adjusting screw, as outlined in Fig. 144.

Since the big end is separate, the rod length is corrected by the insertion of separators between the rod and the big end. The crankpin is lubricated by a drilled passage in the shaft, as indicated in the illustration. The rod is drilled its entire length, and the piston pin receives its lubrication through this passage. The passage in the rod registers with the oil line in the crankshaft once per revolution, thus obtaining the proper amount of lubricant. The crankpin bearing has a central circumferential groove, however, which allows oil to enter the rod passage at all times.

On its two-cycle Hesselman Diesel, the piston and rod arrangements shown in Figs 128 and 129 are employed. The departure from standard design is at the piston-pin end, which has already been discussed.

De La Vergne Diesel Connecting Rod.—On the solid-injection Diesel engine manufactured by the De La Vergne Machine Company the rod follows the lines of Fig. 145.

The rod is drilled for the lubrication of the piston pin. To insure the rod against losing the oil after a shutdown, a check valve is placed in the bore just above the crankpin bearing.

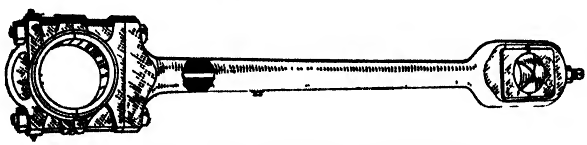


Fig. 145.—De La Vergne connecting rod.

Fulton Diesel Connecting Rod.—The connecting rod of the Fulton Diesel is illustrated in Fig. 146. The bearing for the crankpin is somewhat unusual. The bolts are not a drive fit, as in many engines, but, instead, the holes are drilled to allow radial clearance. The two halves of the housing form a male and female joint. The result is that there is no bending stress

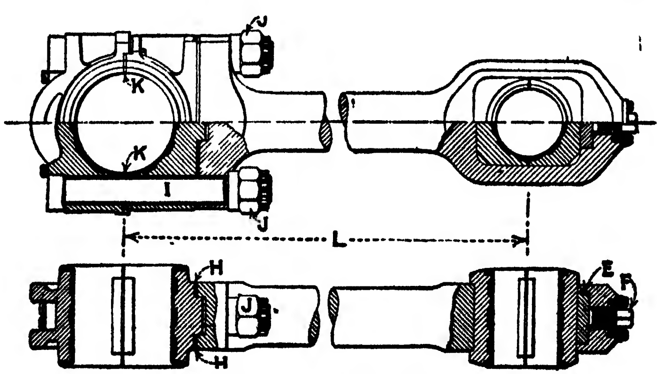


Fig. 146.—Connecting rod of Fulton engine.

in the bolts. This, the builder claims, largely removes the trouble of broken big-end bolts.

The piston bearing is adjusted by the top screw *F*. This, after adjustment, is held fast by the locking plate.

General Motors Diesel Rod.—The General Motors four-cycle Winton Diesel shows in the rod (Fig. 147) the characteristics of the factory, as the piston end of the rod is quite like that used on some automobiles, the Winton factory being an out growth of the Winton automobile factory. The pin is not fastened in the piston with a bearing in the rod, as usual. The pin is provided with two end bearings pressed into the piston, whereas the rod is clamped to the

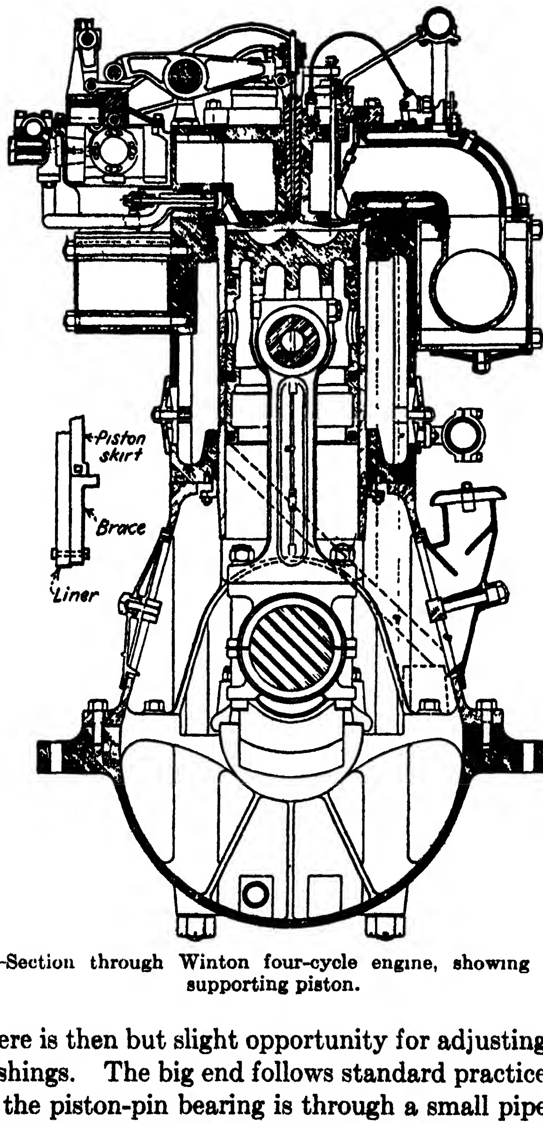


FIG. 147.—Section through Winton four-cycle engine, showing methods of supporting piston.

pin. There is then but slight opportunity for adjusting the wear of the bushings. The big end follows standard practice. Lubrication of the piston-pin bearing is through a small pipe clamped to the rod and leading from the crank bearings up to the hollow piston pin.

Nordberg Connecting Rod.—Many of the Nordberg engines are two-cycle with crossheads. This requires a modification of the usual rod by splitting the upper end into two parts. Each part carries a bearing into which the end of the crosshead or crosshead pin fits (Fig. 148). The piston rod is screwed into the crosshead block as shown in Fig. 102. The crosshead block carries two shoes of the bored type.

FIG. 148.—Nordberg connecting rod

The Nordberg trunk-piston Diesels have rods with the piston-pin end of eye design (Fig. 102).

Fairbanks, Morse Connecting Rod.—The connecting rod of the Fairbanks, Morse two-cycle Diesel is shown in Fig. 149. The piston-pin end is a solid eye into which is slipped the split bronze bearing shell. No adjustment is possible save by replacement of the shell, but wear is insignificant, and the detail is not objectionable. The crankpin end is of the marine type, with the babbitt cast onto the cast-steel bearing.

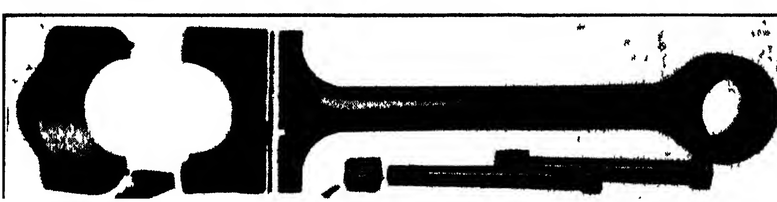


FIG. 149.—Connecting rod used on Fairbanks, Morse engines.

In the 16 by 20-in. engine the rod is drilled to carry lubricating oil to the piston pin and piston crown, but on the 14 by 17-in. engine the piston pin is lubricated by oil picked up from the cylinder wall by a scraper at one end of the pin.

On a two-cycle engine the pressure on the piston-pin bearing is always downward; that is, the contact is between the pin and the lower, or crank, end of the piston-pin bearing shell. As a consequence, the lubrication of the piston pin is not assured. This has led Fairbanks, Morse & Company to adopt a needle, or

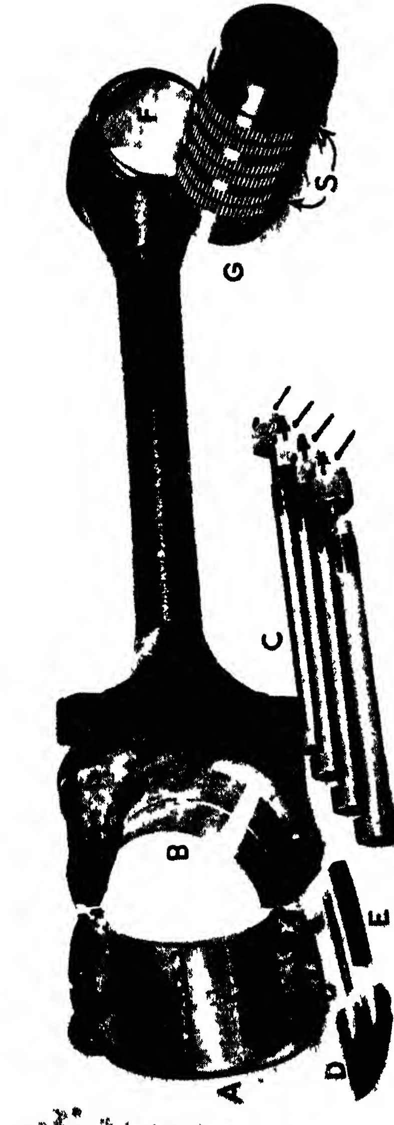


FIG 150 —Fairbanks Morse rod with babbitted crank bearing and needle bearing for the piston pin quill, bearing for its two-cycle engines. This type of rod is shown in Fig. 150.

General.—Each particular make of engine has its own individual design of rod, but space limitations prevent more than mention of those that are markedly different.

Lightweight Rods.—When the rotative speeds are high, the problem of inertia forces due to heavy reciprocating parts is serious. To reduce the weight, some designers use a duralumin rod with reasonable satisfaction. Other designers make the rod of channel section and reduce the web as far as possible, with a view to obtaining a lightweight rod.

Bearing Materials.—The materials used for bearings in Diesel engines include ferrous and non-ferrous metals. The one selected should depend on its own properties for resisting the loads that it will bear, for every material is particularly adapted for a certain application.

In making a selection, the load to be carried and the revolutions of the shaft must be considered, as well as the coefficient of friction; the plasticity, or the ability of the bearing metal to conform to the shaft; and the surrounding conditions, such as moisture, dirt, and chemical fumes. Even after selecting a material that is known to have ample strength, compression, and wear resistance, the matter of lubrication must receive consideration, for it is indifferent engineering to select a material requiring frequent lubrication and then put it in an inaccessible location on an engine, where it may be easily overlooked by the engineer and oiler.

As a group, non-ferrous metals are more plastic and have self-lubricating qualities not found in steel. Of the base metals may be mentioned copper, lead, tin, and zinc, copper being the hardest and lead the softest. With the addition of alloys a large number of useful metals are obtained, adapted for a wide range of loads and conditions. A plastic metal is required for a high-speed shaft with a light load, whereas for slow-speed, heavy-duty application a less plastic one may be better.

Copper-base Metal.—Copper-base bearing metals include the leaded bronzes of copper, tin, and lead. Those with a high lead content are for bearings and bushings occasionally lubricated, but they are not so strong as the low-lead metals, nor will they stand the loads and the severe working conditions.

For bearings for heavy service, copper 86 to 89 per cent, tin 9 to 11 per cent, lead 1 to 2.5 per cent, such as SAE 63, may be considered, but they must be lubricated. Another with copper 80 per cent, tin 10 per cent, lead 10 per cent, deoxidized with phosphorus, is one of the standard bearing bronzes and is well suited for bushings and bearings that are lubricated and subject to moderate loads.

By the addition of nickel to phosphor bronze, giving the composition: copper 84 per cent, nickel 3.5 per cent, tin 10 per cent, and lead 2.5 per cent, deoxidized with phosphorus to give a Brinell hardness of 80 to 93, a metal is secured that is strong and has a small loss of strength on heating. This makes it applicable for bearings where a metal is required that will lose little of its strength from friction heat.

One of the synthetic bronzes is made up with copper 73 per cent, tin 12.5 per cent, lead 10 per cent, and graphite 4.5 per cent. This has a spongelike structure with the graphite uniformly distributed and is capable of absorbing, as a sponge does water, between 2 and 3 per cent by weight of lubricating oil. It will stand high compressive loads with slight deformation, has good self-lubricating properties, but gives better service when lubricated. Requiring little lubrication and not seizing or injuring the shaft, it is suitable for bearings liable to receive little attention such as piston-pin and crankpin bearings.

Copper with cadmium and silver is one of the newer alloys for bearings and bushings. The addition of cadmium alone to copper will produce a hard metal, but increased ductility is given by adding a small percentage of silver. Among the advantages claimed are: high factor of safety at elevated temperature, a higher melting temperature than babbitt, and higher physical properties than the tin-base alloys at all operating temperatures. It has been used in Diesel engines, especially high-speed units.

Lead-base alloys having tin and antimony contents have fair hardness and compressive strength. They are cheap and have good (if carrying a large percentage of lead) self-lubricating qualities. Such a bearing metal is A.S.T.M. specification B 67-28, with lead 87.75 to 88.5 per cent, tin 3.25 to 5.50 per cent, and antimony 9 to 10 per cent; this has a Brinell hardness of about 10. Another is babbitt No. 4, having 12 to 13 per cent antimony and the balance lead. This is good only for light loads.

Copper and antimony added to a tin base give bearing metals that are tougher than those with a lead base. Babbitts, named after the inventor Isaac Babbitt, with an original composition of tin 88 per cent, copper 3.70 per cent, and antimony 7.40 per cent, are particularly suitable for bearings and bushings subject to pounding. They may be classed as "soft," "medium," and "hard," depending on the percentages of copper and antimony:

Soft Babbitt	Per Cent	Hard Babbitt	Per Cent
Tin	88.9	Tin	80
Copper	3.70	Copper	10
Antimony	7.4	Antimony	10

When the pressure is high, larger percentages of copper and antimony are required than if low; but if the antimony exceeds 15 per cent, the metal becomes brittle. Tin-base alloys are superior to, but more expensive than, lead and zinc.

Piston-pin Bearings.—Since the heat absorbed from the piston by the piston pin raises the temperature of the piston-pin bearing, babbitted bearings are seldom employed at the piston pin, for the melting temperatures of all babbitts are low. Furthermore, lubrication of the piston pin is not always assured. These considerations lead to the employment of bronze bushings, needle bearings, or roller bearings.

Bearing bronzes suitable for piston-pin bearings are nearly always alloyed with 80 per cent copper with not less than 10 per cent of tin and from 0 to 10 per cent of lead, phosphorus being used to obtain thorough deoxidation of the melt. The 80-20 copper-tin alloy has maximum hardness, which is ideally desirable for the Diesel but will work only with steel piston pins having highly polished glass-hard surfaces, and has the drawback of brittleness. The 80-10-10 copper-tin-lead bronze with phosphorus deoxidation has ample ductility but is generally considered too soft for Diesel-engine work. Hence the average bronze commonly used contains 80 per cent copper with tin varying between 10 and 20 per cent and the balance lead.

Needle Bearings.—Of late, various engine builders have turned to needle bearings for the support of piston pins. By reason of the small clearances permissible with needle, or small-roller, bearings there is less tendency toward bearing pounding.

The most important of the many reasons for the rapidly increasing use of needle bearings for piston-pin and crosshead-pin mountings is that they alone permit total clearances of an absolute minimum, so that pounding is not experienced. With a proper needle-bearing design the total clearance need not exceed 0.0005 in. on small engines and 0.004 in. on the largest sizes, or roughly only 50 per cent of that customary for bronze bushings.

Needle bearings referred to above are cageless roller bearings using small-diameter rollers which are relatively long with respect to their diameters (Fig. 151).

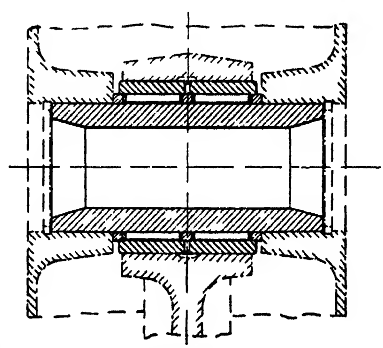


Fig. 151.—Cross section through a needle bearing.

Aside from the advantage of minimum clearance to eliminate knocking, or pound, needle piston-pin bearings also reduce weight and inertia forces because their load-carrying capacity is high, and the space required is no greater in cross section than required for a bronze bushing on a given pin diameter. Assuming that the rollers were a continuous steel ring, and omitting the interstices between individual rollers, there would still be a weight saving over bronze in the order of magnitude of 1 to 1.14, the reduction of inertia being a direct function of the reduction of weight.

Piston slap is very often caused by the static friction of the piston pin in the rod or piston at the instant of reversal of direction of pressure. Here, again, the needle functions to eliminate an annoying condition because it is primarily an antifriction bearing, and the static, or break-away, friction is negligible and is not proportional to load. Consequently, piston slap is not experienced with quill-bearing pins under ordinary piston clearances for good operating conditions.

Lubrication at the piston pin can be reduced materially with the use of needle bearings, for two reasons: First, the actual volume of lubricant needed is small, as this type of bearing distributes the lubricant to the best advantage through the loaded area. Second, because of reduced friction, cooling of the pin is not a factor, and additional lubricant is not needed to conduct this frictional heat away. Where necessary, excess lubricant, however, can be used, and piston cooling obtained. This excess oil, not being under bearing load, is available for the maximum dissipation of piston heat.

In large installations, multiple rows of rollers can be used to advantage, and the total friction value of the bearing thereby reduced. The ratio of roller diameter to roller length should not exceed 1:10. Where multiple rows of rollers are used, they may be separated one from the other by floating rings which should have clearance both over the pin and in the bore of the sleeve, or outer race. A needle bearing is shown in Fig. 150.

Needle-bearing Clearances.—Radical clearances can and should be held to a minimum from 0.0005 in. on engines having pins under 1 in., 0.0075 in. on pins between 1 and 2 in., 0.001 in. on pins from 2 to 3 in., and 0.0015 to 0.002 in. minimum on larger sizes up to 6 in. Added to these minimum radial clearances will be additional clearance necessary by virtue of heat conditions which may cause a differential in expansion between the inner and outer races.

The amount of circumferential clearance, or clearance from roller to roller, should usually be set at approximately 0.001 in. per roller, and the total should in no case exceed 0.9 roller diameter. This latter feature is necessary in order to prevent installing an extra roller over and above the calculated quantity which would reduce the circumferential clearance below the minimum requirement.

Although this type of bearing for these locations has had a phenomenal success, its application is still new, and careful consideration to details of design must be given to obtain the satisfactory results that are possible.

Fairbanks, Morse & Company employs the needle-type bearing for the piston pins of its vertical, two-stroke cycle, medium-speed Diesel. General Motors also applies this type to its two-cycle, high-speed Diesel.

Lubricating Piston-pin Bearings.—Various methods are followed in the supplying of lubricating oil to piston-pin bearings.

In two-cycle engines where the crankcase is used as the air compressor, the cylinders must be fed lubricating oil by a mechanical force-feed pump. The piston pin is supplied with scoops at the ends which pick up oil from the cylinder walls, as in Fig. 152, and feed the oil to the center of the piston pins; the oil then flows through ports to the bearings.

The same plan has been employed on four-cycle engines; but with the advent of pressure feed to all the bearings, at present practically all four-cycle engines have their piston pins supplied with oil through drilled connecting rods or through a tube clamped to the rod.

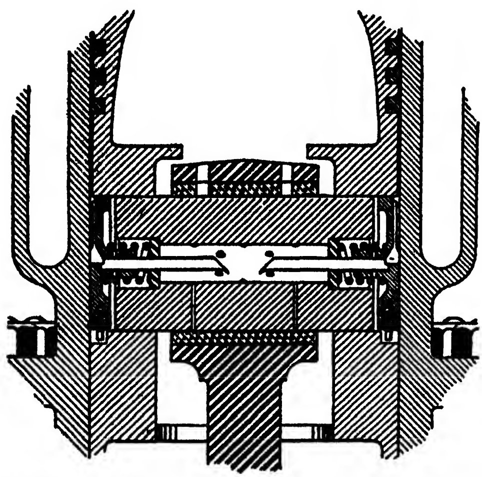


FIG. 152.—Lubricating the piston pin of a two-cycle engine.

Pounding at Piston Pins.—Many operators believe that when there is too much clearance between crankpin and its bearing, pounding will occur at the beginning of the power stroke but feel that the piston pin never pounds in its bearing, since it moves but slightly as the rod swings through its arc. This is by no means true.

The facts are that a crankpin bearing will never pound when the engine is running at its rated speed. Too much clearance between crankpin and bearing will cause a pound when the engine is starting up and before its rated speed is reached. Likewise, the bearing may pound when the engine is being shut down.

A crank pound may occur if the fuel is cut off from a particular cylinder. Where this is done, the only pressure within the cylinder is due to the reexpansion of the compressed-air charge.

This may be so small that it is less than the inertia force, so a change in pressure may occur at mid-stroke of the piston, causing a pound at the crankpin.

Basically, however, a normal, single-acting, four-cycle Diesel will not pound at the crank bearing at the engine's designed speed.

The pounding one hears on some engines is chiefly at the piston-pin bearings. If there is too much clearance, the piston pin will pound when the piston is about midway on the stroke, on both the suction and the exhaust strokes.

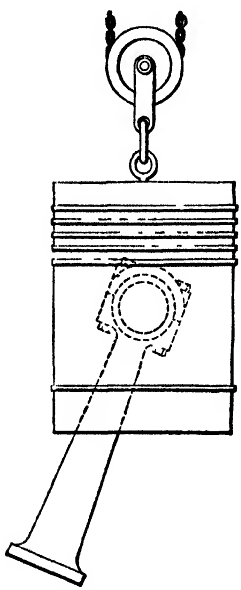


FIG. 153.—Swinging rod to check pin clearance.

Checking Piston-pin Clearance.—Clearance for piston-pin bearings of four-cycle engines should be kept finer than for ordinary bearings which rotate continuously in one direction. Some operators hold the clearance to 0.00075 in. per inch of pin diameter (0.003 in. on a 4-in. bearing); others go so far as to leave no measurable clearance at the pin bearing.

To determine if the clearance is sufficient, an easy method is to hang the piston assembled complete with its rod and to swing the rod by hand to test whether it is free (Fig. 153). The rod should fall to a perpendicular position after being swung to one side but not quickly. When the rod is swung to one side, the piston should not tip; if so, there is too little clearance.

Correct clearance may be obtained by removing enough shims to make the bearing bind slightly and then replacing the thinnest possible shim (0.0025 in.). After each removal of shims and during each hand-swinging test, it is essential that the bearing be keyed up just as tight as though it were intended to run; otherwise serious error and a damaged bearing are likely to result. It is also possible, after having removed just the right number of shims to cause binding, to retouch the high spots in the bearing with a scraper, whereupon it will generally be found to swing free after assembly. Close clearances are absolutely necessary in four-cycle and in all double-acting engines because, if pounding once begins as the result of too much slack, it rapidly gets worse and may easily lead to a fracture of the piston-pin bearing or to a breakage of connecting-rod bolts.

Piston-pin clearances on two-cycle, single-acting engines are generally kept about the same as in ordinary rotating bearings (0.001 in. per inch of diameter), because there is no reversal of forces and hence considerably less tendency to pound, whereas the higher temperatures occurring in two-cycle engine bearings make it necessary to allow more slack for expansion and other forms of distortion.

When the piston-pin bearing is a solid bushing, adjustment is not intended. When the wear becomes marked (which will be only after long periods of operation), the bushing should be replaced by a new one.

High-speed Diesels usually have the pins supported in bushings in the piston bosses, as well as in the rod bushing. In other words, the pin is full floating. These pins have little clearance in the bushings. The pin should have merely a slip fit; that is, it should slip through the bushings with no perceptible looseness, or clearance.

Big-end Bearings.—No matter what the design may be, the operator some day is confronted with the problem of a big end that insists on running hot. The first move is to determine whether or not the lubrication has been faulty. In the majority of cases this proves to be the origin of the trouble. Generally the oil pipe or passage has become clogged with dirt or a bit of waste. The remedy is obvious. There is, for some reason, probably due to misalignment of the rod, a tendency for the big-end bearing to wear more rapidly at one end than at the other; or, at times, both ends wear while the center remains in its original condition. This "belling" of the bearing permits the pin pressure to be distributed over a rather small area of the brass. This produces a local heating which forces the babbitt to drag, filling the oil passages and grooves. An additional result of this unequal bearing wear is the scoring of the cylinder on one side. When this bearing wear has occurred, it is imperative that the babbittt be rebored to the pin diameter and the oil grooves cut. Even if the wear seems excessive, it is, as a rule, possible to avoid rebabbitting. Part of the shims between the bearing halves can be removed, and the halves clamped together and rebored to size.

It sometimes happens that part of the babbitt cracks and drags around the pin. This results in heating and a badly scored bearing. If the trouble is local, the rough spots can be smoothed with a scraper, and the bearing can then be placed in service again. In all instances where big-end bearings become so hot that the babbitt is thrown, the engine should not be stopped immediately; rather, the load should be taken off, and the engine turned over very slowly, with the particular cylinder cut out; that is, the fuel is cut off. In rebabbitting rod bearings the same method as described for main bearings can be followed. The babbitt should always be cut on a bevel at the junction of the two halves. The oil grooves should not extend to the bearing edges; and when a pressure-oiling system is used, the oil grooves should be eliminated, since they allow the oil to escape too rapidly.

Side play is of frequent occurrence in Diesel operation. The best method is to tin the bearing sides and run a collar of babbitt around the bore. In addition, the alignment of the rod bearing should be corrected. This collar must be turned square with the pin. It is not necessary to cover the entire side; consequently, the collar can be machined parallel with the side of the big end. Inspection of the connecting-rod bearings should be performed at least every 3 months.

Along with the wear of the big-end bearing occurs the wear of the pin. In old engines that have seen several years of service, the pins may become flattened on one side. This can be corrected by grinding and lapping, but the task requires great care and patience. It is, ordinarily, not difficult to detect a worn rod bearing. The engine will emit a thump or pound on both the suction and the exhaust strokes.

Rebabbitting Bearings.—The engineer will some day face the proposition of renewal of the pin bearings. In some engines the piston-pin or crosshead-pin bearings are phosphor-bronze or Parsons white-metal shells without babbitt linings. These bearings may become scored or pounded out of shape. It is necessary only to place the box on a lathe and rebore to the correct diameter. If enough shims cannot be removed to allow reboring of the bearing, the edges may be faced off on a planer or lathe, followed by reborings.

In the rebabbitting of the crankpin bearing the engineer will find some difficulty. The process is the same as for main-shaft bearings. It is desirable to heat the housing and keep a temperature of about 300°F. before the babbitt is poured. After babbitting it is essential that the big-end bearing be bored

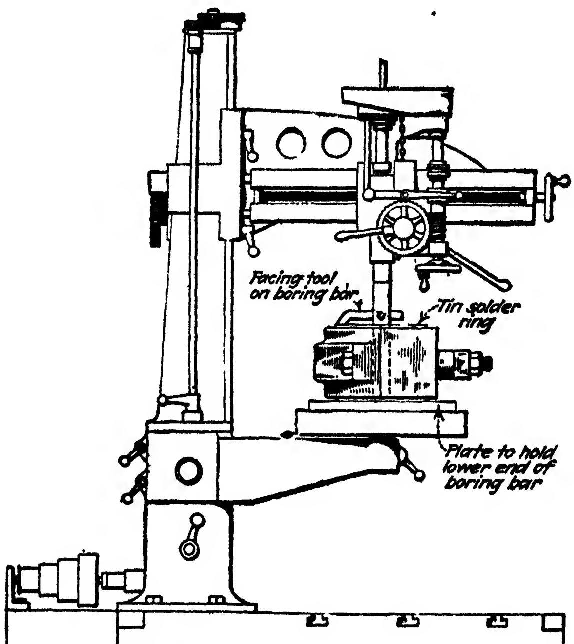


FIG. 154.—Boring crankpin bearing.

exactly square with the side faces. These faces are always made square with the big-end flat surface when machined at the factory.

To machine the bearing, the housing should be placed on a lathe carriage, and the boring tool held on the face plate, or, better still, it can be bored on a drill press. A plate with a center hole to hold the end of the boring bar should be bolted to the drill press table. The housing should be leveled up on the table, and with the cutting tool in the boring bar, the spindle should be turned by hand at first. In this way it is possible to see that the bar is central with the housing and that the housing is square with the bar. The method is outlined in Fig. 154. Such a method will give a bearing bore that is exactly square with the faces and with the flat surface upon which rests the big end of the rod.

Checking Rod Alignment.—After rebabbitting and boring, the assembled rod should be checked for alignment. If the bores of the piston-pin and the crankpin bearings are not parallel, the connecting rod will not move in a vertical plane but will tend to bell the crank bearings as well as to pound badly. Many instances of lengthwise motion of the shaft are due to error in one or more rods.

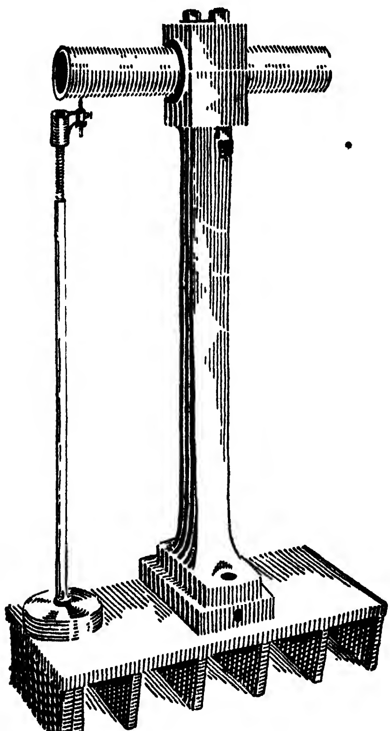


Fig. 155.—Connecting rod gage.

To test the rod, it is necessary to obtain two mandrels of the same diameters as the two pins. These may be turned up out of cast iron and can be made by any mechanic who can handle a lathe. If pipe of the correct diameter is handy, a piece machined true makes a light serviceable mandrel. There is no reason why the mandrels may not be used as the mandrels for babbitting as well.

A gage is made up consisting of a brass disk (steel will do as well) to which is attached a threaded rod of 3/4-in. drill steel. The top of the rod is threaded to receive a brass bushing carrying a sharp point, as shown in Fig. 155. The piston-pin mandrel is put in the bearing, and the bearing halves clamped tight. The rod is now placed on a flat surface, such as the table of a shaper or similar machine. In fact, every engine plant should have a surface plate. The gage is adjusted until the tip will just touch the mandrel as shown. By passing the gage under the opposite end of the pin it is at once apparent if the mandrel is square with the big-end surface of the rod.

To check the crankpin bearing the top half of the bearing is laid flat on a surface (Fig. 156). If it is necessary in order to clear the center boss, the bearing may be supported on thin blocks and leveled with a spirit level. By using a surface gage across

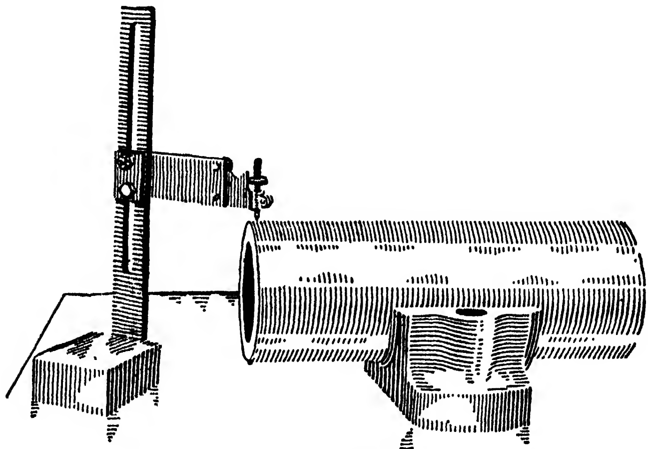


FIG. 156.—Gaging crankpin-bearing alignment.

the tops of the mandrel ends, the correctness of the bearing is quickly determined.

It is possible that the pins are square with the rod and yet may not lie in the same plane. To check this the rod is placed as shown in Fig. 157 with one mandrel supported on V-blocks and

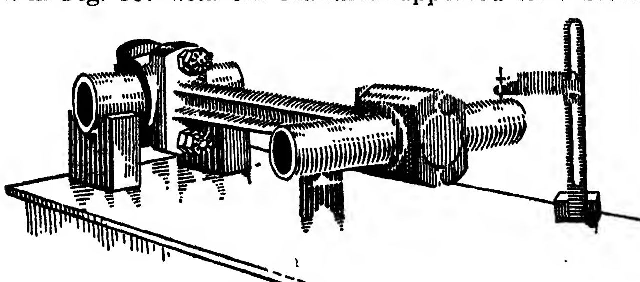


FIG. 157.—Checking parallelism of the pins.

the other end held up by packing pieces. The surface gage will at once determine the fact that the pieces are true or out of parallel. This procedure may seem tiresome to some, but Diesel engineers must understand that care is necessary and that with such precautions a quiet sweet-running engine will result.

After a bearing is rebored, it should be scraped in the same manner as a main bearing.

Even though the two rod bearings may be parallel and in the same plane, it is possible that the rod may not be true when the engine is running, because the bosses on the piston for the piston-pin ends have not been bored at right angles to the piston skirt.

In almost all instances the base of the piston is machined square with the piston skirt. It follows, then, that if the crankpin bearing is square with the lower edge of the piston, it will swing parallel to the piston skirt.

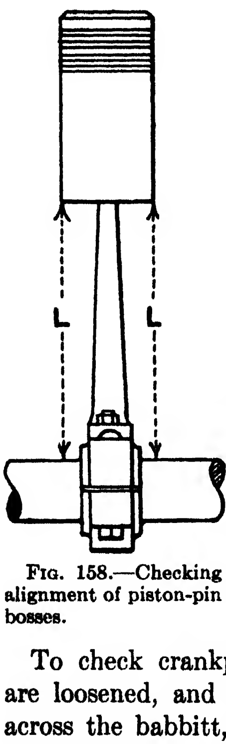


FIG. 158.—Checking alignment of piston-pin bosses.

This alignment can be checked by measuring distances L (Fig. 158) between crankpin mandrel and skirt edge when the piston is held suspended, with the rod in place.

Checking Pin Clearance.—When fitting a new bearing, care must be taken to provide proper clearance. Too small clearance may cause the journal to wipe out the babbitt; too much will allow pounding and excessive oil leakage. Manufacturers' recommendations should be followed, but in their absence the following should be used: For crankpins 2¾ to 3¾ in. in diameter, allow 0.00075 to 0.001 in. per inch of diameter; and for pins over 3¾ in., allow 0.007 to 0.001 in. per inch of diameter.

To check crankpin-bearing clearance, the bearing-cap bolts are loosened, and two lead wires are placed circumferentially across the babbitt, about one-quarter the way from each end. The bearing cap is then drawn up tight with its bolts, flattening out the lead wire. Upon lowering the cap the leads can be removed and their thickness measured with a micrometer. The measured thickness of the flattened lead is the actual bearing clearance.

Lead wire should not be used to check the clearances in small high-speed Diesels. The wire should not be compressed more than about 0.01 in. If more than this is attempted, the wire may be embedded in the babbitt, in which case the compressed wire will indicate more clearance than actually exists. Feeders should be used to check clearance in small engines if the shaft design permits slipping the feeder in at the end of the bearing cap.

Clearance can also be determined by jumping the crankpin bearing with a bar under the bearing box with the crank at top center. To obtain accurate results, a dial gage should be

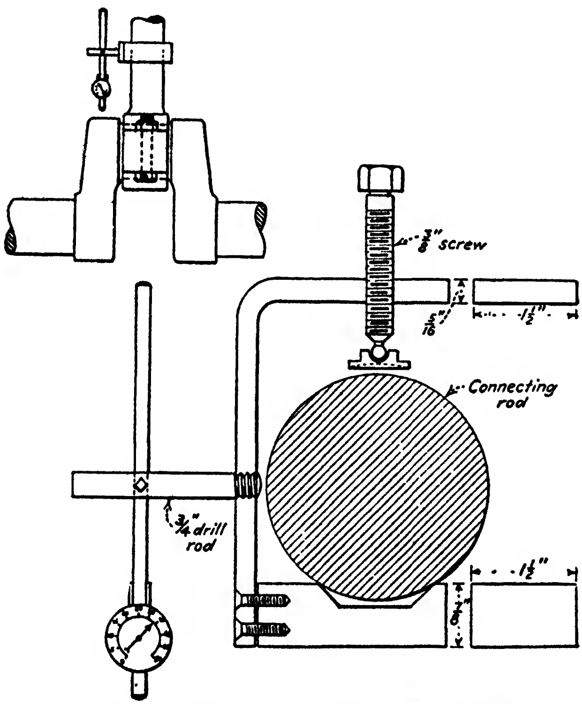


FIG. 159.—Checking bearing clearance with dial gage.

used (Fig. 159) to indicate the amount of movement that is the bearing clearance.

Still another method is to remove some of the shims, until the bearing grips the pin when the bolts are tightened. The bolts are then unshipped, and an amount of shims added to equal the clearance the operator thinks is proper.

Generally when a crankpin bearing cap is removed to check clearance, it is advisable also to remove the upper shell for examination. In small engines this is a simple operation, but in large units with large bearings the job is apt to be a mean one.

A 2 by 4-in. or 2 by 8-in. timber should be placed across the crankcase doors, and the bearing cap lowered onto it as it is removed. By greasing the timber and raising one end the bearing cap can be slid out of the crankcase. Before the top shell can be removed, the piston must be supported in its cylinder. This can be done with a piece of timber, its lower end resting on the engine frame and its upper end resting against the cylinder bore, or by bolting a bracket through holes in the lower part of the cylinder liner; both methods are shown in Fig. 147.

Tightening Crankpin-bearing Bolts.—A general custom followed in tightening up crankpin bolts is to run the nut up by hand and then sledge the end of the wrench until the engineer thinks the nut has been tightened sufficiently. The amount of the blow that he gives the wrench depends upon his judgment or, rather, his energy. He has heard so much about "initial tension" that he knows the bolt must be drawn up—but how much he does not know.

Someone has advanced the suggestion that a wrench could be arranged with a "shear pin," or a release, so that only a predetermined force could be applied to the nut. Such a plan is hardly feasible, for the reason that a wrench, when applied to the nut, must overcome the frictional resistances of the thread contact and of the nut against the bearing housing. The remainder of the applied force elongates the bolt, giving the initial tension that everyone talks about, so no manner of wrench will give a definite bolt tension.

Some engine builders instruct the operator to slip a pipe of some definite length over the wrench and pull up on this pipe, the so-called "cheater." But who can estimate the forces exerted by a willing 135-lb. oiler and a 200-lb. loafer? Or the condition of the threads or contact surface of the nut?

It is possible by the use of a C gage, or large micrometer, to measure the elongation of a bolt. All one needs do is to apply the gage and read the bolt length before the wrench is applied. While sledging, the gage will indicate the new bolt length.

A prevalent assumption is that, if the connecting-rod bolts are given initial tensions slightly above the heaviest load to which the bearing cap will be subjected during operation, the bolts will be subjected to no additional stress until the loading exceeds the initial stress. This is true if we assume that the crank-box parts (and the shims) are incompressible. This, of course, is by no means exact from a scientific standpoint.

Even a forged-steel crank box is compressible, as are also shims.

That a bolt does experience more than the initial tension may be explained as follows:

If two or three pieces of metal as in Fig. 160 are incompressible, then, when clamped together by a bolt, the bolt is stressed, and its total load is equal to the compression stress between the metal pieces. As shown in Fig. 160 *A*, the total stress on the bolt due to tightening (which stretches the bolt and reduces its diameter slightly) is *L*. At the joints this same total load exists, as indicated, and there also exists a reaction *R* at the joints,

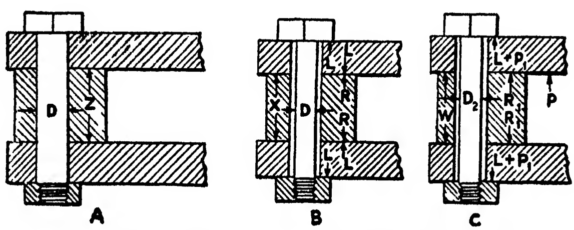


FIG. 160.– Action of non-compressible material.

equal to the bolt loading *L*. If, now, an additional load *P* is imposed, tending to separate the three metal sections as shown in Fig. 160 *B*, it is possible to imagine that for an infinitesimal period of time this additional pressure stretches the bolt a little more. This additional elongation would immediately separate the three metal sections, for it has been assumed that they are totally incompressible. If they separate, then the load on the bolt due to reaction between the blocks disappears; consequently, the only loading is *P*. But to cause the reaction *R* to become zero, the load *P* must be enough so that the total elongation of the bolt due to the load *P* is greater than the original elongation due to the reaction *R*. It follows, then, that, if the applied load (such as the working load on a bearing) is no greater than the reaction *R* due to initial tightening of the nut, there will be no additional stress on the bolt.

If, however, the metal sections are compressible, a different action exists. Suppose that, as in Fig. 161 *A*, we have the three compressible blocks held loosely by a bolt having a diameter of

D. The center block has a thickness of Z. If the nut is tightened, as in Fig. 161 B, the bolt will be stretched and will be reduced in diameter from D to D_1. The metal blocks will be compressed, so they are now not so thick as before. The total stress on the bolt is L, and this is equal and opposite to the reaction R at the joints.

Suppose that the metal blocks are reduced in thickness by 0.0010 in. when the bolt stress is 1,000 lb. (or 0.0001 in. for each 100 lb.) and that the bolt stretches 0.00005 in. per 100 lb. If, now, a working load of P lb. (assumed it to be 500 lb.) is added, it may be assumed that the load on the bolt for an infinitesimal period is $L + P$, or $1,000 + 500 = 1,500$ lb. The additional

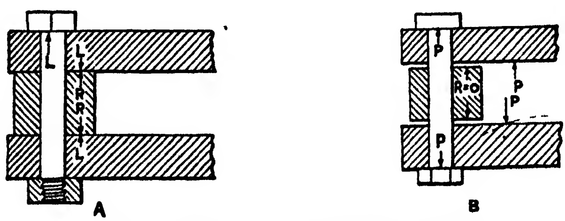


FIG. 161.—Action of compressible material.

bolt stretch under the added 500 lb. is 0.00025 in. (for a load of 1,000 lb. stretched the bolt 0.0005 in.); and since this stretch allows the blocks to expand the same amount, namely, 0.00025 in., the reaction at the joints is reduced by 250 lb., for the metal was compressed 0.001 in. by load of 1,000 lb., so a reduction of 0.00025 in. in the amount the blocks are compressed, reduces the reaction by 250 lb., since the rate is 0.0001 in. per 100 lb.

The reaction at the joint is then $1,000 - 250 = 750$ lb., and as the additional working load P is 500 lb., the total load on the bolt is 1,250 lb.

It follows, then, that if the bearing material is compressible, or if there are interposed between the steel bearing halves a material, such as a shim, that is compressible to any extent, the bolt stress is increased to some extent by the working load over and above the initial stress given the bolt by tightening up. To this fact may be traced some of the failures of crankpin-bearing bolts. In all probability the inertia forces on the crankpin bearing may increase the stress on the bolt over the initial stress by about 25 per cent.

There is, further, an additional stress due to shock, if the change in the load applied to the bearing during operation is high and is applied rapidly.

The amount of initial tension that should be given depends, naturally, upon the maximum operating load to be expected. In the case of the crankpin bearing, the maximum load will be the inertia force at top dead center on the exhaust stroke plus the centrifugal force possessed by the rod end. This can be figured without trouble from data on engine speed, weight of reciprocating parts, and piston stroke. Probably the best way is to obtain the stress information from the engine builder.

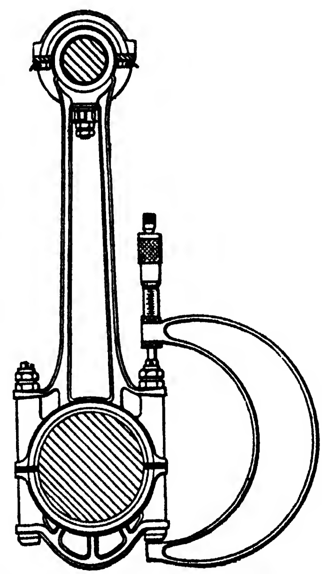


Fig. 162.—Checking elongation of crankpin-bearing bolts.

The actual elongation of a bolt may be measured by a *C* micrometer (Fig. 162). It is necessary under this method to grind a perfect flat surface on both bolt head and threaded end. Each time a bolt is unshipped and again put into service, the free length must be measured.

To understand properly how to determine the proper amount of tightening to give the bolts, one must have a grasp of the meaning and the method of computing the inertia force and bolt tension.

Inertia Force.—To cause an object to move, a force must be exerted. This force increases the velocity of the object to some maximum, after which, even though the force is removed, the object will continue to move at the acquired velocity until it is slowed down by friction or some other opposing force.

This action occurs in the Diesel cylinder. The engine's flywheel is moving at a constant rate; that is, a point on the rim covers a constant number of degrees in a given time interval. Inspection of Fig. 163 shows that, when the crank is at dead center *A*, the piston is at the end of its stroke at *C* and is stationary. The combustion of the gases exerts a force on the piston and connecting rod, so that the crankpin will continue to rotate at a uniform rate and transmit power to the shaft. The piston, which is at rest when combustion starts, must be accelerated at a rate such that the crank may continue its rotation at a uniform rate. It will be observed that, while the crank passes over 30 deg. from A to F, the piston moves from C to F_1 and that while the crank is covering the 30 deg. from F to G, which is equal in angle and time to CF, the piston moves from F_1 to G_1, a distance much greater than CF_1. At some point close to mid-position B of the crank, the piston attains its greatest velocity (it would be exactly at mid-position except for the effect of the connecting-rod angularity).

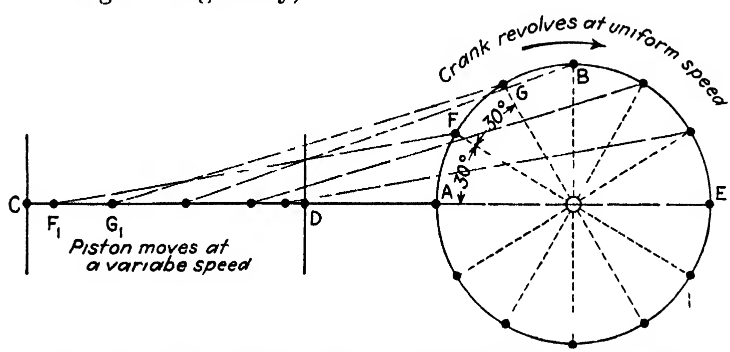


FIG. 163 – Position of piston while crank rotates at constant speed

The crankpin continues its rotation at constant velocity, but, as will be seen, after mid-position is reached, the piston must gradually slow down until it stops at D when the crank reaches outer dead center E. During this slowdown period the crank retards the piston, with the result that all the kinetic energy, or energy of motion, possessed by the piston is consumed in opposing this retardation. In fact, all the energy employed in speeding up the piston and connecting rod is delivered to the crank during the period the piston is slowing down. No energy is lost, but the time during which its effect is felt is delayed.

The same events occur during the exhaust stroke of the engine, the suction stroke, and the compression stroke. Force is needed during the first part of each of these strokes to accelerate the piston and connecting rod, and a force is given back when the piston is slowed down during the second part of each stroke.

Although all parts of the connecting rod have a reciprocating motion, that is, in line with the cylinder axis, these parts also have a motion at right angles to the axis. It is usual to assume that one-third of the rod weight has a reciprocating motion only and that two-thirds is centered about the crankpin and has a circular path. The reciprocating parts are then: (1) the piston, (2) piston pin, and (3) one-third the connecting rod. This gives a close approximation in results to a more accurate analysis.

The force required to accelerate the parts varies continuously but may be found with ease for any crank angle.

It will be seen that the existence of inertia causes the force transmitted along the rod to be greater after the piston reaches mid-position than would be the case if inertia did not exist.

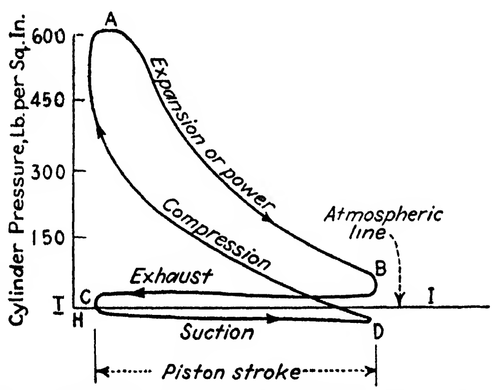


FIG. 164.—Schematic indicator diagram from a four-cycle Diesel.

Inertia, then, actually, makes the force at the piston more uniform. On the compression stroke the same action occurs.

Taking up a four-cycle engine, an indicator diagram of the engine would be somewhat as in Fig. 164, in which the piston stroke is represented by horizontal measurements. The diagram is thus representative of a horizontal engine. It would represent a vertical engine by being turned 90 deg. The pressure in the cylinder at any point is represented to some scale by the height above the atmospheric, or zero-gage, pressure line.

In the diagram, *DA* is the compression stroke; *AB* the power, or expansion, stroke; *BC* the exhaust stroke; and *CD* the suction stroke.

To obtain the net force acting along the rod at all points of the stroke to cause the crank to be turned, the force used in accelerating the parts must be subtracted from the total cylinder pressure for the first part of the stroke, and the force given up by the deceleration of the parts added for the second portion. This force may be calculated quite closely by the formula

$$F = 0.0000284N^2WR(\cos\theta + K\cos 2\theta)\text{ lb.}$$

where

N = r.p.m. of engine.
W = weight of reciprocating parts.
R = crank radius, inches.
L = connecting rod length, inches.
$K = R/L$.
θ = crank angle at given time.

Of this expression, $0.0000284N^2WR$ is constant for any given engine. The portion in the brackets may be computed but is more easily obtained from Table VIII, which gives values of the bracketed expression for various crank angles and various ratios of connecting rod to crank arm, this being

$$\frac{L}{R}, \quad \text{or} \quad \frac{1}{K}$$

To make this discussion concrete, let us assume we have a Diesel with a 10-in. bore and 12-in. stroke, operating at 600 r.p.m. We shall further assume that the compression pressure is 450 lb. per square inch, rising, after fuel ignition occurs, to 650 lb., and that the piston, pin, and one-third rod weigh 150 lb.

With the value of N and W known, the inertia force during one stroke from head to crank dead-center positions may be computed.

In Fig. 165 the power stroke of Fig. 164 has been laid out, giving the pressure-volume relations indicated by the curve AA, identical with AB of Fig. 164. The total gas pressure acting on the piston at the start at A is 51,051 lb., which is arrived at by taking the pressure per square inch, 650, and multiplying it by the piston area (0.7854×10^2).

On the same figure the inertia-force curve FF has been drawn, as computed from the values in Table VIII for a $1/K$ value of 4.5. During the first part of the stroke this force is exerted on the piston; consequently, it reduces the pressure of the gases transmitted through the piston to the connecting rod. For this reason, the portion FP is drawn below the zero line HI, and PF

is above the zero line from the point P to the end of the stroke, for during this latter time the piston is slowing down and exerting a force on the rod.

The two forces AA_1, the gas pressure, and FF_1, the inertia force, when combined give the net positive force RR_1 acting during the power stroke to turn the crank. It will be seen that during the entire stroke RR_1 is always a plus force.

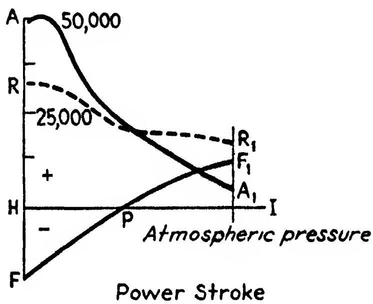


FIG. 165

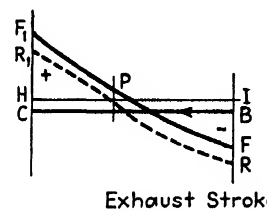


FIG. 166.

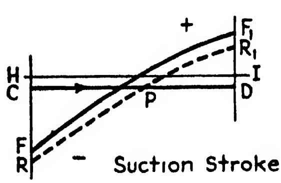


FIG. 167

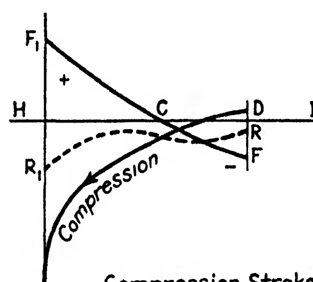


FIG. 168.

During the next stroke, the exhaust stroke (Fig. 166), the piston forces the gas out of the cylinders; and as the force comes from the crank, it may be regarded as *negative*, as shown by BC. The inertia force line FF_1 is the reverse of the line of Fig. 165 and is negative during the first part of the stroke in Fig. 166 and positive during the later part.

Combining the two, the net force RR_1 is obtained. This reveals that until the crank reaches the point P (Fig. 166), the crank and rod are pushing the piston.

As soon as crank position P is passed, the motion of the rod is slowed down by the crank action, and the piston, moving at

TABLE VIII.—VALUE OF ($\cos\theta + K\cos 2\theta$) FOR VARIOUS VALUES OF L/R

θ	L/R or $1/K$					θ
	3.50	3.75	4.00	4.25	4.50	
0	1.286	1.267	1.250	1.235	1.222	360
5	1.277	1.259	1.242	1.228	1.215	355
10	1.253	1.235	1.220	1.206	1.194	350
15	1.213	1.197	1.182	1.170	1.159	345
20	1.159	1.144	1.131	1.120	1.110	340
25	1.090	1.078	1.067	1.058	1.049	335
30	1.009	0.999	0.991	0.984	0.977	330
35	0.917	0.911	0.905	0.900	0.895	325
40	0.816	0.813	0.810	0.807	0.805	320
45	0.707	0.707	0.707	0.707	0.707	315
50	0.593	0.596	0.599	0.602	0.604	310
55	0.476	0.482	0.488	0.493	0.498	305
60	0.357	0.367	0.375	0.382	0.389	300
65	0.239	0.251	0.262	0.271	0.280	295
70	0.123	0.138	0.151	0.162	0.172	290
75	0.011	0.028	0.042	0.055	0.066	285
80	0.095	0.077	0.061	0.048	0.035	280
85	0.194	0.175	0.159	0.145	0.132	275
90	0.286	0.267	0.250	0.235	0.222	270
95	0.368	0.350	0.333	0.319	0.306	265
100	0.442	0.424	0.409	0.395	0.383	260
105	0.506	0.490	0.475	0.463	0.451	255
110	0.561	0.547	0.534	0.522	0.512	250
115	0.606	0.594	0.583	0.574	0.566	245
120	0.643	0.633	0.625	0.618	0.611	240
125	0.671	0.665	0.659	0.654	0.650	235
130	0.692	0.689	0.686	0.683	0.681	230
135	0.707	0.707	0.707	0.707	0.707	225
140	0.716	0.720	0.723	0.725	0.727	220
145	0.722	0.728	0.734	0.739	0.743	215
150	0.723	0.733	0.741	0.748	0.755	210
155	0.723	0.735	0.746	0.755	0.763	205
160	0.721	0.735	0.748	0.760	0.769	200
165	0.718	0.735	0.749	0.762	0.773	195
170	0.717	0.734	0.750	0.764	0.776	190
175	0.715	0.734	0.750	0.764	0.777	185
180	0.714	0.733	0.750	0.765	0.778	180

its existing speed, starts to pull on the rod, exerting the positive force as indicated by the position value of RR_1 beyond the point P.

During the suction stroke (Fig. 167), the crank and rod pull the piston, and the force so used is equal to the negative pressure in the cylinder, as indicated by the line CD; the line is drawn below the zero line HI. The inertia force curve is FF_1, negative at first but becoming positive later in the stroke, and the net force on the piston is represented by RR_1.

On the next stroke, the compression stroke, the crank, acting through the rod and piston, exerts a force to compress the air in the cylinder; consequently, the cylinder pressure force may be regarded as *negative*, as indicated by curve DA (Fig. 168).

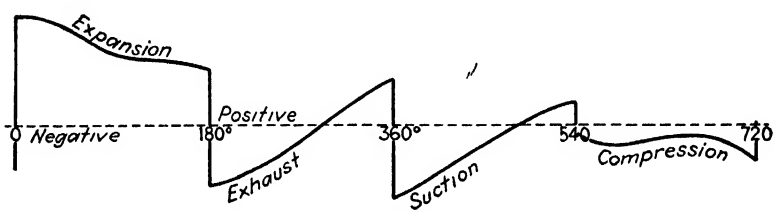


FIG. 169.—Net piston pressure during four strokes.

However, it will be noted that for a small part of the stroke the compression-pressure curve is positive from D to C. This is for the reason, as reference to Fig. 164 will reveal, that the air pressure acting against the open end of the piston is greater than the air pressure in the cylinder, so that for this part of the stroke, because of the external air pressure, the piston exerts a force along the rod to turn the crank.

Combination of the compression curve DA and the inertia force curve RR_1 gives the net force acting on the piston FF_1. It will be seen that this is negative.

The net forces acting at all points in the piston stroke are shown in Fig. 169. These are the RR_1 values of Figs. 165 to 168, properly placed to show positive and negative values but at a reduced scale.

Inertia Force on Crank Bolts.—Inspection will reveal that, when the crank approaches top dead center on the exhaust stroke, the piston and connecting rod try to continue their motion while the crank is trying to bring these parts to rest.

The entire inertia force of these parts exerts a stress on the crankpin bolts.

The bolts must be strong enough to withstand this inertia force and so must be given an initial tension greater than this inertia force. In fact, the initial tension should be at least 25 per cent above the inertia-force value, to take care of the compressibility factor already mentioned.

To show how it proceeds, let it be assumed that the 10 by 12-in. engine, discussed on previous pages, is to have its crankpin-bearing bolts tightened. Previously it had been assumed that the effective reciprocating parts weighed 150 lb. When the crank is at top dead center, the constant $1/R$ in the formula has a value of 1.222 for a crank ratio of 4.5. Inserting the several values in the formula, it will be found that the inertia force at top dead center on the exhaust stroke is approximately 35,000 lb. To resist the force, it will be assumed that the crankpin bearing has two $1\frac{1}{4}$-in.-diameter bolts. The cross-sectional area of each bolt is $0.7854 \times 1.25^2 = 1.23 +$ sq. in. The unit stress on each bolt, in pounds per square inch, is

$$35{,}000 \div (2 \times 1.23) = 14{,}240 \text{ lb. per square inch.}$$ [1]

This is quite safe for an alloy-steel bolt that has an ultimate strength of, say, 100,000 lb.

Each bolt, it is here assumed, has a length of 12 in. An initial tension should be given each bolt equal to the inertia force load plus 25 per cent, or 17,800 lb. per square inch.

The modulus of elasticity of the alloy steel is 30,000,000; that is, to cause the bolt to stretch to twice its original length, a load of 30,000,000 lb. per square inch would need to be applied. Practically, a steel sample would break long before it was doubled in length, but it is possible to measure the elongations with a series of loadings and draw a curve which, when extended, will give the modulus value.

A load of 20,000 lb. (this value is here used in place of the more exact value of 17,800 lb. unit loading) would stretch a bolt $20{,}000 \div 30{,}000{,}000 = 1/1{,}500$ of its length. Since the bolt is 12 in. long, an initial tension of 20,000 lb. would stretch it $\frac{1}{1500}$ of 12 in., or 0.008 in.

It follows then that, if the engineer tightens up the crankpin-bearing bolts of the 10 by 12-in. engine, here used as an example, until each bolt is enlongated 0.008 in., the initial tension is 20,000 lb., and the bolts should not stretch in operation.

Measuring Bolt Tension.—The C gage, shown in Fig. 162, enables the bolt elongation to be measured during the tightening process.

Removing Rod.—If the rod is to be removed, in most designs the piston and rod must be pulled through the cylinder after removing the cylinder head. If only the crankpin bearing is

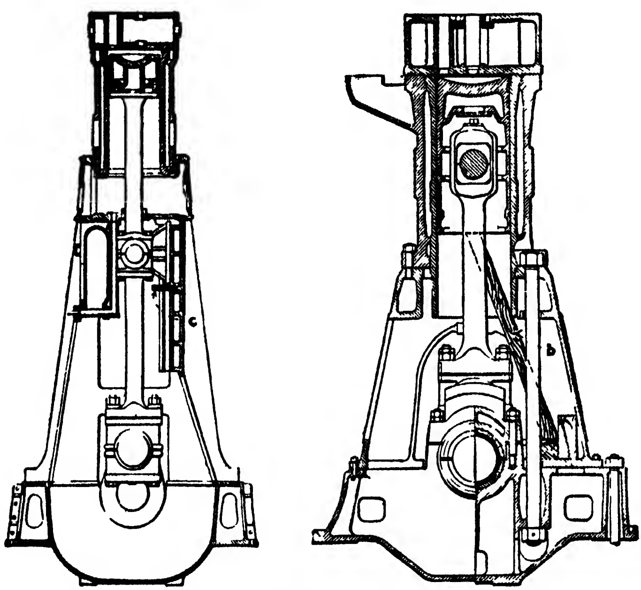


FIG. 170 Methods of holding piston

to be removed for inspection or measurement of clearance, the piston need not be taken out. In trunk-piston engines the piston can be wedged in place by a timber *b* resting on a cross timber and supporting the bottom edge of the piston, as illustrated in Fig. 170. The big-end bolts are now removed, and the bearing halves lifted out. To measure the clearance, leads are placed on the bottom half of the bearing, and the two halves are drawn together by bolts, using a collar or larger nut in place of the bottom end of the rod.

On large crosshead-type engines the bearings are too heavy to handle without block and falls. The crosshead may be held up by tapping a hole in one guide and inserting a pin as shown at *c* in Fig. 170. By using two small blocks and falls, with eyebolts screwed into tapped holes in the bolt ends and small enough to pass through the nuts, the lower bearing half may be owered with ease, as illustrated in Fig. 171.

In many engines the valve cages may be removed, eyebolts screwed into the piston head and the latter held by small blocks and falls.

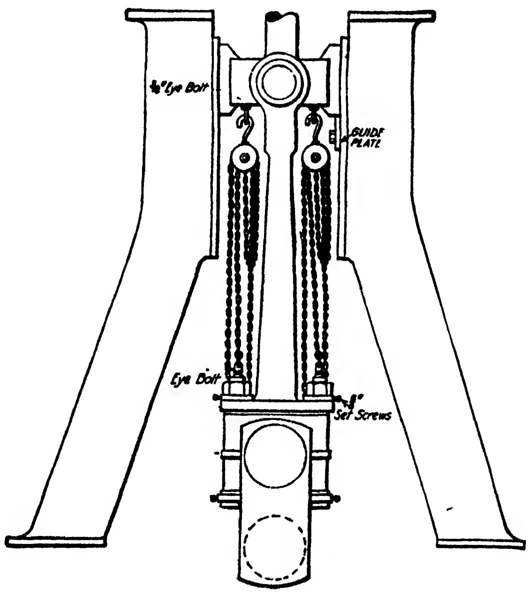


FIG. 171.—Lowering bearing with small blocks and falls.

Before removing the nuts a series of graduations should be marked off and numbered, and an arrow placed on the bolt end.

Bearings of Crosshead-type Engines.—The construction of the crosshead in most large Diesel engines follows conventional marine steam-engine practice, the two crosshead pins projecting from the forged crosshead block, and the upper end of the connecting rod being forked to support two sets of bearing boxes. In most existing engines the space around the crosshead is restricted to such an extent that it is not possible to use chain falls or other lifting gear when adjusting; but this does not act to handicap the work, for in most cases the crosshead-pin boxes are small enough to be readily handled by hand.

The crosshead being worked on is jacked down to its lowest position, and leads inserted to measure the pin clearance. The leads, instead of being laid in the lower box, as in the case with the crankpin bearing, are laid across the top of the crosshead pin, the reason for this being that the weight of all the moving parts above the connecting rod is supported by the lower boxes; when the engine is stopped, the clearance is all on top of the pin. After the leads are taken and before any adjustments are made, the babbitt lining in the lower box should be examined for signs of uneven wear, wiping, cutting, etc. This can be done by hanging the crosshead slipper on the stop bar through the guides and jacking the crankshaft around until the connecting rod moves away from the crosshead pins far enough to expose the bearing surface of the lower boxes.

If the engine has been in service long enough to warrant expectation of appreciable wear of crosshead pins, the pins should be examined for shoulders. Unlike the crankpins and main bearings, the crosshead pins do not have a rotative motion within the bearings. As the lower end of the connecting rod rotates in a circle the radius of which is the crank radius, the upper end oscillates so that the boxes turn on the crosshead pins in one direction during half a revolution of the crank and in the opposite direction during the remainder of the revolution. This rubbing of the boxes back and forth on the pins will sometimes cause a shoulder to wear on each side of the pins, and it is necessary to file this shoulder down before the bearing can be properly adjusted. A very good way to prevent the occurrence of these shoulders is to cut away the pin so that there is a flat spot on each side. This allows the bearing surface of the boxes to override the edge of the cut each time, and no shoulders can form. This does not reduce the effective bearing surface of the pins, because this effective surface is only on the top and bottom portions of the pin.

Bolt Material.—Various materials have been used, such as wrought iron and mild- and high-carbon steel, the last being quite generally used at present. The following specifications are those of a prominent Diesel builder; and if the operator decides to have bolts made rather than to purchase from the manufacturer, these requirements should be followed.

SPECIFICATION

Carbon: 0.30 to 0.40 per cent.
Silicon: 0.30 maximum per cent.
Manganese: 0.50 to 0.80 per cent.
Nickel: 2.75 to 3.25 per cent.
Sulphur and phosphorus percentage as low as possible.

The material should be oil hardened and tempered to give the following values:

Tensile strength: not less than 50 tons per square inch.
Yield point: 65 per cent.
Elongation: 20 per cent.
Reduction of area: 45 per cent.
Impact value: 40 ft.-lb.

The ordinary life of a bolt varies from 2,000 to 30,000 hr. of service. Probably 20,000 hr. represents the period of service advisable before replacement. Each time a bearing is removed, the bolts should be inspected. This consists of polishing the bolt and then coating it with oil. The oil is then rubbed off, and a coat of white lead is rubbed on. If there are any minute incipient cracks, they will show up the oil oozing out, making a mark on the white lead. In addition it is a good plan to go over the bolt with a strong magnifying glass.

CHAPTER XIII

CYLINDERS AND CYLINDER HEADS

General.—The first Diesel engines built in America had the outer jacket wall and the inner bore, or liner, cast in a single unit. At the present time, a few designs embody the unit casting; but as a rule, most manufacturers employ a removable liner.

The advantage of the combined jacket and liner is that it simplifies machining and reduces manufacturing costs. As long as a cylinder is not heavily loaded, that is, called upon to burn a too large amount of fuel per cubic foot of cylinder volume, the combined design is satisfactory. Cylinder temperatures are not high at moderate rates of combustion; consequently, there is less expansion stress to cause fracture of the liner, and there is less danger of absence of lubricating oil on the liner and piston.

Wear of the liner is not excessive under moderate combustion rates. It is not surprising, then, that the one-piece cylinder design is found mainly on two-cycle, crankcase-scavenging Diesels where combustion rates are low by reason of the relatively small amount of air in the cylinder.

The separate, removable liner finds preference among engine builders owing to the ease with which a damaged cylinder wall can be replaced by insertion of a new liner.

Manufacturing processes also favor the removable liner. On small engines, it is much easier to cast the en-bloc frame and insert a thin liner than it is to cast the liner in place. Furthermore, the liner should be made of a high-Brinell cast iron, whereas the frame itself might well be made of a softer nickel gray iron which is easy to machine. Also, the amount of wear that a small cylinder bore can bear before the clearance between piston and liner introduces blow-by and loss of power is small. One cannot afford to discard an entire engine frame simply because a single cylinder wall is badly worn.

On large engines, manufacturing costs, replacement, and materials specifications make the removable liner an absolute necessity.

For examples of cylinder design, the reader should turn back to engine cross sections shown in previous chapters.

Fractured Cylinders.—In cylinders having the liner cast in one piece with the outer jacket there is a tendency for the liner to elongate more than the outer jacket wall, since the temperature is much higher. If the cylinder is not designed with some means of allowing jacket elongation, a crack will eventually show up. In many cases the fracture will appear at the base of the jacket, where the metal walls converge. Unless the cylinder is well designed, there is danger.

This difficulty does not arise in cylinders with liners. Here, if fracture does occur, due to other than cooling-water troubles, it will be found that the damage may be traced to a thick top flange. This will prevent the water from keeping the metal at a low temperature. In other cases, especially on large engines, fractures have been traced to excessively thick liner walls. This made the temperature gradient between the inner and outer walls too great, with the result that the expansion of the liner created excessive tensile stresses in the outer skin, thus starting the break.

In high-speed, high-powered Diesels of the marine type, the amount of heat generated per square foot of combustion-space surface is large, as high as 600,000 B.t.u. per square foot per hour. A good deal of this is radiant heat, which is absorbed by the iron walls. The rate of heat absorption is so high that enormous compressive stresses are set up in a thin skin of metal on the inner side of the liner walls. The skin tends to separate from the rest of the metal which is at a much lower temperature. Fractures thus created will soon go entirely through the liner.

These difficulties are present only in engines of large cylinder diameters or of high capacity, that is, with a high mean effective pressure. With the modern Diesel of medium speed and moderate rating these troubles seldom arise.

The difficulty of fractured cylinders has been largely eliminated as a better understanding of the necessity for proper cooling has come to the operating force. It can be safely stated that most cylinder fractures are traceable to improper cooling. Many plants follow a custom of cutting off the flow of cooling water as soon as the engine is shut down. Since there is about as much heat absorbed by the water as is given up in useful work, on shutting down, a large quantity of heat remains in the iron parts; this must be taken up by the water contained in the cylinder jacket. This produces a rise in temperature sufficient to cause precipitation of the salts suspended in the water. These salts are deposited in the form of scale on the jacket walls. The action continues until the scale becomes so thick as to preclude the possibility of proper cooling. The cylinder walls attain a high temperature and develop fractures because of the inability of the red-hot walls to withstand the high cylinder pressure. Due attention to the cooling water will prevent any fracture in the cylinder liner.

Scored Cylinders.—In the chapter on pistons, several defects that would cause piston scoring are pointed out, and the discussion applies to cylinders. Where the scoring is purely local in character, the surface can be placed in working condition by rubbing with an emery stone, finishing up with a patient application of a scraper. Ordinarily, since the scoring is due to piston or liner distortion, the defective surfaces are not in the plane of the crank and piston; consequently, the reduction of the scored surface below the cylinder-wall circle is not of any moment, although if too much, the piston ring will not make contact, and gas blow-by will occur. Another type of scoring is at times encountered. This has the character of grooves and ridges, due in the main to the side thrust of the piston under the high cylinder pressure. As long as the depth of the grooves is 0.005 in. or less; no serious damage will occur, but when secondary ridges appear between the original ridges, the liner must be rebored.

Cylinder Wear.—It has been stated that the amount of liner wear in an accurately fitted oil engine operated under favorable conditions should be at the rate of 0.001 in. per 1,000 hr. of service. As a basis for comparison this figure is useless. It fails to specify the particular part of the liner subject to wear and may be too low an estimate for the wear at the critical point, that is, just after firing takes place. Results on a large number of plants do not justify this generalization for any part of a liner surface. The wear varies enormously. A set of liners in the same engine, of the same age, exhibits different depths and rates of wear. For instance, the average rate of wear per year in thousandths of an inch in the case of four 12-in. liners of the same age in the same engine was found to be, measured in thousandths of an inch, as follows:

Parallel to shaft near top	2 37	3 5	3 06	2 43
At right angles to shaft near top	2 25	3 43	2 93	3 06
Showing maximum variations up to	29 and 51 per cent			

And again:

Parallel to shaft at bottom of stroke	1.06	2.43	0 69	0 75
At right angles to shaft at bottom of stroke, showing even greater variations	1.25	1.43	0 88	0 75

No definite inference as to rate of liner wear for the several parts of the wearing surface can be drawn. The wear should be gaged at intervals of, say, six months. When the compression becomes so low as to cause ignition failure or high fuel consumption, the liner should be rebored or renewed.

Cylinder liner wear should be checked periodically, as it may become excessive because of a correctable condition. Most Diesels are given a general overhaul annually, and then the pistons are removed. Micrometer measurements of the cylinder bore should be taken both crosswise and in parallel with the shaft. The measurements should be made at points 2 to 4 in. apart vertically, depending on the size of the engine, along the line of ring travel. In rare instances the rate is as low as 0.001 in. per 5,000 hr. of running. The chart in Fig. 172 gives an indication of where liner wear occurs.

The amount of permissible wear before liner renewal is necessary depends upon the character of the wear, the load conditions, and the cylinder diameter. It is obvious that, if liner wear is at the rate of 0.001 per 1,000 hr. of operation, a 4½-in. bore liner will need replacement long before a 20-in. liner would require it. For example, the clearance between liner and piston of a 4½-in. cylinder might be $4.5 \times 0.0007 = 0.00315$. A wear of 0.010 would raise this clearance to 0.013 in., or over four times the original value. In case of a 20-in. cylinder the original clearance might have been $20 \times 0.001 = 0.02$ in. A wear of 0.010 would raise the clearance to 0.03, or only 50 per cent over the original clearance. The 4.5-in. bore would permit blow-by, due to ring action, long before the wear reached 0.01 in.; consequently, it is to be expected that under the same service conditions the high-speed engine will need liner renewals more frequently than its slow-speed competitor. Offsetting this, however, is the relative cheapness of the small-bore liner.

Wear of liners is probably proportional to the number of times the engine is started.

Reboring Cylinders.—The natural course of wear in the liner increases the clearances to such an extent as to require reboring.

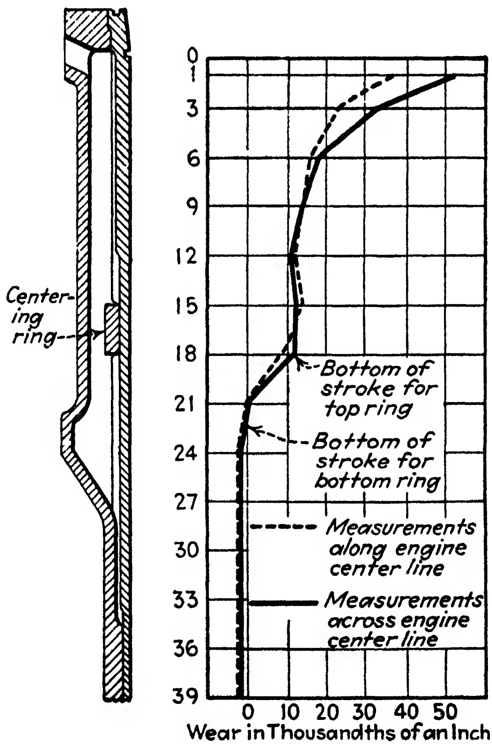


Fig. 172.—Typical cylinder wear.

The liner, as an average, will need replacement or reboring every 3 to 7 years, dependent on the hours of service and the attention that it has received.

When the engineer is confronted with the problem of cylinder reboring, it is well for him to shift the work to the shoulders of some machine shop that makes a specialty of such work. The actual reboring is not hard; neither is the setting up of the boring machine; however, it requires a boring bar which will cost around

$800, and few shops are willing to place their machine on a rental basis.

It is well, save in unusual cases, to secure a new liner from the builder as well as a new piston, if this is necessary. In many cases the piston may be turned down, and a liner to fit purchased.

Liner Replacement.—All cylinders are of a thickness that will allow at least one reboring. If the liner becomes worn after it

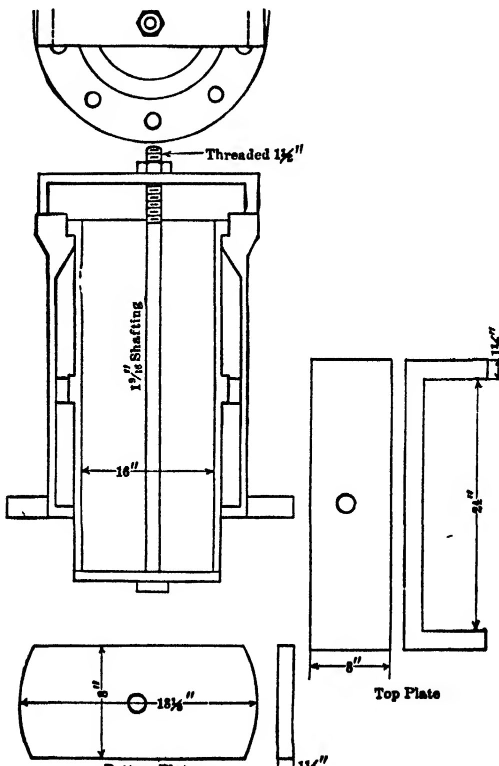


FIG. 173.—Removing cylinder liner with draw bolt.

has had one reboring, or if it is fractured, the withdrawal of the damaged liner is easily effected by the use of the draw bolt, as outlined in Fig. 173. The spider may be made with either two or three fingers; two are as serviceable as three. The bolt is of $1\frac{9}{16}$-in., cold-rolled shafting, the thread having a $1\frac{1}{2}$-in. diameter. The spiders are placed over the cylinder flange and the front end of the liner as indicated. A part turn of the nut will bring the rod under tension; a few sharp blows on the inner surface of the liner at the crank end will, in most cases, loosen it so that the bolt can pull it out with ease. If the liner resists, additional bolt tension, followed with hammer blows along the liner supports, will expedite the removal.

If the engine is a large one, the method as outlined may not suffice, since the liner requires more stress on it than can be given by the bolt. In such cases a steel I-beam should be slipped through the crankcase doors; with one of the beam ends supported by blocking, a hydraulic jack is placed under the other end. With blocking placed between the I-beam and the lower end of the cylinder liner, the hydraulic jack will exert enough force on the liner to lift it.

Before starting to remove the cylinder liner, drain the water from the engine, and remove sediment from the bottom of the water-jacket space surrounding the liner. Drain lubricating oil from the engine base. Remove the cylinder-lubricator connecting nipples. Remove the cylinder head and piston assembly. Carefully mark the top of the liner and housing, so that the liner later may be reinstalled in the same position. Arrange pan to catch sediment or water that may enter the crankcase as the liner is raised. Remove the liner.

Clean the water-jacket space; remove scale and other deposits. File smooth and round off the entering edge of chamber E in the lower bore D (Fig. 174). Clean the liner, removing all scale or other deposits and burrs. If required, recondition the top of the liner bore by grinding or scraping and stoning any ridges caused by piston-ring wear. Do not extend grinding into the region below the upper travel of the top piston ring.

Replacing Cylinder Liner.—The contact surface of liner flange A (Fig. 174) is then cleaned, and a very thin coat of Prussian blue applied to this surface. No Prussian blue should be applied to the contact surface in the liner housing. Lower the liner into position without rubber rings B and copper gasket C. With the liner flange resting on seat A, and the marking on top of the liner approximately in line with that on the housing, turn the liner back and forth through a small angle. Raise the liner, and examine the areas indicated by bluing markings on the liner seat in the housing. If high spots are indicated, scrape seat A of the liner housing to remove the high spots. Only a slight amount of metal should be removed between each inspection and scraping as above. Precaution should be taken to prevent any of the scrapings from entering the crankcase. Repeat scraping and inspection as above until the liner-flange contact surface bears evenly all over and only under conditions where the liner is turned through the small angle to either side of the position to which it is

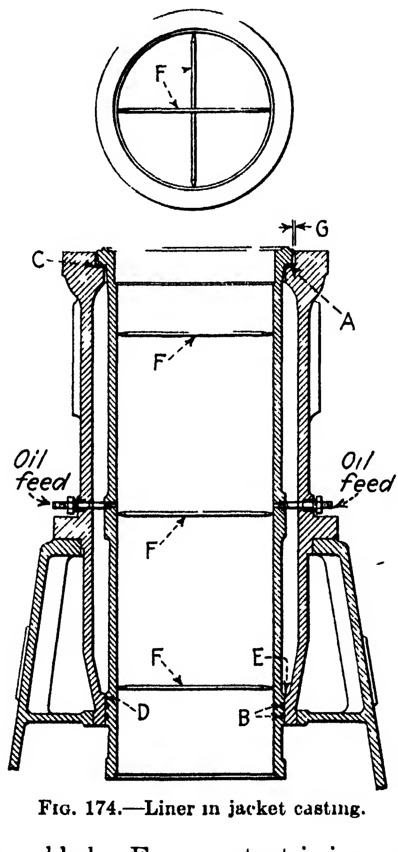


FIG. 174.—Liner in jacket casting.

to be finally assembled. Even contact is important to prevent liner distortion when finally assembled.

Clean the housing seats thoroughly; install a $\frac{1}{32}$-in. thick copper gasket *C* (Fig. 174). The gasket should be new, soft annealed, and without burrs or other surface defects. Do not apply any joint compound or paint to the gasket-seating surface. Apply castor oil liberally to the rubber rings *B*, and fit into position on the liner. Use only new soft-rubber rings. Cover the inside bore *D* of the liner housing with castor oil. Do not use joint compound or lubricating oil on the rubber rings or bore *D*.

Lower the liner into the cylinder housing, using a fixture (Fig. 174) and taking care to locate the liner so that its markings and housing will be in line. Observations should be made through the handhole opening in the lower section of the liner housing, to be sure that the rubber rings do not shear when entering the lower bore *D*. Use the cylinder head to force the liner into its final position in the housing. If it does not enter when applying moderate hand force to the wrench on the cylinder-head nuts, remove it, and make corrections as required. With the liner in final position, determine bore *F* at top center and bottom, in planes parallel to and at right angle to the center line of the crankshaft, as indicated in Fig. 174. Liner dimensions should indicate bore roundness within 0.0015 to 0.002 in. as applying to new liners. In cases where old liners are used, these tolerances may have to be modified to allow for condition of wear. Inspect the clearance *G* between the liner flange and the counterbore in the housing. This clearance should be uniform all around. Install the new cylinder lubricating-oil connecting nipples cut to proper length, and thread them. Connect the piping.

When installing a worn liner, the condition of the liner bore may warrant installation of the liner at an angle of 90 deg. from its original position. In this case, the tapped holes for cylinder lubricator nipples are sealed with suitable cast-iron plugs which are faced off flush with the liner. In cases where new liners are installed or old ones are installed in the turned position, new cylinder lubricator nipple holes must be drilled and tapped after the liner has been assembled in its final position. Install the cylinder head, and tighten down. Refill water jackets, and inspect upper and lower liner joints for leakage. After this remove the cylinder head, and install the piston and cylinder-head parts.

If the liner is cast in one piece with the jacket, of course a new liner cannot be installed. It is customary with such engines to bore the liner larger in diameter and to install a new piston of the correct diameter to fit the enlarged liner bore. The cost is about the same as it would be if a new liner were installed in an engine. Furthermore, in many cases, wear of the piston occurs simultaneously with wear or scoring of the cylinder surfaces; consequently, a new piston would be required in any event.

True, the thickness of the metal wall must be sufficient to permit reboring; preferably at least two reborings should be provided for. If reboring occurs, as many estimate, after 7 years of operation, the two reborings would give a life of 15 years. But it is not impossible that heavy and constant engine loading, especially with dirty fuel oil, will occasion so much wear that reboring may be required in a much shorter time.

Certain engine builders offer to supply a rebored cylinder and a new piston in return for the old cylinder plus a moderate amount of cash. The old cylinder is then rebored and shipped to fill the order of another engine owner. In many respects this is advantageous. Surely the old cylinder should be seasoned and free from machining distortion and so better than a new cylinder.

Dry Liners.—In the event that reboring makes the metal wall too thin, it is possible to rebore the worn surface and install a "dry" liner, that is, a liner that slips into the smooth bore created by reboring and upon which the piston travels. Some difficulty has been experienced with "dry" liners because of the imperfect contact between the liner and the rebored cylinder wall. The reboring should be followed up by reaming or grinding, and the liner's outer surface should be ground, to insure intimate contact between liner and cylinder. Otherwise, the resistance to heat-flow to the cooling water may be so great that the piston overheats.

The best method of inserting the dry liner is to have it about 0.005 in. larger in diameter than the cylinder bore and then chill it with dry ice, so that the contracted diameter will permit the liner to slip in freely.

Metal Spraying of Liners.—At present, many engineers are repairing worn liners by spraying molten metal on the surface.

Metal spraying is done with a portable gun weighing 2½ to 5 lb., which may be placed in the toolholder of a lathe or held in the hand and worked like a paint-spray gun. Several sizes and types of guns are available from three American firms, but in principle they are similar, and all have a spray nozzle, which may be considered as being made up of three concentric tubes.

The metal to be sprayed, which may be steel, monel metal, bronze, lead, zinc, aluminum, or, in fact, almost any metal that melts below 5000°F., is fed through the center opening in the form of wire. Acetylene and oxygen are supplied through the second annular passageway and melt the wire in the inner part of the conical flame. Compressed air at 40 to 75 lb. pressure in the outer annular space atomizes the molten metal and sprays it at high velocity against the surface to be built up. A compressed-air turbine drives the mechanism in the gun which draws the wire from a coil and feeds it into the oxyacetylene flame.

Standard guns when held 5 in. from the surface spray a circle about 2 in. in diameter and deposit a coating 0.001 to 0.006 in. thick per pass. Wire feeds vary from 12 to 37 ft. per minute, depending upon the metal sprayed and the type of work. The spray gun should preferably be held at 90 deg. to the surface, though satisfactory bond and density may be obtained with the spray making an angle as little as 45 deg.

The most important part of metal spraying is preparation of the surface, which must be clean, dry, free from all oil, and roughened so that the metal spray will key fast to it. This condition is best secured by blasting with steel grit, a good grade of silica sand or aluminum oxide, under 40- to 80-lb. air pressure. After the surface is blasted it should be sprayed as soon thereafter as possible, at least within 24 hr., to avoid oxidation.

Cylinder and Head Joints.—Although a few engines depend on a metal-to-metal joint at the head to withstand the cylinder pressures, some form of gasketing is now well-nigh universal. The gasket may be a flat copper ring, a copper wire, or a round rubber ring.

The flat copper ring is very successful as a gas check and is not difficult to make. Its objectionable feature is the large amount of sheet copper that is wasted in cutting the ring. Experience proves that the thinner copper sheet makes the best gaskets; $\frac{1}{32}$ in. thickness of metal is ample and enables the gasket to conform to the flange face. The gasket is best cut scant so that it fits easily into the gasket recess. If it is so wide as to require driving, the edges will bend, and the gasket will not prove gas-tight. If a gasket cutter is not at hand, a pair of tinner's shears will be very satisfactory. A wooden mallet is handy to hammer the gasket to a flat surface.

In case sheet copper cannot be procured, an equally serviceable ring can be made of No. 10 gage soft-copper wire; when the bare copper is not available, waterproofed electrical wire of No. 10 gage may have its insulation burned off, and the bare wire used. The wire is formed into a circle of the proper diameter, and the ends are soldered together. If the cylinder flange is not provided with a recess to receive the ring, the latter should be placed inside the bolt circle, touching each stud. This allows the leverage to be a minimum. The wire must be free from kinks or bends.

Round rubber gaskets are often used on vertical engines. The rubber tubing is shaped into a ring of the proper diameter,

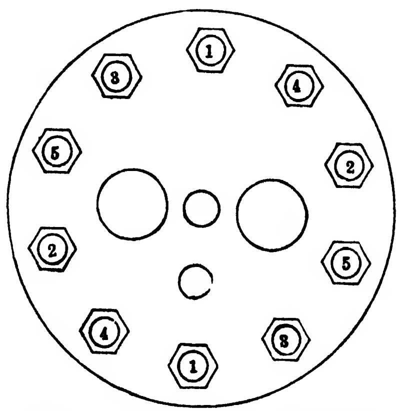


FIG. 175.—Method of drawing down cylinder-head bolt nuts.

and the ends are united by rubber glue. In engines where the cooling compartment of the head communicates with the cylinder jacket by cored openings at the flange, the openings are surrounded with like tubes. These gaskets of rubber tubing can always be obtained from the engine builder, but any mill supply house will furnish the tubing in coils at far less cost.

Drawing Up Cylinder-stud Nuts.—In tightening up the cylinder-head nuts, many engineers draw up one nut as snugly as possible before drawing up any of the others. Such handiwork is evidence of a lack of mechanical knowledge and is to be shunned. If the tops of the studs are numbered in pairs, similar to Fig. 175, and tightened in rotation, the head can be drawn down quite evenly. For example, all the nuts are run down against the head; then the No. 1 nuts are given an eighth turn, followed by a similar performance on No. 2 nuts; etc. Returning to the No. 1 nuts, they are given another eighth turn, etc. When giving the nuts the final movement, a workman may strike the wrench handle several sharp blows with a sledge.

It is highly desirable to know the amount of initial tension given the bolts. On each bolt this should be the total gas load in the cylinder divided by the number of bolts. This value should be increased by 25 per cent. The method of calculating the initial tension by the elongation was discussed in the chapter on connecting rods.

Fractured Heads.—Fracture of a cylinder head is a malady that occasionally appears to afflict all makes of Diesels, no matter what the design may be. There is no doubt that the more complicated heads fracture often, but even the simplest of head castings do give way, usually on heavier loads. Bad water, without question, occasions the greater number of the fractured cylinder heads. On shutting down, the water gives up its salts, which deposit on the iron surfaces. The greater part settles on the lower surfaces. This coating accumulates until it is as much as an inch thick. Since scale is an excellent non-conductor, little cooling effect is experienced on the hot cast-iron head wall which is in contact with the intense flame of the burning fuel. On cooling, the contraction of the iron gradually weakens the bond of the scale. This scale ultimately loosens while the engine is under load, exposing a red-hot iron surface to the cooling water. The sudden localized contraction of the iron on being chilled results in a fractured head. Evidently the horizontal head is more likely to shed the scale than the vertical head. It becomes necessary for an engineer to inspect the cylinder head at stated intervals; if scale is present, it can be removed by scraping. If a solution of muriatic acid and water in the proportion of 1:10 is allowed to remain in the jacket a few hours, all the scale can be washed out with a hose.

Heat stresses due to faulty design of the head also contribute to these fractures.

The combustion space in a Diesel engine is bounded by the cylinder cover, the piston, and the upper portion of the cylinder liner or, in the case of an opposed piston engine, by two pistons and a portion of the cylinder liner. The temperature of combustion is about 2500 to 3000°F. and falls during the working stroke to about 1200°F., these figures varying, of course, according to the relative weight of the fuel and air charge. The heat flow through the material of the cylinder liners, piston, and cylinder head gives rise to stresses in the material which become more serious as the size of the cylinder increases. The cylinder cover has always proved the greatest difficulty, as the casting is usually the most complicated and at the same time is subjected to the maximum temperature and pressure conditions. The cylinder cover and piston are together responsible for over 80 per cent of total heat flow to the cooling medium, the remainder of the heat being carried away through the cylinder liner. Although the water- or oil-cooled piston of a large engine carries away at least as much heat as the cover, the casting is much smaller than the cover casting, and freedom for expansion is more easily obtained. The liner of the normal type of engine is also a less difficult problem than the cover, as it is a perfectly symmetrical casting at the region of the combustion chamber, where the maximum temperatures are encountered.

Cylinder heads at times crack through the bridge between the exhaust and fuel-valve cavities. These fractures are often due to improper cooling rather than to faulty design, although there are instances where casting fractures develop at these places as soon as the head is machined. To safeguard the engine head from such fractures that may develop in service, the flow of cooling water to the head must be positive, and the temperature kept at a constant reading, the value of which can be determined only by experiment on the particular engine.

On marine Diesels, the fracturing of cylinder heads is directly traceable to excessive overloads. In a heavy sea the propellor is at times exposed and, immediately thereafter, completely buried. This leads to engine hunting, and neither governor nor manual control can cope with the situation. The fuel pumps deliver excessive charges to the cylinder, creating pressures beyond the capacity of the heads to withstand.

Furthermore the marine engine, to reduce the weight per horsepower, is speeded far higher than the stationary Diesel. This applies especially to submarine and light cruising engines. The heat absorbed per square inch of surface by the heads of a 500-hp. engine at 400 r.p.m. is double the amount absorbed by the heads at 200 r.p.m., since the cylinder bore area is practically one-half that of the latter engine, and the total amount of heat absorbed is approximately the same. The cooling system, then, must be absolutely correct in design if fractures are to be avoided. The tendency of the salt water to scale is, of course, more pronounced where the temperatures are as high as exist in the marine Diesel heads.

Comparison of Two- and Four-stroke-cycle Cylinder Heads.—It has often been claimed that, since twice as many combustion events occur per minute in a two- as in a four-stroke-cycle cylinder, the heat stresses in the two-stroke-cycle head are much higher. This is partly true; but as an offset the two-stroke-cycle

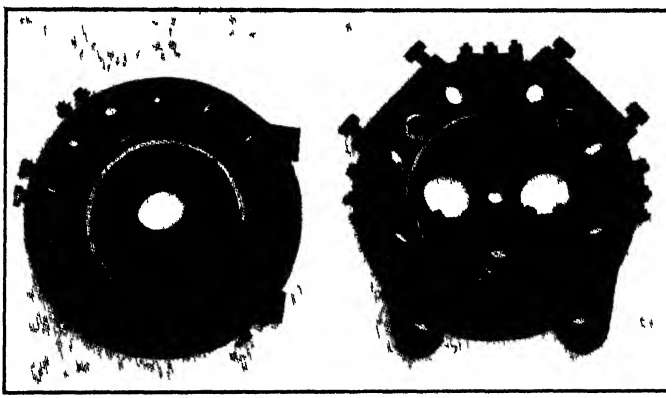


FIG 176—Two- and four-stroke-cycle cylinder heads

head is a more symmetrical casting and has a uniform cyclic temperature.

In the four-stroke-cycle engine the head has an exhaust-valve cavity that absorbs much heat from the exhaust gases and an admission-valve cavity kept cool by the flow of air. This uneven temperature condition is the cause of head fracture. This fracture appears between the exhaust- and fuel-valve cavities, as in Fig. 176.

In the two-stroke-cycle engine with scavenging valves in the head there are no exhaust valves, and the temperature is fairly uniform. Little danger of fracture from uneven temperatures exists. This is especially true in the case of the port scavenging engine where the head has only the one cavity for the fuel valve shown to the left in Fig. 176.

There are at least 5,000 two-stroke-cycle Diesel cylinders in service with the single-fuel-valve opening, in all parts of the world, and of these over 2,000 cylinders are above 20 in. in diameter. The mean indicated pressure is generally about 100 lb. per square inch, and the power per cylinder in several cases is over 750 i.hp. If a defect were inherent in the two-stroke-cycle design, as, for instance, if heat stresses had not been taken properly into account, there would surely be a considerable percentage of failures due to cracked heads. The fact that this has not been the case is the clearest proof that the design of the symmetrical head is sound in both theory and practice.

Another point of advantage associated with this design of two-stroke-cycle cover is that owing to the slightly concave shape of the under side, taken in conjunction with a similarly concave-shaped piston head, a well-shaped combustion space is obtained, contributing to efficient burning of the fuel.

Repair of Fractured Cylinder Heads.—Oxyacetylene welding has not been a marked success in the repairing of cracked cylinder heads. The cylinder head contains a great weight of iron, and, in welding, the flame is localized. The consequence is that the metal immediately about the fracture is highly heated and expands. After the molten metal is added, closing the fracture, the head is allowed to cool. The mass of the head has not been heated and so shows no contraction. The obvious result is the shrinkage of the metal at the edges of the fracture, reestablishing the fault.

To weld a cylinder head successfully, a furnace can be constructed of firebrick. The floor should also be made of firebrick supported by old grate bars or iron rods. After a coke fire has been burning about the head for 24 hr., the entire casting becomes red hot. The oxyacetylene flame is then applied; the fracture is enlarged to a trough shape, thus allowing the added metal to reach the bottom of the fracture. The new metal is deposited in small quantities and thoroughly welded to the cast iron before more is added. The head should then be left in the furnace to cool for 48 hr. Since the entire furnace has been at a high temperature, the cooling will be very gradual, thereby avoiding all shrinkage strains. If the fracture has been across a valve seat, the part must be machined, and the valve refitted. The process of welding here outlined has been followed with complete success, saving hundreds of dollars in a plant where five Diesels were installed.

Materials for Cylinders and Heads.—For Diesel engines the cylinders, heads, and liners are generally of nickel-alloy cast iron, although cylinders have been made of chrome-vanadium steel.

In the cylinder heads are the valves set in cast-iron cages. The fuel and air prevent their respective valves from being subjected to the excessive heat experienced by the exhaust valves. The exhaust disk and valve stem get the full heat of the exhaust gas, plus the pounding of the disk on its seat. The stem cannot be satisfactorily water cooled, with the result that a special heat-resisting alloy is required. Disks of nickel chromium or nichrome made up of 61.2 per cent nickel, 24.88 per cent iron, 12.05 per cent chromium, and 1.44 per cent manganese can stand high temperatures. Stems and disks of silcome, containing 95 per cent chromium, 3.5 per cent silicon, and with a medium-carbon content will not scale at a temperature of 1750°F.

Cylinder liners are subjected to particularly hard usage, not only to high temperatures and pressures but also to the wearing action of the piston rings. This calls for a grade of cast iron having a hard surface which will wear well at high temperatures. The iron should be fine-grained and have a high-carbon content. For instance, iron with 2.86 per cent graphitic carbon and 0.77 per cent combined carbon has a Brinell hardness of 230, whereas one with a graphitic of 2.77 per cent and a combined carbon content of 0.72 has a Brinell of 167. Brinell hardness above 600 has been reached, but the usual hardness in large engines is around 350, and even at this figure many shops have experienced difficulty in machining their liners.

CHAPTER XIV

ADMISSION AND EXHAUST VALVES

TYPES; ADJUSTMENTS; REPAIRS

Valves.—The four-stroke-cycle Diesel engines are provided with air admission and exhaust valves, as are all four-stroke-cycle gas engines. In fact, a number of Diesels are built with cylinder heads and valve mechanisms that are but slight deviations from the similar parts of a gas engine. This applies particularly to the high-speed Diesels of which a number possess what might well be termed "gas-engine valve gear." In the large engines the entire valve equipment is distinctly a Diesel feature, differing in many respects from any of the gas-engine designs.

In practically all Diesels of over 50 hp. per cylinder the admission valves seat in valve cages. In some engines the exhaust valve seats directly on the cylinder casting; in others a removable seat, which is fastened to the head casting, is supplied. Strictly speaking, the exhaust rather than the admission valve should be fitted into a cage. The hot exhaust gases, in passing through the valve opening, wear the ground seats rapidly, but the admission valve is not subject to such an erosive action. From the operator's viewpoint, as will be discussed later, neither valve should be caged.

In the small horizontal engines where the valves are placed with the stems in a horizontal position, usually neither valve is caged.

Camshafts.—The few horizontal Diesels have layshafts geared to the crankshaft, running along the frame, or the valves are operated by pushrods, from a camshaft placed close to the crankshaft. The cams that open the various valves are placed on this shaft, as is also the drive to the fuel pump. Essentially, this is the same as that employed on all horizontal gas engines.

Camshafts for vertical Diesels may be placed inside the crankcase (Figs. 177 to 180), as in automotive practice, with long pushrods to the valve rockers, or the shaft may run outside the frame at a level with the cylinder bases, with short pushrods, as in Fig. 181. The third arrangement has the camshaft on top of the cylinders, with the valve rockers bearing on the cams on this shaft. This latter method is most popular in stationary engines

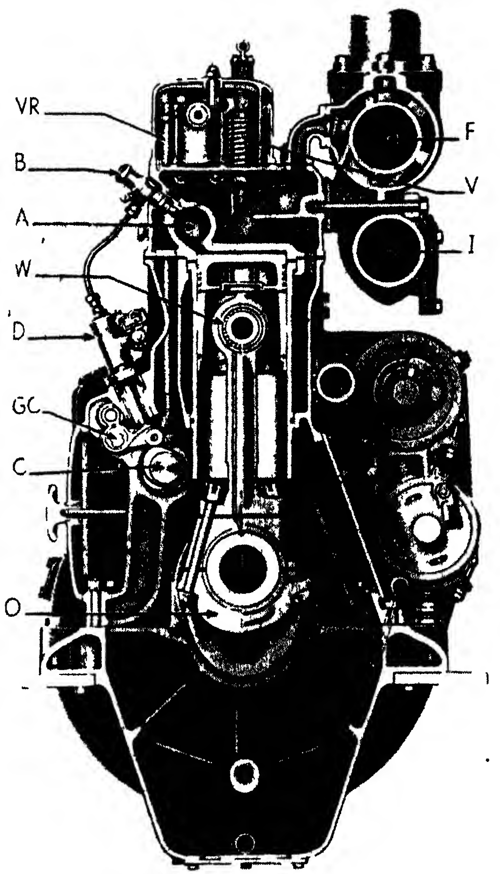


FIG. 177.—Fairbanks-Morse high-speed Diesel. *A* = turbulence chamber; *B* = spray nozzle; *C* = camshaft; *D* = fuel-injection pump; *GC* = governor control; *F* = exhaust header; *VR* = valve pushrod; *I* = air intake; *V* = intake valve; *W* = piston pin; *O* = crankpin bearing.

with cylinder diameters between 16 and 28 in. For reversible marine Diesels the camshaft is almost always at the level of the cylinder base, as this permits a ready transverse shifting of the camshaft.

Drives for Camshafts.—With the camshaft located at the top of the cylinder, the usual drive in America has been by a vertical shaft, upon which the governor is mounted. This shaft is geared both to the crankshaft and to the camshaft; it may run

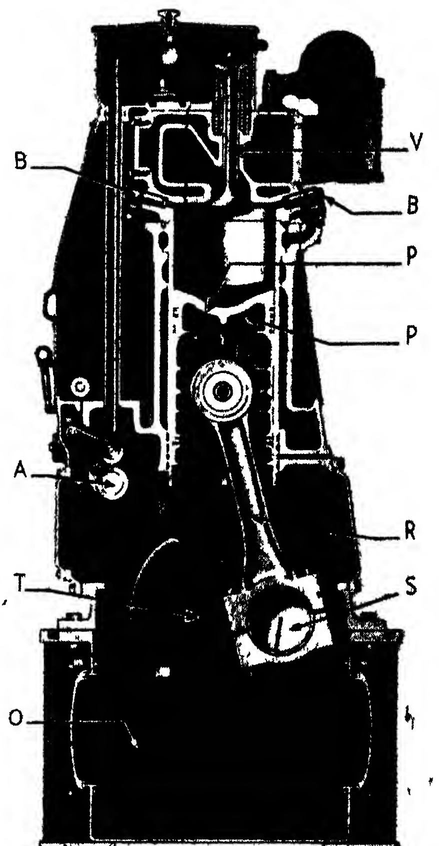


Fig. 178.—Ingersoll-Rand Diesel with camshaft in crankcase.

at engine speed with the 2:1 gear at the top or at half engine speed, with the reducing gear at the bottom.

The vertical shaft is objectionable if the height of the engine makes the shaft too long. Vibrations will set up, which will be reflected in the governor and cam actions. When the shaft is placed on the engine at the end opposite to the flywheel, the

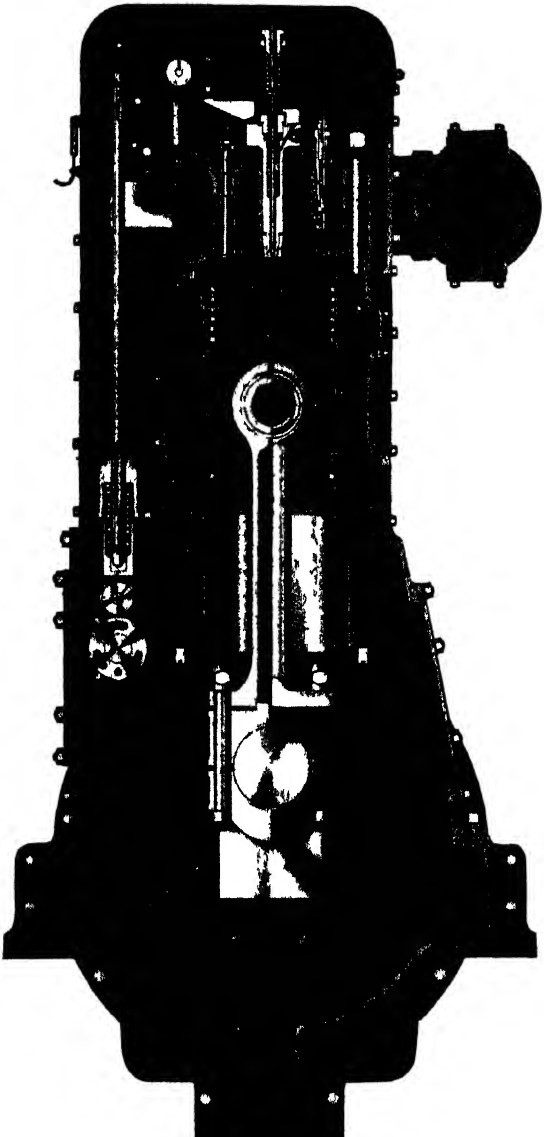


FIG. 179.—Superior common-rail Diesel.

gear and shaft experience the maximum torsional vibration that may exist in the long crankshaft, for this end is the free end, and the shaft whipping is greatest here. Regardless of torsional vibrations that are noticeable, the vertical shaft always is out of phase by the amount of twist in the crankshaft, which at the time of firing may reach 2 deg. By placing the vertical

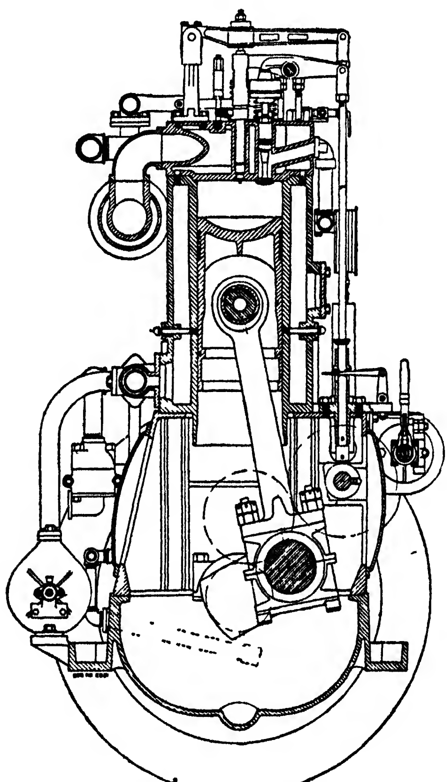


FIG. 180.—Atlas-Imperial Diesel has camshaft in crankcase, and geared to crankshaft.

drive at the flywheel end, the disturbance is practically non-existent.

To offset for elongation of the crankshaft which might cause the drive gears to cramp, a spur gear may be interposed, which, by sliding along a feathered key, accommodates itself to any shaft elongation. An arrangement of this kind is used in the Worthington air-injection engine.

On certain engines the drive to the camshaft is by spur gears, with one floating or idle gear. As long as the distance between the crank- and camshafts is not great, this is a practical arrangement and is easy to keep in adjustment. But when the distance

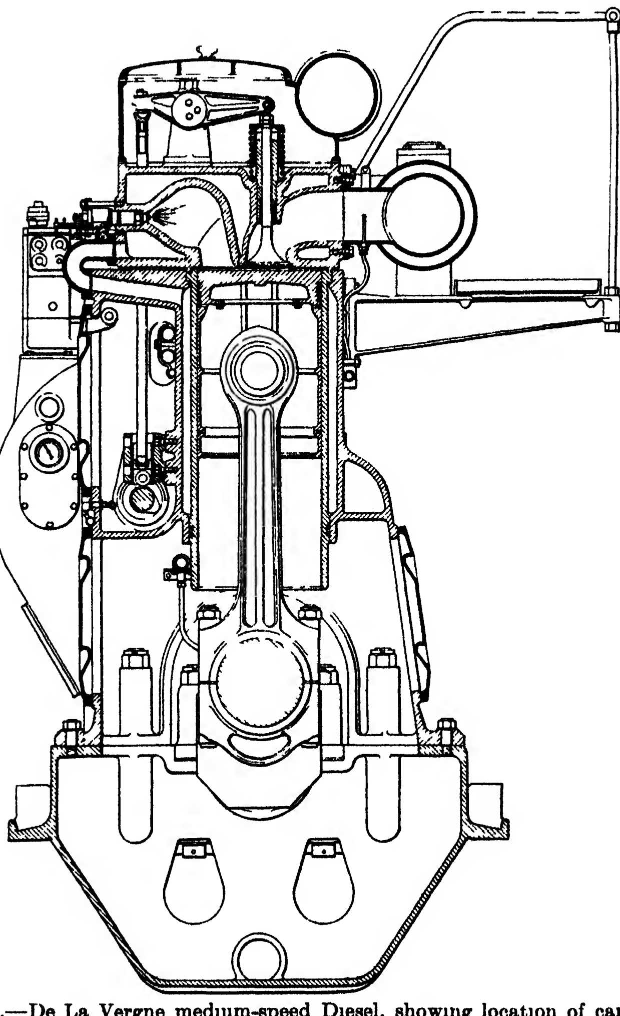


FIG. 181.—De La Vergne medium-speed Diesel, showing location of camshaft.

is such that a train of three to five gears must be used, the author is inclined to favor some other method.

In Europe four cranks and rods have been used to convey the motion from a stub shaft in the crankcase to the overhead camshaft. At slow engine speeds, under 100 r.p.m., this is an excellent arrangement, but at high r.p.m. the long drive rods are subject to vibration, which entails the use of large-diameter hollow rods. These in turn cause rapid wear of the rod bearing.

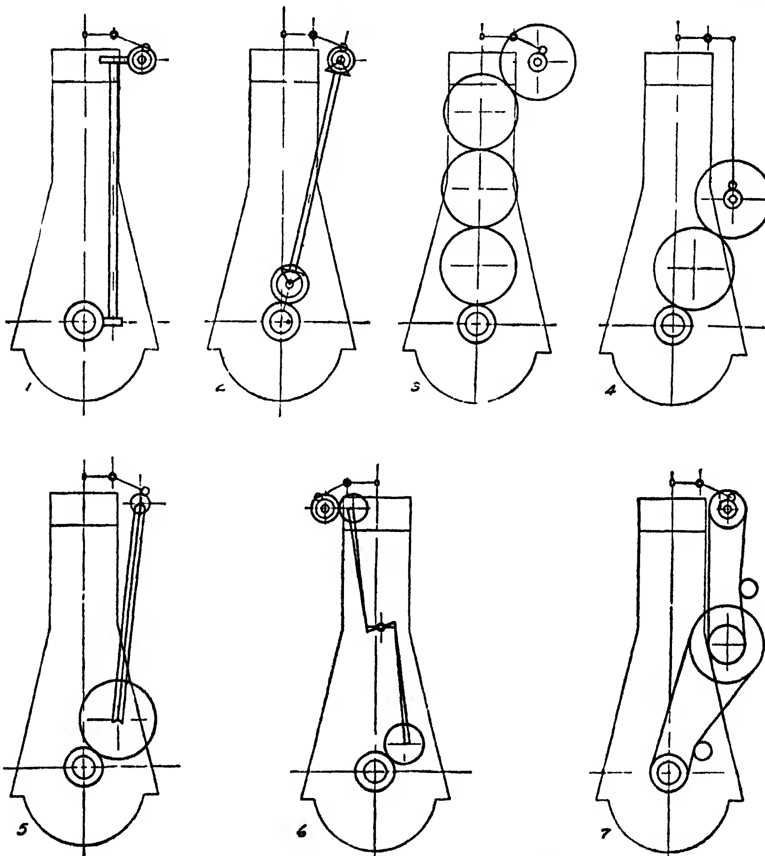


Fig. 182 —Typical camshaft drives.

Many designers have adopted silent-chain or roller-chain drives for the camshaft, this originating with the De La Vergne design. In most of these designs, one chain drives a shaft at the level of the crankcase top, from which the fuel pumps are operated, and a second chain connects this stub shaft with the overhead camshaft. Unless the distance between shaft centers is exceptionally great, this drive, in the author's opinion, is one of the best, if not the best, of all arrangements.

The chain must be exceptionally well made and of ample width, for the shock it experiences when a cam contacts with a rocker arm may be of considerable moment, and there must be provision for slack take-up and for replacement of a link to compensate for wear.

Typical drives are shown in Fig. 182.

Types of Valves.—It is, of course, impossible to show all designs of valves used on the many different engines. In this chapter

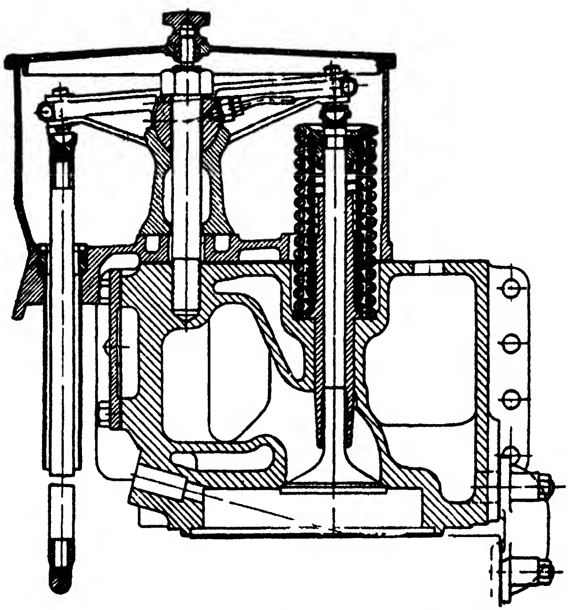


FIG. 183.—Cylinder head and valve gear of Ingersoll-Rand Diesel.

only a few typical designs will be mentioned, but the reader will find that many illustrations elsewhere in this volume show the valve gears employed.

Uncaged Valves.—All small Diesels have the admission and exhaust valves seat directly on the cylinder head. This arrangement has the advantage of being cheaper to produce, as the manufacture of cage is avoided. The cooling of the valve is brought about by the flow of water through the cylinder head and should be efficient. A typical uncooled valve for a cylinder of 12 in. is shown in Fig. 183. It will be seen that, although the valve seats on the cylinder head, there is a removable bushing to guide the valve stem. Furthermore, the valve is provided with two springs. Since two are used, each spring may be light in weight and so offer less inertia resistance when the push rod and rocker start to open the valve.

Caged Valves.—When the diameter of the valve is large, the amount of heat absorbed by the valve is enormous and, if the

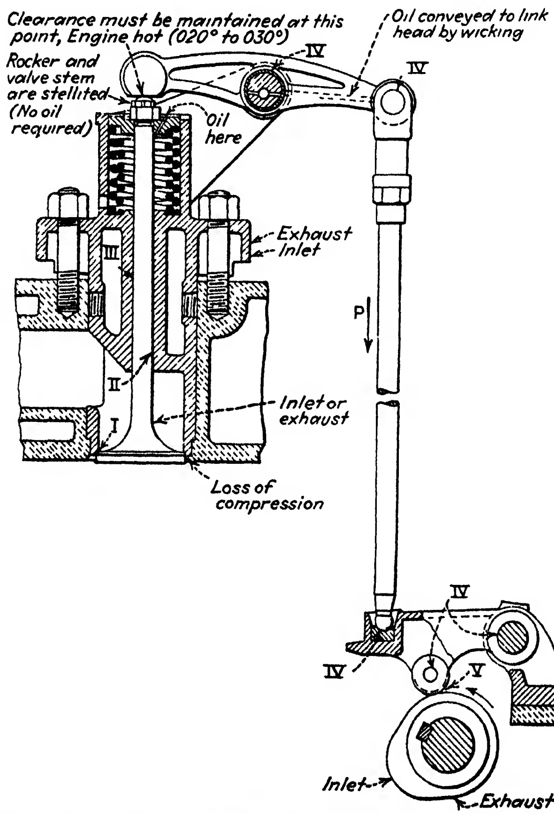


Fig. 184.—Valve rocker and pushrod of Chicago Pneumatic Tool Diesel.

cooling of the valve is not good, the valve disk will burn. This situation led to the use of cages for the valves. Basically one may regard a cage as a bushing in the head and carrying the valve.

For the cage to perform the mission for which it was added to the engine design, it must be cooled. This is accomplished by incorporating a cooling compartment in the cage, with water entering and leaving through pipe connections. In Fig. 184 is shown such a valve gear as applied to the Chicago Pneumatic-Tool Diesel.

Busch-Sulzer Type B Diesel Valves.—In exterior form the admission and exhaust valves and cages are quite similar. Both are placed vertical in the cylinder. The interior arrangements of the cages, however, are by no means identical. The exhaust-valve cage has a removable valve-stem bushing; the cavity between this bushing and the cage body constitutes the cooling-water jacket. The admission-valve cage has the valve-stem support cast integral with the cage proper. Both

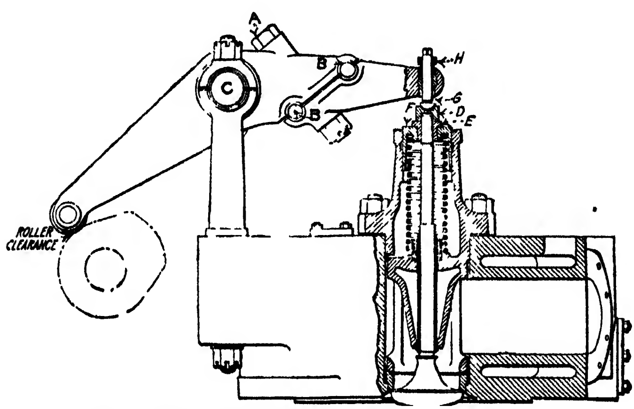


FIG. 185.—Busch-Sulzer Type B exhaust valve and rocker.

cages are provided with removable seats, as appears in Fig. 185. This feature is of importance to the operator, since it obviates the necessity of replacement of the complete cage when the valve seat has been worn excessively.

The valves, both admission and exhaust, have cast-iron bodies with steel stems. In disassembling the valves, the locknuts *D* and *E* are unscrewed, the valve being held by a pin wrench inserted in the two pinholes in the valve body. The removal of the spring cap *F* allows the valve to be withdrawn. The exhaust-valve seat, of course, experiences the greatest wear; consequently, frequent refacing of the cage seat becomes necessary. After a number of refacings with a reamer, the cage seat lies too deep in the cage; when this occurs, a new removable seat ring can be obtained.

Cam Levers.—The valves are actuated by levers which receive their motion from a camshaft lying along the cylinder head. This camshaft is driven by the engine shaft through the intermediation of the vertical governor shaft and two sets of helical gears.

To allow the valves and cages to be lifted without the removal of the entire rocker arm and shaft, the rockers are in two parts.

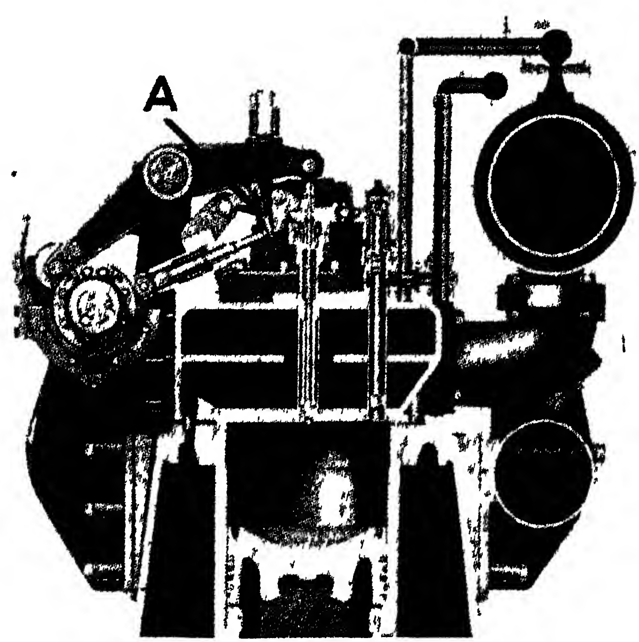


FIG. 186.—Valve gear of a Cooper-Bessemer Diesel.

These two parts are held together by one bolt *A* (Fig. 185) while two dowel pins *B* preclude the possibility of the parts being improperly aligned on reassembly. One end of the rocker carries a hardened-steel roller, and the other is fitted with the adjusting screw *G*, which screws into a steel pin and is locked by the nut *H*.

Worthington Valve Gear.—As shown in Fig. 104 *A*, the valves of a Worthington Diesel are operated from a camshaft in the crankcase.

Exhaust and intake valves are made of heat-resisting material. In the smaller engines they seat directly in the cylinder head and operate in removable bushings. In the larger engines they operate in removable cages. Exhaust and intake valves are interchangeable. Exhaust-valve cages are water cooled.

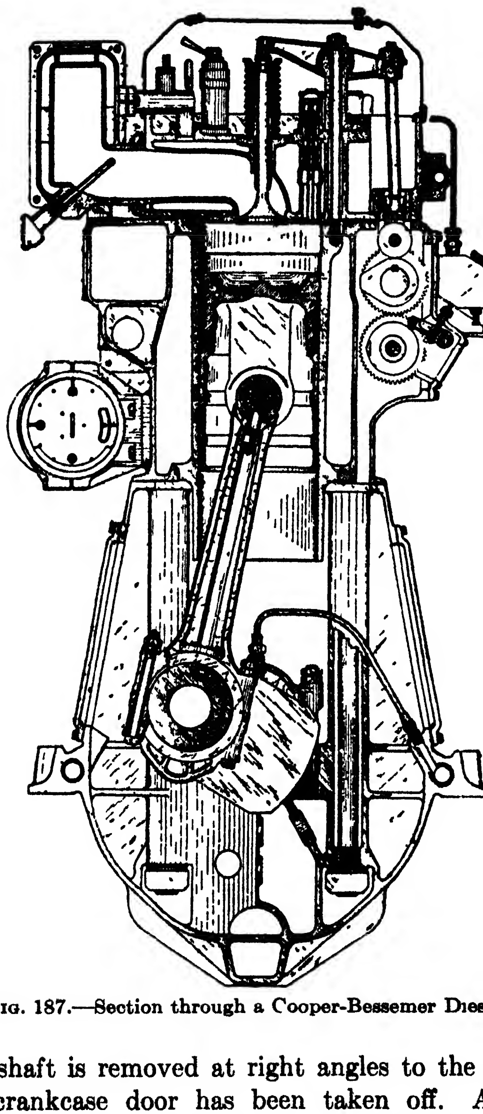


FIG. 187.—Section through a Cooper-Bessemer Diesel.

The camshaft is removed at right angles to the engine after the front crankcase door has been taken off. All camshaft bearings are split and adjustable. The camshaft is driven by spur gears with silent spiral teeth, at the flywheel end, because at that end the crankshaft is steadied by the flywheel, and the gears run smoothly and silently.

Cams, rollers, and pushrod guides are lubricated from a cast-in oil chamber extending over the full length of the engine and supplied by the main oil system.

Cooper-Bessemer Valve Gear.—Originally the Bessemer Diesel followed the Atlas-Imperial design, but engines built since 1926 have overhead valve gears driven by silent chains. One Cooper-Bessemer arrangement is shown in Fig. 186. The valves are provided with cages of standard design.

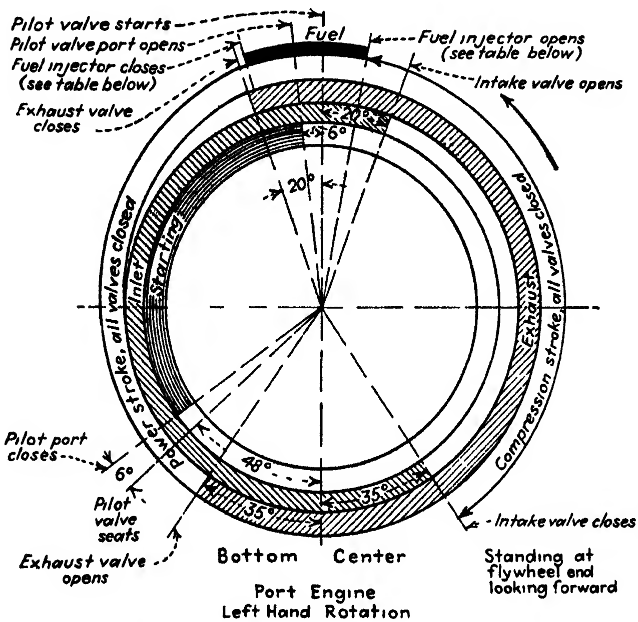
On other Cooper-Bessemer Diesels the valve gearing is as shown in Fig. 187. The camshaft is hollow, with separate forged-steel, casehardened cams, keyed in place with three tapered keys. It is carried in bronze-bushed bearings mounted on the side near the top of the cylinder block. Lubricating oil is forced to each of these bearings through the hollow camshaft. It is driven by an adjustable roller chain and gears from the after end of the main shaft.

The inlet and exhaust valves seat directly against the head and operate in removable guides. They are operated by forked rocker arms and short pushrods, the lower ends of which rest against hardened cup blocks carried in guides with hardened rollers. All wearing surfaces are hardened and automatically lubricated. The exhaust valves are forged from a special high-heat-resisting metal.

Valve Clearances.—The clearance between valve bushing and exhaust-valve stem should be 0.005 to 0.006 in. when first fitted up. If the clearance is less than this, the valve will stick at full load. The intake valve can be run closer.

After the valve and cylinder head have been assembled on the engine, and the rocker arms replaced, turn the engine until the cam roller is on the concentric part of the cam. Then adjust the rocker arm until it just begins to pinch a feeler, or thickness gage, placed between the button and the valve stem, which is 0.018 in. thick for exhaust valves and 0.012 in. for intake valves. Set up the locknut, and check again, for if the threads are slack, they usually increase this clearance. After the clearance is correct with the engine cold, check again after the engine has run an hour or two on its full rated load. Clearance should then be between 0.006 and 0.010 in. for both intake and exhaust valves.

Timing the Intake and the Exhaust Valve.—The valve-timing diagram for a port engine is shown in Fig. 188. Note that the direction of rotation is taken facing the main drive and looking forward. This drawing will serve for the starboard engine, by regarding it as looking toward the engine from the forward



Fuel Timing			
Eng.	RPM	Opens	Closes
MR	225 to 300	10	20
LR	225 to 300	10	20
KR	200 to 300	10	20
JR	250 to 350	10	20
JT	200 to 325	12	18
FP	350 to 400	10	20
FP	400 to 550	12	16
EP	400	10	14
EP	500	12	12
EP	650	14	10
EN	700	18	10
EN	800 to 900	22	6
GN	300 to 400	12	16

FIG. 188.—Typical valve timing of a Cooper-Bessemer Diesel.

end. The camshaft is gear and chain driven from the crankshaft, turning in the opposite direction and at half the speed.

General Motors Valve Gear.—As will be seen from Fig. 194, the high-speed Diesel built by General Motors employs two exhaust valves in each cylinder head. These valves seat on the cylinder head and are operated by short pushrods from the camshaft which runs along the cylinder casting. The camshaft is gear driven from the crankshaft.

Other Valve Designs.—No attempt has been made to cover the valve gears used by all of the engine builders; only typical examples have been taken up.

Valve Material.—At one time all exhaust and admission valves were of cast iron with steel stems. At present the higher speeds are calling for better valve material; consequently, a number of builders are using vanadium steel and other special alloys.

Valve Lifts.—When the valve seat is at a 45-deg. angle, the lift must be $0.35d$ if the port and valve areas are to be equal; in this d is the valve diameter.

Pitting and Corrosion of Valve Seats.—Leaky exhaust valves may be due to any of the following causes: cutting effect of the highly heated gases passing at high velocity between valve and seat; carbon or dirt from the fuel becoming caught between valve and seat and hammered into the metal; corrosive effect of acid in the fuel; warping of valve disk, causing uneven seating; valve stem striking in guide, preventing the valve from seating firmly.

Cleaning Valves.—Every 3 months each valve and cage should be removed for inspection and cleaning. A regular schedule can be followed whereby the valves of one cylinder are removed every 3 weeks; in a four-cycle engine this gives a three-week inspection, whereas in a three-cylinder engine an inspection every 3 months is secured. When the engine is a single- or double-cylinder unit, the schedule should be arranged so that the trimonthly examination is obtained. At these inspections the valve and cage should be completely disassembled and thoroughly cleaned with kerosene, washing off with gasoline. If a valve and cage are kept on hand, the odd one may be cleaned at leisure. This spare set is very essential where an engine operates almost continuously. The time required for removal and replacement of cage and valve should not exceed 30 min. when the engine-room force is well organized.

Grinding Valves.—If the valve seat becomes rough, allowing the compression to escape, it must be reground. Where the valve seats in a cage the unit is disassembled. The valve and dashpot, or spring cap, as it is more popularly termed, is replaced in the cage with a light spring resting between the valve body and the stem housing, along the lines of Fig. 189. The cage is inverted, placed on some form of support, and the valve-pin wrench is then set in position. The spring raises the valve off the cage until a slight downward pressure is exerted by the man doing the regrinding. A mixture of emery flour and oil or emery flour and vaseline should be coated over the valve face, and the valve rotated back and forth. The valve should, in no case,

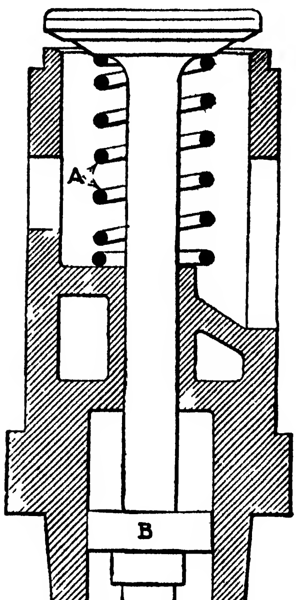


Fig. 189.—Regrinding valves.

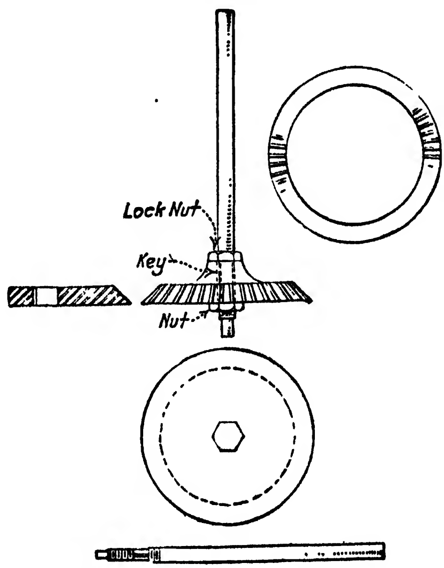


Fig. 190.—Valve-seat reamer.

be completely revolved; the rotation or movement should cover a trifle more than a quarter circle. After rotating a few minutes the valve should be moved another 90 deg., and the rotation renewed. This grinds every portion of the valve face to conform to the entire seat of the cage. As the operator grinds the valve, he should release the downward pressure on the valve, allowing it to rise from the seat. This serves to distribute the emery paste over the entire face. If this is not observed, the paste forms at the edges only, causing the valve seat to be ground concave.

The operator need not secure a ground seat over the entire valve face. A narrow contact $\frac{1}{16}$ in. wide is ample; in fact, a line contact of $\frac{1}{32}$-in. width is as serviceable as a seal as is a wider space. After the seat is sufficiently ground, the emery paste can be removed. The valve should then be rotated without any paste between the two surfaces; this metal-to-metal grinding or rubbing will make the area of contact as smooth as a mirror and prolongs the tightness of the valve.

When the valve becomes so pitted or grooved that grinding will not be of any avail, the cage and valve faces must have a light cut taken off their surfaces. The valve can be centered

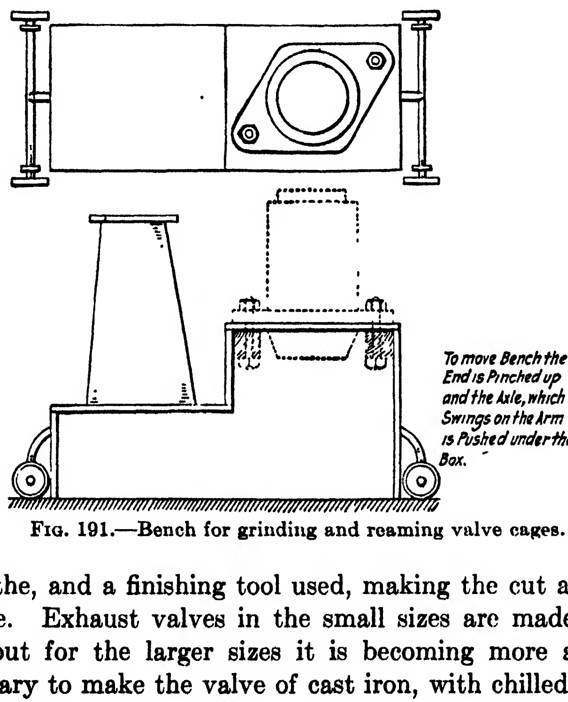


Fig. 191.—Bench for grinding and reaming valve cages.

in a lathe, and a finishing tool used, making the cut as light as possible. Exhaust valves in the small sizes are made of alloy steel, but for the larger sizes it is becoming more and more customary to make the valve of cast iron, with chilled surfaces, and fit it on a steel stem. In such cases recourse is had to the grinding wheel, as the metal is too hard to be cut by a lathe tool or reamer.

To reface the cage a refacing machine, similar to a rose reamer, is necessary. This reamer must have a stem resting in the valve-stem housing to hold the reamer square with the valve stem. Most manufacturers are in a position to supply this machine, although any machine shop can build one like that shown in Fig. 190. The cutting head should be of tool steel, but the stem can be of either tool steel or machine steel. The latter is preferable, since the cutting head can be hardened and the machine-steel stem turned to bring the cutter concentric.

Few plants possess any stand to hold the cage other than a wooden box. The box is a poor accessory, since it is almost impossible to seat the irregularly formed cage on it with any feeling of security. A wooden grinding stand, shown in Fig. 191, can be made during odd hours and certainly repays all labor spent on it. The opening in the top should conform to the shape of the cage, and the two stud bolts should fit into the regular stud-bolt holes in the cage. With such an apparatus an engineer can sit down while grinding, thus lessening as much as possible the labor involved in this operation.

On new engines it will nearly always be found that both the exhaust and the air-inlet valves will take a permanent set after a short time in use and must be taken out for regrinding, after which they are less liable to deformation. In the case where exhaust and air-inlet valves are duplicates, it is advisable, after a few weeks' running, to interchange them. This is known as "seasoning" the valves. In some engines, especially the larger sizes, the blowtorch action of the fuel spray causes the exhaust-valve disk to become much hotter on the side adjacent to the fuel-injection valve. This uneven heating may cause serious warping of the valve. If the valve construction is such that the valve can be rotated by means of a wrench on the upper part of the stem while the engine is running, the operating routine should call for giving the valves a turn at least once every hour. This will tend to prevent permanent warping, due to the uneven heating.

Sticking of the exhaust-valve stem in its guide may be caused by too much clearance, permitting gummy oil and carbon to work up between stem and guide, as in the case of the spray valve. This action is accelerated, when very heavy fuel oil is being used, by the fact that the heavy unburned residue that may be carried in the exhaust gases in a finely divided state strikes against the exhaust-valve stem and sticks. This action occurs when the valve is pushed downward to the open position, thus exposing a part of the stem which travels up into the guide when the valve closes.

When a valve is removed for overhaul, the stem should be carefully cleaned, as described in the case of the spray valve.

Both the valve and its seat should be examined for fine heat cracks which may develop into leak paths. It is good practice to run kerosene through the valve-stem lubricator occasionally while the engine is in operation. This tends to clean out gummy oil that may collect in the guide.

The treatment of air-inlet valves should be the same as for the exhaust valves, but it will be found that they will not require regrinding so often. Because of the cold air passing through the inlet valve during the suction stroke, it never gets so hot as the exhaust valve, and it has none of the dirt and unburned oil in the exhaust gases to handle.

Replacing Valves.—When a valve is fitted with a cage, extreme care must be taken to have the gasket at the lower end of cage perfectly clean before the cage bolts are tightened.

In almost any group of engineers discussing Diesel maintenance, someone remarks that for some reason he cannot grind a valve to a seat in the valve cage without a leak developing after the cage is put back into the cylinder head and tightened up. This trouble does occur occasionally, and more than one chief engineer has charged his maintenance man with doing a poor grinding job. Usually the man is not to blame.

In fact, in one particular plant where two engines had valves seating on the cylinder head, no trouble was ever experienced with these, but an occasional leak developed with the engines having caged valves.

The cause is not poor grinding but distortion of the valve cage during the process of tightening the bolts.

When the cage is free, it assumes its natural shape. The maintenance man proceeds to grind the valve to its seat. He gets a nice finish, and a test with a pencil mark proves that there is a contact all around the seat.

He proceeds to put the cage back into the cylinder head, cleans the bottom gasket, and tightens up the two nuts. The valve cage is drawn out of shape, with the result that the valve fails to close tightly, or the gasket fails to seal the cage.

If the leak is at the gasket, the tendency is to tighten up some more, which merely makes matters worse.

When this trouble occurs frequently with a particular engine, the only solution is to grind the valve while the cage is bolted to the cylinder head. This, of course, is more time-consuming, for the head must be removed from the engine.

Valve Timing.—In timing valves the first step is the establishment on the flywheel of the points of dead center of the cranks. The simplest method of marking the dead centers is to use a steel trammel having both ends pointed. A steel block, with a counterpunch mark on the surface, can be inset into the foundation, being held by lead or cement grouting (Fig. 192). To establish the flywheel position, when the crank is on upper dead center, one of the valve cages can be removed, and the distance

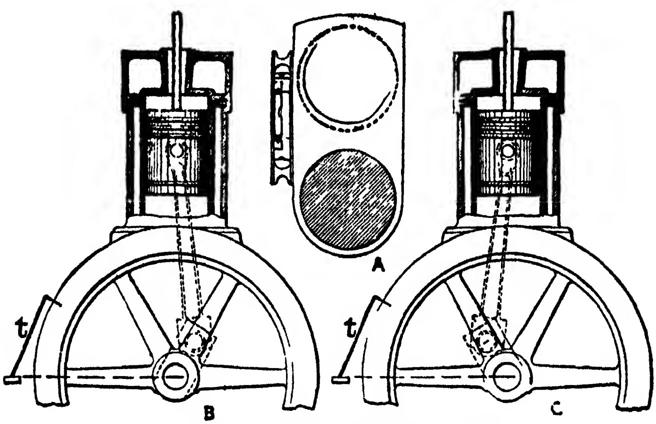


FIG. 192.—Locating crank center position.

from the surface of the cylinder head to the piston, when the piston is approximately at upper dead center, can be measured, and a trammel mark made on the flywheel rim. The wheel is then turned on over past dead center until the piston is again the same distance from the cylinder head. A second mark is placed on the flywheel rim; the bisection of the distance between the two trammel marks gives the exact dead center for the piston in question. A second but not so exact method is the use of a spirit level on the crank throws; this, of course, is impossible if the throws are not machined accurately on all four sides. The center mark should be placed on the flywheel rim and stamped with the cylinder number, as example 2TC, indicating top dead center of No. 2 cylinder. The same procedure is followed on all the cylinders for both top and bottom dead centers. These center positions being determined, the next step is the checking of the exhaust valves. The engine should be turned over until the piston of the cylinder in question is about 50 deg. from bottom dead center. A steel tape line can be used to measure the distance on the flywheel rim from the bottom dead-center line to the point of correct exhaust-valve opening. Since the timing given is in degrees, the value must be transformed into inches on the flywheel circle. Presuming the wheel is 8 ft. in diameter, the circumference being 302 in., each degree represents, very closely, 5/6 in. If the proper opening of the exhaust is 42 deg. ahead of bottom dead center, a distance of 35 in. is set off ahead of the dead-center mark. This is spotted and stamped E2*O*, that is, exhaust opening for No. 2 cylinder. The flywheel should then be turned slowly until the trammel cuts this E2*O* mark. The exhaust cam rocker should be firmly in contact both with the cam and the valve stem. To be doubly certain, the adjusting screw on the rocker should be backed off and brought up again until the operator can feel the resistance of the valve spring against the screw. In case, while revolving the wheel, the mark E2*O* is passed, the valve should not be checked by bringing the wheel back to the mark. Instead, the wheel should be brought back past the mark at least 12 in. and then again turned until the trammel cuts the mark. This removes all effect of any backlash that might exist in the cam gears. The same process is followed in checking the exhaust closing position. After this is accomplished, the exhaust-valve settings of the other cylinders are checked, the flywheel being properly marked for each position. The admission valves should next be gone over in proper sequence, and all points should be indicated on the wheel for future reference.

In the event that any of the valves fail to check up correctly, the operator is confronted with the question as to the method of changing the setting. If the discrepancy is only a few inches, the clearance between the cam and the valve rocker can be changed, bringing the setting back to the stated values. If the valve opens vastly early or late, the only recourse is the shifting of the cams by the use of an offset key. This condition is encountered in old engines only. In these engines quite often the entire valve layout is timed late. This is attributable to the wear in the cam-gear teeth and can be partially corrected by advancing the cam gear one or two teeth.

The average operator, until he is very familiar with engine timing, does well to time the exhaust and admission valves of one cylinder before proceeding to any of the valves of the other cylinders. There is, in this way, little danger of becoming confused as to the proper stroke upon which the valve should function. The trained operator customarily checks the valves as they come in sequence. For example, the exhaust opening of one cylinder will be set; then the admission closing of a second cylinder will be checked. This reduces by at least 75 per cent the time occupied in going over the valves. The checking and setting of the fuel valves will be taken up in the chapter on fuel valves.

It is possible to place a dial upon the end of the camshaft. Many builders of marine Diesels fit a similar dial before the engine leaves the factory.

Leaky Valves.—A leaky admission valve is readily detected by placing one's ear close to the intake nipple or screen. If the leak is from a scored seat, a whistling noise will be heard; this whistling increases in volume as the scoring becomes more pronounced.

A leaky exhaust valve is more difficult to detect. The best method is to place the engine in that position where both the admission and exhaust valves are closed, that is, at the beginning of the power stroke. The injection air-line valve should be "cracked," allowing a small amount of air to blow into the cylinder through the fuel-injection valve. If the exhaust leaks, the air can be heard flowing through the valve. This same procedure will also locate admission-valve leaks. While the engine is running, a smoky exhaust and a decrease in power are often due to a leaky exhaust valve.

Gas Velocity through Valves.—A fair value for the mean velocity of the exhaust gases through the valve is 120 ft. per second. The pressure drop can be calculated from the formula

$$V = 18.3C\sqrt{\frac{h}{d}}$$

where

V = velocity, feet per second.

C = discharge coefficient, 0.7 to 0.95.

h = pressure drop, inches of water.

d = density of gas.

Charging of Two-stroke-cycle Diesels.—As the working piston of the two-stroke-cycle engine does not make an induction or suction stroke, other means must be provided to introduce a charge of air into the cylinder, both to clear it of the burned gases

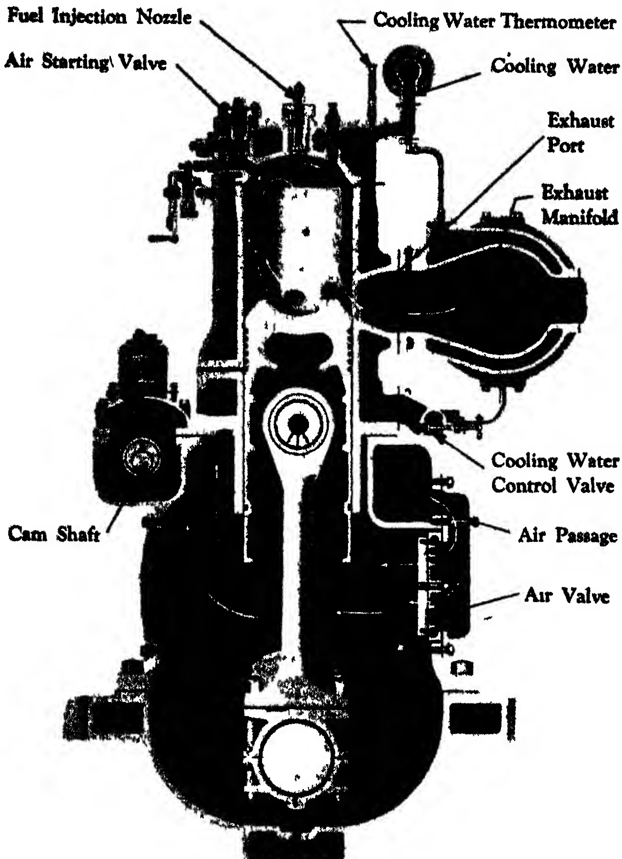


Fig. 193.—Crankcase scavenging design of Fairbanks, Morse.

from the previous cycle and to supply the charge for the next combustion period.

Several general methods are used. In a few large engines the air, the so-called "scavenging air," is introduced into the cylinder through valves in the cylinder head; the air is supplied by a reciprocating air pump driven by the crankshaft or by an independent centrifugal blower. Another type of two-cycle charging employs a row of ports in the cylinder wall to introduce the air into the cylinder and allows the exhaust gases to exit through valves in the cylinder head; the General Motors Diesel is of this design.

Other Diesels employ a row of ports in one side of the cylinder wall to admit air into the cylinder; and at the same time that the piston uncovers these ports, it also uncovers a second row of ports, to allow the exhaust gases to sweep out of the cylinder before the fresh air enters. The air may be supplied by a blower or pump. A most popular design for engines of medium capacity embraces the use of ports for air and exhaust, with the air supplied by the crankcase as on the Fairbanks, Morse Diesel (Fig. 193).

Nordberg Scavenging Design.—As will be seen from Fig. 34, the two-cycle Diesels now built by Nordberg have one row of ports for the entrance of the scavenging air and a second row for the exhaust exit. When the piston moves downward in the cylinder, it uncovers both the scavenging and the exhaust ports. The burned gases tend to rush into the scavenging as well as into the exhaust ports but are restrained by the automatic scavenging valves from proceeding farther. These gases are, therefore, forced to leave by way of the exhaust ports; and when the pressure in the cylinder has dropped sufficiently, air from the scavenging-air header rushes through the automatic valves, expelling the burned gases from the cylinder. The scavenging valves remain open while the piston moves to bottom dead center, reverses its travel, and moves upward until the pressure in the cylinder again exceeds that in the scavenging-air header, whereupon the valves are automatically forced shut. Gases from the cylinder cannot blow back into the scavenging header, since the valves permit flow in but one direction. The scavenging pressure on this type of engine varies from 1.5 to 2.5 lb., depending on exhaust and other conditions.

Scavenging-air Pump.—The scavenging air is supplied by a double-acting pump whose piston is driven through a rod, by a crank at the end of the crankshaft.

The air-pump valves are steel strips mounted in cages. Because of the low pressure carried, these valves have a long life. Replacement is not difficult. The unshipping of a single nut enables the engineer to remove an entire valve cage, and the valve guard may be removed by unloosening two holder bolts.

The valves should be examined at the end of 3,000 hr. of operation, or sooner if the compressor seems unable to maintain the air-header pressure of 5 to 8 lb.

Busch-Sulzer Scavenging.—The Busch-Sulzer engines have port scavenging, with two sets of air ports. Originally, the upper set (Fig. 34) was controlled by a rotary valve. This prevented loss of air through these ports; the scavenging air entered through the lower ports; and the exhaust gases escaped through the exhaust ports on the opposite side. When the exhaust ports were covered by the up-moving piston, the rotary

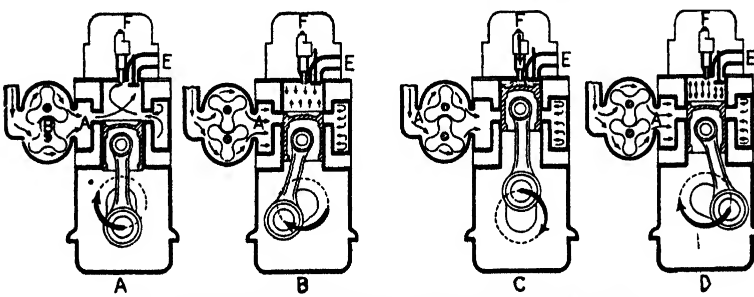


Fig. 194.—Scavenging action in General Motors Diesel.

valve turned to the open position, and more air entered the cylinder to provide a full air charge. Later, the rotary valve was replaced by a shutter valve, and the cylinder gas pressure held these closed until the exit of gases through the exhaust port lowered the cylinder pressure below the 2½ to 4-lb. air pressure in the manifold. Several years ago the shutter valve, which could not be kept tight, was replaced by strip, or ribbon, valves mounted on boxes, or cages. The design has the merit of simplicity, and the wear of the valves is insignificant.

The scavenging air is supplied by a motor-driven blower, although in the smaller units a reciprocating air pump may be driven from the crankshaft.

General Motors Scavenging System.—In Fig. 194 are shown four diagramatical outlines of the General Motors two-cycle Diesel. This engine has port scavenging and valves in the head exhaust. Scavenging air is supplied by a lobed blower *B* driven through gears (not shown) from the crankshaft. Air enters this blower and is forced by the lobes, or vanes, into the air header *A*, which consists of two passages along the sides of all the cylinders of an engine.

When the piston is at bottom dead center, as shown in Fig. 194 *A*, air in the header *A* at about 5-lb. pressure flows through ports into the engine cylinder. The air sweeps up into the cylinder and forces from it all burned gases; these gases along with some of the air escapes through the exhaust *E* by way of the exhaust valve which is opened by a cam (not shown) driven by gears from the crankshaft. In this particular engine built by General Motors there are four exhaust valves in each cylinder head.

The crankshaft continues to rotate, and the piston moves up and, by passing across the air ports (called "scavenging ports"), prevents more air from entering the cylinder. At the same time the exhaust valve closes under action of the cam already mentioned.

The cylinder is now filled with pure air; and during the compression stroke, whose beginning is shown in Fig. 194 *B* and whose end is shown in Fig. 194 *C*, this air is compressed to almost 500 lb. pressure.

At the piston position shown in Fig. 194 *C* a spray valve *F* in the cylinder head opens to inject a spray of fuel oil at high pressure into the mass of hot air crowded into the small space above the piston crown.

Ignition occurs at once, and the gas pressure rises sharply. This pressure forces the piston down as shown in Fig. 194 *D*. Here the piston is almost in a position to permit air to enter through the rows of scavenging ports, and the exhaust valve in the cylinder head has already opened.

The cylinder gases start to pass out through the exhaust *E*; and as soon as the scavenging ports are free, air blows into the cylinder, while the piston moves from the position shown in Fig. 194 *A* to that shown in Fig. 194 *B*. In this way all the burned gases are forced out of the cylinder, and the cylinder is charged with fresh air.

The advantage of this system of scavenging is that the flow of air and of burned gases is in one direction, from bottom to top, so the cylinder is well cleared of burned gases, and the engine is able to burn a high rate of fuel per cubic inch of cylinder volume.

Crankcase Scavenging.—The majority of small two-stroke-cycle Diesels, up to 300 hp. each or 90 hp. per cylinder, make use of the crankcase as the scavenging air compressor. The advantage is one of convenience only, for it is obvious that the volume of air handled can be only the piston displacement, or a 1:1 ratio—too small to give good scavenging. As a result, the amount of fuel possible to burn per unit of cylinder volume is low, for about one-half of the cylinder contents is burned gases from the previous explosion. A mean effective pressure of over 45 lb. is difficult to obtain. Tests have indicated that the inertia of the exhaust gases is chiefly responsible for the induction of the air charge.

Fairbanks, Morse Scavenging.—In Fig. 193 is shown a cross section of the 14 by 17-in. Fairbanks, Morse two-cycle Diesel. This engine is provided with crankcase scavenging, with the air entering the crankcase through an air valve placed at the side of the crankcase. The air enters while the piston is moving upward on the cylinder compression stroke. On the piston's downstroke this air is compressed by the piston action to about 3 lb. pressure; and when the piston uncovers the air ports, air from the crankcase flows up along a passage, not shown in the illustration, and enters the working cylinder. The ports are shaped to cause the air to flow upward and not across to the exhaust ports. The improved scavenging has permitted the horsepower output to be increased.

Rear of Cylinder Compressor.—Several solid-injection Diesels employ the lower end of the cylinder for the scavenging air compressor. Here also the air ratio is but 1:1. The design, however, eliminates the carrying over of the crankcase lubricating oil to the cylinder. On the other hand, the construction is expensive, and the piston is poorly cooled, so that oil or water cooling must be resorted to when the piston diameter becomes greater than 12 in.

Reid Scavenging.—In Fig. 101 is shown a section through the Reid two-cycle Diesel. This engine employs rear-of-cylinder compression of the air charge, which enters the working cylinder through a row of ports.

Superior Two-cycle Scavenging.—In Fig. 195 is shown a section through the National Superior vertical, two-cycle engine which may be used as a gas engine or a Diesel. As will be seen, the lower end of the cylinder serves as the compressor for the scavenging air. An air header runs along the side of the engine frame, and the air enters the compression space through a set of light ribbon valves. Naturally, the cylinder space is separated from the crankcase by a partition which carries a set of metallic packing rings about the piston rod.

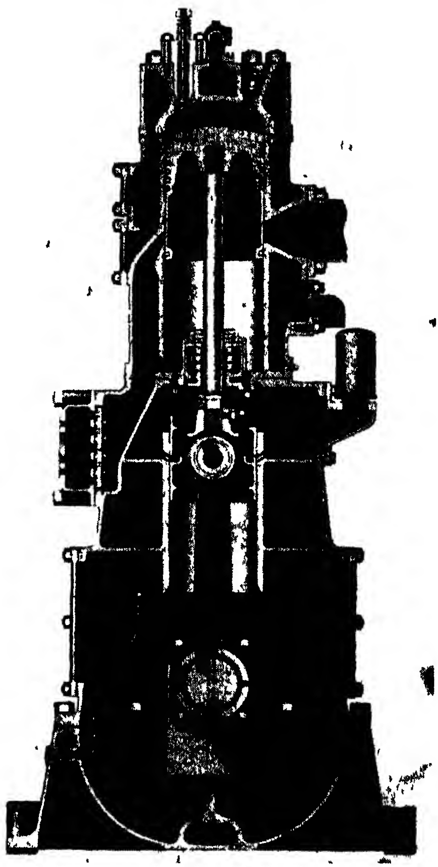


FIG. 195.—Superior two-cycle Diesel employs lower end of the cylinder for the scavenging-air pump.

Carbon Deposits in Exhaust Pipes.—In two-cycle solid-injection Diesels there is a tendency for carbon and unburned fuel to deposit in the exhaust ports and pipes. To remove the carbon the best method is to burn it out by applying a torch. The inspection plate should be opened enough to admit sufficient air to consume the carbon. Care must be taken to insure against firing the roof or adjoining property. The removal should never be done when there is a wind.

Pyrometers.—As a ready check upon combustion in the engine cylinders the pyrometer is a two-piece, efficient accessory. In addition, it enables the operator to detect misadjustments of the several fuel pumps and spray valves.

A six-cylinder Diesel engine, for example, is really operated as six separate units. Perfect coordination in the operation of the six units must be obtained so that the loads are uniformly distributed throughout the cylinders or units of the engine. Vibration must be held to a minimum to avoid the setting up throughout the engine of uneven stresses which are detrimental to the operation and life of the engine. Knowledge of the exhaust temperature of each cylinder is of vital importance to check the operation of the various units. To insure smooth, harmonious action, the exact temperatures of all the cylinders must agree with each other very closely.

Among other things, a pyrometer will detect such ills as partially clogged injectors, causing a deficiency in the fuel feed; sticking injectors, causing an excessive fuel feed; and faulty action of the valves.

An unbalanced condition in a Diesel engine, which a pyrometer will show, may cause broken pistons, broken connecting rods, or broken crankshafts.

Theory of Pyrometer.—All pyrometers used on Diesel exhaust lines are based on the same principles. Two dissimilar wires (Fig. 196) are brazed together; and when they are heated, a difference of electrical potential or voltage exists between the joint (which is exposed to the heat) and the outer cold end. The temperature difference between the two ends is measured by a voltmeter reading minute potential differences, that is, a millivoltmeter.

Most of the pyrometer thermocouples are made of iron and constantan metals. The direction of current flow depends upon the metals used and upon the temperature. If the lead to the millivoltometer is of the same metal as the thermocouple, the cold ends will be at the instrument, which is usually in a position where the temperature variation is small. Such leads are expensive, so it is becoming a practice to use copper wire.

The only disadvantage of using copper wire is from the cold-junction standpoint; with the copper wire the cold junction is at the terminals of the thermocouples, and these terminals, being relatively close to the engine, are subjected to wider temperature fluctuations or variations than if the cold junction were at the instrument.

It is therefore necessary when using copper wire to see that the cold-junction setting of the pyrometer is carefully done, in order that there may be no error due to this cause. The cold-junction temperature reading should be taken when the engine is running, so that the pointer can be set to the proper cold junction, as the heat of the engine may have an effect upon this setting.

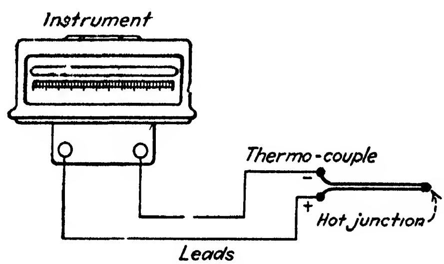


FIG. 196.—The millivoltmeter type of pyrometer.

In some cases where they are using relatively short thermocouples with the terminals close to the engine, it has been observed that the cold-junction setting varied in some instances up to as high as 140°; therefore the pointer of the pyrometer when the switch is in the off position should have been set for 140° to have obtained accurate readings.

Since the exhaust temperature readings of the Diesel engine are largely for comparison, absolute accuracy is not so essential, for what is wanted is accurate comparison between the exhaust temperatures of the several cylinders.

Usual practice is to attach the leads from each of the thermocouples to a contact switch, so that each in turn may be connected to the voltometer, permitting reading of the temperature of each engine cylinder.

Although the millivoltmeter type of pyrometer is more common, some manufacturers supply potentiometer types. In a pyrometer of the potentiometer type, the e.m.f. developed by the thermocouple is balanced against a known e.m.f., and the reading is obtained when this balance is exact, that is, when no current flows. Under these conditions it does not matter what changes may take place in the resistance, length, or size of the leads or of the thermocouple.

Pyrometer Installation.—When an exhaust pyrometer installation is being planned, several points should be taken into consideration:

1. If more than one engine is to be used in the same location, careful consideration should be given to the question of whether it is desirable to have one pyrometer to serve all cylinders of the several engines or to have a separate pyrometer for each engine.

2. In asking for recommendation and prices on a pyrometer installation, it is suggested that the following information be given:

a. Size and type of engine and total number of cylinders.

b. Service for which engine is used—stationary, marine, mobile.

c. Does service require a waterproof installation?

d. Where will the pyrometer itself be mounted? On a switchboard, post, wall, or directly on the engine?

e. Style of thermocouple desired or required.

f. The average length of connecting wire that will be required to connect each thermocouple to the switch of the pyrometer.

g. Are extremely accurate readings desired, or is an accuracy to within 15 or 20°F. permissible?

Temperature readings of the exhaust gases are largely for comparison between the cylinders. Extremely accurate readings are, therefore, not necessary, but rather an accurate comparison of the several cylinders. This being true, duplex copper wire can be generally used to connect the thermocouples to the switch of the instrument.

As the pyrometer reads the difference in millivolt output between the hot end and the cold end of the thermocouple, the pointer of the pyrometer must be set for the temperature at the cold end. Copper wire is not compensating; when it is used, the cold end of the pyrometer system is at the point where the copper wire is connected to the thermocouples. Therefore, with copper wire the pointer of the instrument must be set for the temperature that prevails at the thermocouple terminals. This setting is made when the switch is turned to the zero position. If the temperature at the thermocouple terminals where the copper wires are attached is found, for example, to be 75°F., then the pyrometer indicator pointer with the switch in the zero position should read 75°F. on the scale on the face of the instrument.

The stems of the thermocouples should extend into the approximate center of the exhaust-gas passage in the exhaust pipe in the direct path of the gases and should be inspected periodically, particularly if the engine is operating with smoky exhaust or high heat. Carbon must not be allowed to accumulate to any extent on the thermocouple stems, as this will insulate them from the temperature and cause a lag in the reading. How often this examination should be made will depend on the condition of the engine; a dirty engine may deposit excessive carbon on the thermocouples which must be cleaned off as needed to give correct and sensitive readings. This involves no difficulty other than removing the thermocouples from the exhaust pipe. If the wires have been disconnected from the thermocouples to allow for removal, be sure that they are properly connected when replacing the thermocouples and that each wire is connected to the same terminal from which it was taken.

Exhaust Temperatures.—At least 75 per cent of modern oil engines are fitted with exhaust pyrometers or thermometers. Unfortunately, these instruments generally are not operated as they should be—the engineer either places a blind faith in them or disregards them entirely. In spite of this situation there is no doubt that when properly used and interpreted, exhaust-gas-temperature readings can be relied upon to show how combustion is proceeding in the engine cylinders.

Slight variations in the readings from the several cylinders of an engine should not be taken as proof that the spray-valve timing needs adjustment. Furthermore, engines of different types show marked differences in the temperature readings at various loads, and one must know the characteristics of a particular engine if the pyrometer readings are to be of value.

In general, however, an engine of a given type will show temperature characteristics peculiar to that system. In Fig. 197 are four curves of exhaust temperatures plotted against loads. These are based on actual readings obtained from engines of each type shown. It will be seen that the exhaust temperature of the four-stroke-cycle, air-injection Diesel is directly proportional to the load, the several points forming a straight line.

A four-stroke-cycle, solid-injection, common-rail, or timed-spray-valve, engine develops a line that is curved, with the full-load temperature showing a higher value that the one-half- and three-quarters-load value would lead one to expect. This is due, no doubt, to the type of spray valve used on this engine. Probably the fuel charge was such that not all of it burned immediately upon injection, in which event afterburning and a high exhaust temperature would naturally follow. In spite of this, however, the full-load exhaust temperature is below that of the air-injection engine. This is due, in the main, to the greater weight of air per pound of oil burned, which reduces the temperature of the exhaust gas.

The two-stroke-cycle, crankcase-compression engine, a four-cylinder, 480-hp. unit, has a curve that is much lower than those of the two engines already discussed. This is due, primarily, to a low mean effective pressure, which is a result of a large air-to-fuel ratio. Just why it should show such a steep inclination is debatable, probably because of late- or after-burning on heavy loads.

That the two-stroke-cycle engine has a lower temperature than have the other two engines is also due to the scavenging arrangement and to the fact that the pyrometer does not indicate the actual temperature of the burned gases resulting from the combustion of the fuel. When the piston uncovers the exhaust and scavenging ports, hot, burned gases sweep out and are followed immediately by a flow of scavenging air from the scavenging ports. The pyrometer then indicates a temperature that is the mean of the burned gas and the scavenging air temperatures.

The curve of the two-stroke-cycle engine with separate scavenging pump should not be taken as characteristic of that type of engine. The high temperatures at three-quarters and full load are undoubtedly caused by the condition of the fuel-spray nozzles. This engine should have a curve closely approximating that of the crankcase type.

Attention is called to the extensions of the curves to zero load. Apparently all four engines have a no-load exhaust temperature of about 230°F. If the curves are extended to the left of the zero-load curve to a point corresponding to zero indicated horsepower, the four-stroke-cycle engine shows an exhaust temperature of about 150°F., which would likewise be that in the cylinder at the beginning of compression, assuming adiabatic compression and expansion. On the other hand, the two-stroke-cycle solid-injection engine would have a zero indicated horsepower exhaust temperature of about 200°F. and a corresponding initial temperature, which is about the value to be expected. It appears from this that the exhaust-temperature curve could be used to ascertain

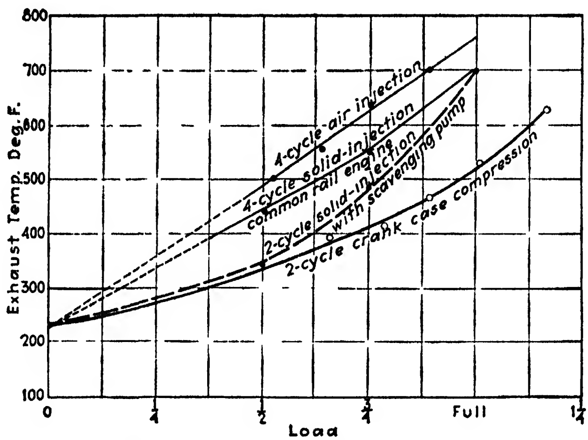


FIG. 197.—Comparative exhaust temperatures of various types of engines.

the weight of the cylinder air charge, without chemical analysis or complicated calculations.

The conclusion reached from the curves in Fig. 197 is that before using the exhaust temperature as an indication of cylinder events, the engine's characteristics must be known.

The several cylinders of an engine should have exhaust temperatures of identical values. It is, however, difficult in some designs to prevent variations of 10 to 30°. Usually this range is not objectionable. The several cylinders of large engines will show more uniformity than will those of small engines, because the fuel-spray valves of the large slow-speed engines can be timed more closely.

In Fig. 198 is a series of readings taken from a six-cylinder, four-stroke-cycle, air-injection engine. It will be seen that the temperatures vary with the load, but Nos. 1 and 6 cylinders show temperatures about 25° under the others. Here the discharge valves of Nos. 1 and 6 fuel pumps leak a trifle and need regrinding. The variation does not indicate enough leakage to justify stopping the engine.

Sometimes the temperature of an engine's cylinders is not uniform and does not show consistency in its variations. For

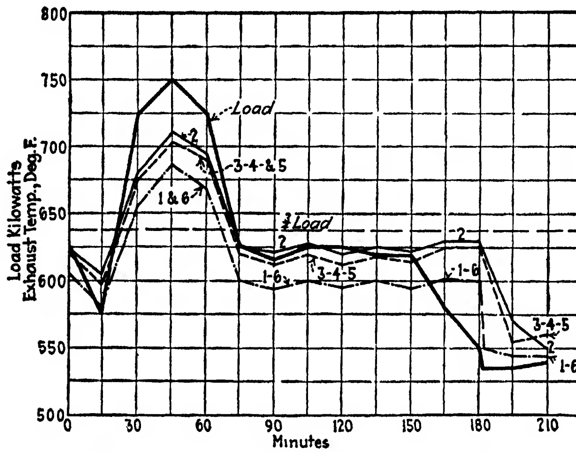


FIG. 198.—Exhaust-temperature changes plotted against time.

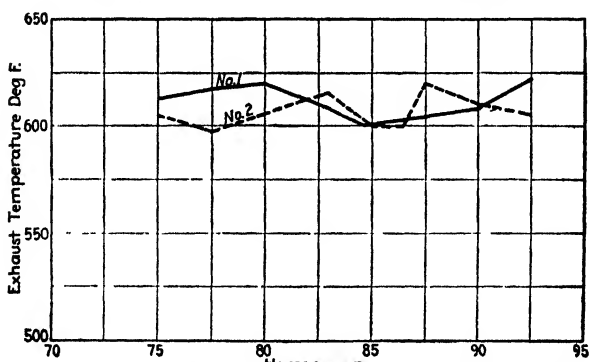


FIG. 199.—Temperature variations due to governor linkage.

example, in Fig. 199 are shown the readings obtained from a two-cylinder, 90-hp., four-stroke-cycle, mechanical-injection engine. The temperatures of the two cylinders vary as much as 40°, although the load is fairly uniform. The inconsistency of the readings is due to the governor mechanism. The travel of the two fuel-pump plungers is controlled by wedges and a linkage from the governor sleeve. Irregularities in the shape of the wedges and lost motion in the linkage cause erratic fuel charges.

Similar behavior of a four-stroke-cycle, common-rail, or timed-needle-valve, solid-injection engine is shown in Fig. 200. The four cylinders failed to give consistent readings over the load range. In this instance the needles of the spray valves were

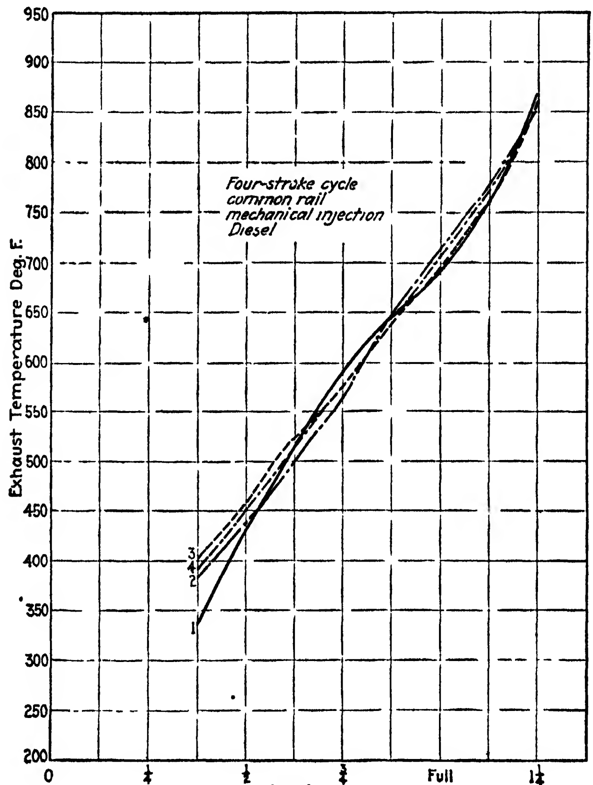


Fig. 200.—Governor and pump clearances cause temperature variations.

actuated by rockers and pushrods, and the governor, by shifting a wedge in the bottom of each rod, altered the valve lift. The temperature variations were due to the governor mechanism. An engineer should not become excited when he finds a similar condition in an engine of this design.

The engine speed also has an influence upon the exhaust temperature for two reasons: The higher speed gives less time for heat to flow to the cylinder walls, consequently raising the exhaust temperature. Of more importance is the handicapped combustion at high speeds. The time interval of ignition and combustion is so short that burning may continue until the end of the stroke.

One of the two charts reproduced in Fig. 201 was developed by E. G. Magdeburger, and the other by M. L. Maleev. It will be seen that with both the common-rail and the check-valve-atomizer engine the exhaust temperatures increase with the load but that at a given mean effective pressure and speed the temperature readings from the two engines are not the same. For

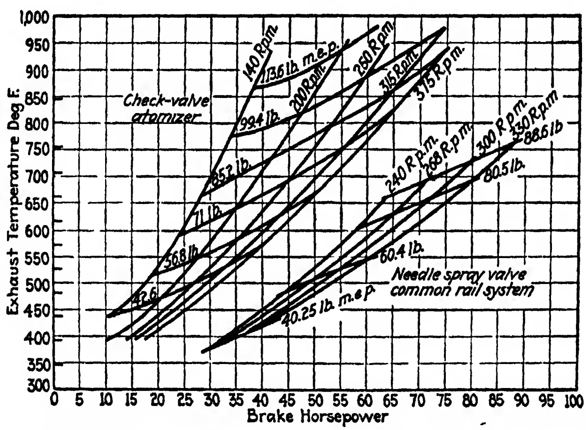


FIG. 201.—Temperature variations with speed and load of two mechanical-injection Diesels.

example, at 315 r.p.m. and 85.2 lb. m.e.p. the engine with the check-valve atomizer has an exhaust temperature of 850°F., whereas the common-rail engine at 300 r.p.m. and 85 lb. m.e.p. shows approximately 700°F. exhaust temperature. A marked difference exists at all loads.

The proposition has been advanced that the engine builder should test an engine of a given size and supply a series of curves to the purchaser of any other engine of the same dimensions and type. With this before him the engineer could tell whether or not he was obtaining reasonable results in fuel consumption.

A similar set of curves from a two-stroke-cycle, solid-injection Diesel is shown in Fig. 202. With this engine the maximum mean effective pressure is 64 lb., and the limiting temperature 700°F.

The conclusion to be drawn from the several sets of temperature readings is that the exhaust pyrometer is a necessary adjunct to any engine, and hourly readings should be taken and set down in the plant log. But the operator should not allow a difference of 10 to 20° in the readings from the several cylinders to cause him to adjust the fuel pump or valves. He must remember that the error may be in the pyrometer itself; for if the terminal in the exhaust connection becomes coated with soot, the reading from that cylinder will be low because of the insulating effect of the carbon.

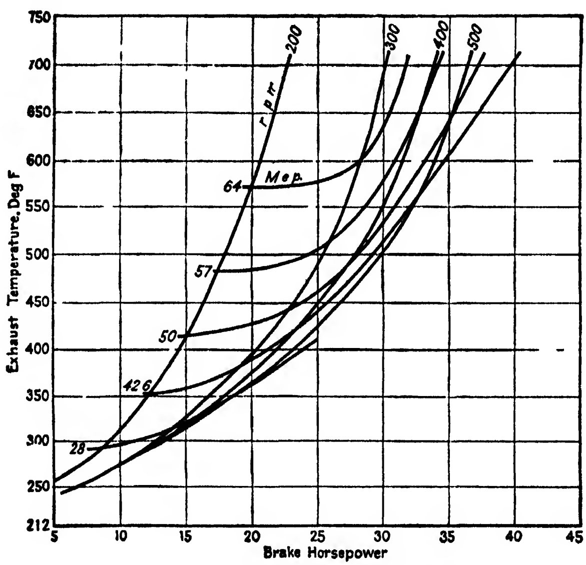


FIG. 202—Results obtained from a two-stroke-cycle engine.

Exhaust Heat Recovery.—Diesel engines use from 16 to 40 lb. of air per pound of fuel burned, and the waste gases leave the cylinder at temperatures ranging from 400 to as high as 800°F. Obviously, these gases hold a great amount of heat, part of which could be recovered if it were transferred to some other fluid, such as air, water, or steam.

Steam Recovery.—By sending the exhaust gases through a boiler or heat exchanger, steam can be generated by heat absorption from the hot gases. The amount of steam per engine horsepower depends upon the engine design and economy. If an engine converts 38 per cent of the heat in the fuel into mechanical work and uses only 16 lb. of air per pound of fuel, the exhaust temperature might be fairly high; but owing to the low air-fuel ratio, the actual heat in the exhaust would be far less than with a Diesel using 25 lb. of air per pound of fuel but which had so poor a combustion system that the exhaust gases ran high in temperature. Consequently, all rules for estimating possible heat recovery are open to question unless the actual weight and temperature of the exhaust gases are known.

The recovery is also dependent upon the permissible gas-exit temperature. This, in case of steam generation, should be at least 50°F. above the steam temperature. In case of air or water heating, the exit temperature must be high enough above the water-exit temperature to hold the heating-surface area down to an economical value.

A formula developed by Bradford and Clarkson is

$$H = \text{b. hp.} \times C \times \frac{D}{4}$$

where

H = heat recoverable, B.t.u., per hour.
b. hp. = brake horsepower.
C = a constant; 12 for four-cycle and 20 for two-cycle engines.
D = gas-temperature drop through the boiler, degrees Fahrenheit.

This is reasonably accurate. However, it is based on approximately 30 lb. of air per pound of fuel for a four-cycle and 50 lb. per pound of fuel for a two-cycle, assuming 0.4 lb. of fuel per

Table IX.—Test on Clarkson Waste-heat Boiler at Cushing, Okla. (Four-cycle, eight-cylinder, 17½ by 25-in., 225-r.p.m., solid-injection, Alco Diesel)

Engine Load	½	¾	4/4
Hp. output	375	562.5	750
Lb. fuel per hp.-hr. (19,270 B.t.u.)	0.400	0.353	0.349
B.t.u. input to engine per hp.-hr	7,700	6,800	6,720
Exhaust temp. off engine, deg	395	470	640
Steam produced per hr., lb	23	220	226
Steam produced per hp.-hr., lb	0.061	0.392	0.302
Steam pressure, lb. per sq. in. gage	2	4	7.8
Feed-water temp. at boiler inlet, °F	98	93	92
Heat added per lb. steam, B.t.u	1,089	1,096	1,099
Heat recovered per hp.-hr., B.t.u	115	430	332

brake horsepower per hour. These values do not apply in case of all Diesels and may be regarded as somewhat high for all types.

In Table IX appears the amount of steam obtained from a 750-hp., four-cycle Alco Diesel at various loads. These results

TABLE X.—STEAM GENERATED FROM EXHAUST GASES
(Four-cycle Diesel engines)

Hp.	Lb. steam per hr. full load			Lb. steam per hr. ¾ load			Lb. steam per hr. ½ load			Sq. ft. ht. surface
	5 lb. per sq. in.	10 lb. per sq. in.	15 lb. per sq. in.	5 lb. per sq. in.	10 lb. per sq. in.	15 lb. per sq. in.	5 lb. per sq. in.	10 lb. per sq. in.	15 lb. per sq. in.	
75	77	74	72	60	58	56	44	42	40	72
100	92	89	87	72.5	70	67	53	50.4	48.5	72
200	154	150	147	122	118	115	90	85	81.5	72
300	212	205	199	167	160	156	122	116	111	72
400	327	316	309	257	248	240	188	178	171	144
500	392	380	372	310	298	290	226	215	205	144
600	483	466	455	380	366	354	277	265	253	192
700	550	530	520	432	417	404	317	301	287	192
800	638	617	600	504	481	468	368	348	334	240
900	700	675	664	550	530	515	414	383	366	240
1,000	780	750	735	612	588	572	450	425	416	288

Two-cycle Diesel engines

Hp.	Lb. steam per hr. full load			Lb. steam per hr. ¾ load			Lb. steam per hr. ½ load			Sq. ft. ht. surface
	5 lb. per sq. in.	10 lb. per sq. in.	15 lb. per sq. in.	5 lb. per sq. in.	10 lb. per sq. in.	15 lb. per sq. in.	5 lb. per sq. in.	10 lb. per sq. in.	15 lb. per sq. in.	
75	63	60	57	40	37	34	17	14	12	72
100	78	74	71	49	45	42	21	17	14	72
200	165	157	150	102	94	88	44	36	30	144
300	229	217	208	144	133	124	60	50	42	144
400	300	285	273	191	176	165	80	66	55	192
500	370	351	336	234	216	202	98	80	68	240
600	450	427	410	282	260	244	118	98	82	288
700	525	498	477	330	305	285	139	115	96	336
800	600	570	545	378	358	320	158	130	109	384
900	675	640	615	425	393	368	178	147	123	432
1,000	750	710	682	475	440	410	198	163	136	480

are less than should be realized ordinarily, due to the rather low exhaust-gas temperature at all loads.

In Table X is given the amount of steam that may be generated from the heat in Diesel engine exhaust, according to Foster Wheeler Corporation.

Air Heating.—When exhaust gases are used to heat air, a formula developed by G. C. Boyer, Burns & McDonnell Engi-

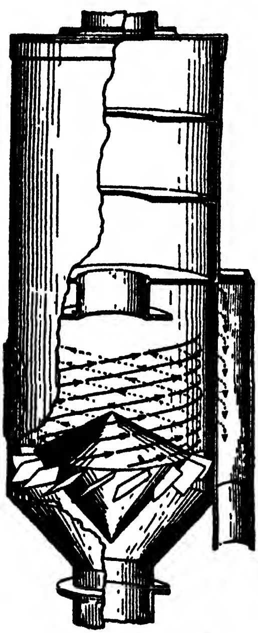


Fig. 203.—Vortex spark-arrester silencer.

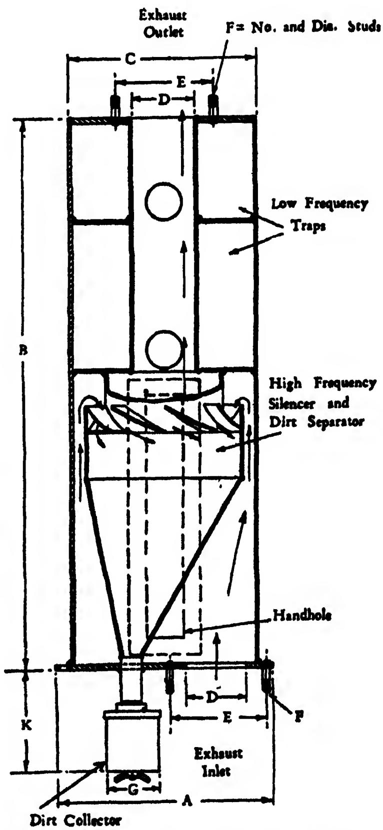


Fig. 204.—Section through Maxim spark-arrester silencer.

neering Company, is

$$0.30\,H = 0.01659V(120 - t)$$

where

H = total heat, B.t.u. passing through exhaust silencer per minute.

V = cubic feet heating air circulated around silencer per minute.

t = inlet temperature of heating air.

Silencers.—The deadening of exhaust noises is a science and can be handled only by those trained in the art.

The silencers now on the market may be separated into four classes: (1) absorption type, (2) baffle type, (3) internal-friction type, and (4) expansion type.

In the absorption type the silencer consists of a perforated tube surrounded by an outer casing, with the space between filled with an absorbent material. The idea is to have the peak of the pressure wave absorbed by the material and in this way level the wave and kill the sound.

In the baffle type the sound wave is suppressed by passing around various forms of baffling. Type 3 depends upon internal friction and consequently creates considerable back pressure.

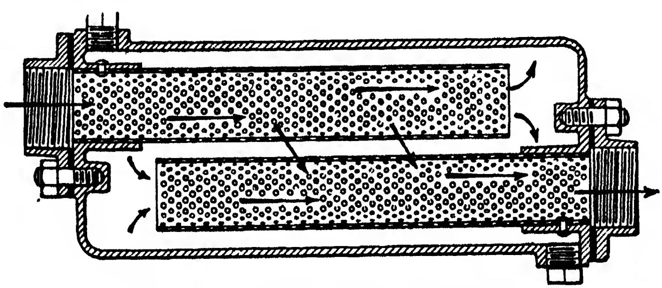


FIG. 205.—Burgess spark-arrester silencer.

In the expansion type 4, water is fed into a chamber and cools the gases, and thereby reduces the pressure wave.

The Vortex spark-arrester silencer is shown in Fig. 203. As will be observed, the gases entering at the base are given a circular motion by a series of vanes, and the centrifugal force thus given the dirt particles and the sparks cause these to be thrown through a series of small openings into the drain line. The clean gases finally pass through the control opening in a baffle and expand in a series of reactance chambers before entering into a stack above the silencer.

A Model SC2 Maxim spark-arrester silencer is shown in Fig. 204. The direction of gas flow is shown by the arrows, as are the vanes which cause the sparks and dirt to be thrown down into the dirt collector.

A Burgess spark-arrester silencer is shown in Fig. 205. The sparks and carbon are thrown out by their own inertia.

CHAPTER XV

CYLINDER PRESSURES, COMBUSTION ACTION, INDICATOR DIAGRAMS, INDICATED HORSEPOWER, ENGINE PERFORMANCE

General.—Before taking up the important subjects of fuel-injection designs and combustion principles found in the modern Diesels, it is advisable to outline the method by which one may ascertain the gas pressure created by combustion of fuel in an engine cylinder and may study the action of these combustion gases during the entire engine cycle. The method is based on the cylinder-pressure indicator.

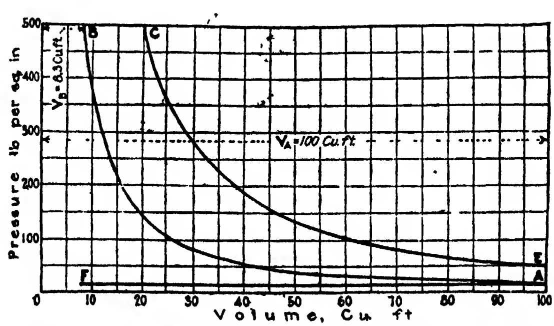


FIG. 206.—Diagram showing cylinder pressures at all points in the piston stroke.

The value of an indicator in locating engine faults is not understood by the average Diesel engineer. For this reason the process of indicating the engine will be covered in some detail.

If a sheet of cross-section paper were laid off so that horizontal measurements represented to some scale the distance traveled by the piston of an engine from the beginning of the stroke, and vertical measurements represented to scale the pressure existing within the cylinder at the various points in the strokes making up a complete cycle, a diagram similar to Fig. 206 would be developed.

The several events on the diagram may be listed as follows:

Suction Line.—Fresh air is drawn into the cylinder by the suction of the piston, filling the space vacated by the retreating piston (line *FA*).

Compression Line.—The reversal of the piston and its movement toward the cylinder head compresses the air trapped in the cylinder until, at the end of the stroke, the compression pressure is around 480 to 500 lb. gage (line *AB*).

Combustion.— The fuel, or spray, valve opens, allowing high-pressure injection air or a pump to force the fuel charge into the cylinder. The rate of fuel admission is such that no increase in pressure occurs, the heat added being sufficient to keep the pressure practically constant (line *BC*).

Expansion.—The fuel valve closes, and, the combustion being complete, the hot gases expand, forcing the piston to the end of the stroke (line *CE*).

Exhaust.—At or near the end of the stroke the exhaust valve opens, and the hot gases flow out of the cylinder (line *EA*). The cylinder pressure drops to atmospheric, and on the next stroke the piston forces the remainder of the gases out of the cylinder (line *AF*).

If we could have such a diagram, giving the pressure in the cylinder at every point, it would show the freedom of the intake, the work lost by suction, the extent and perfection of the compression, the point at which ignition takes place, the pressure realized by combustion or explosion and the rapidity of the process, and the manner in which the heated gases expand; also the power that the engine is developing, as well as defects in the mechanical details of the engine.

The Indicator.—James Watt, inventor of the steam engine, developed the cylinder indicator to ascertain the power that an engine actually developed, that is, the horsepower usefully employed plus the horsepower consumed in friction.

As applied to a Diesel cylinder, the cylinder indicator (Fig. 207) might be represented as consisting of a board sliding in a frame *B* and connected to the engine piston by a cord *O*. The board is covered by a sheet of paper. A cylinder *D* is attached to the engine cylinder and carries a piston *P* to which is attached a rod *R* and a spring *S*. The board moves back and forth with the travel of the engine piston. The gases from the engine cylinder enter the indicator cylinder *D* with a pressure equal to the pressure within the engine cylinder, and the piston P is pushed upward until the compression of the spring S balances the cylinder pressure. If the engine cylinder pressure drops, the pressure

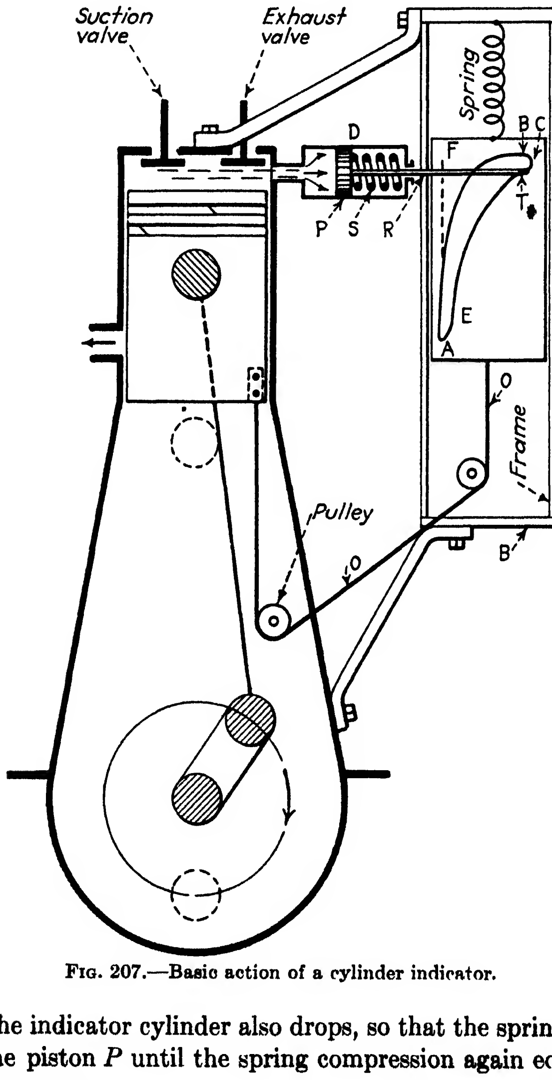


FIG. 207.—Basic action of a cylinder indicator.

inside the indicator cylinder also drops, so that the spring pushes back the piston P until the spring compression again equals the cylinder pressure.

Recapitulating, we have, then, the board moving back and forth in unison with the Diesel's piston travel, while the indicator piston moves up and down in conjunction with the change in the pressure inside the Diesel's cylinder.

The pencil end T of the piston rod R then makes a mark on the card, and the height of the mark at any point in the engine's piston stroke is proportional to the cylinder pressure.

If the indicator spring S is so made that it will be compressed 1 in. by the application of 100 lb. per square inch pressure, then, when the cylinder pressure is 100 lb., the pencil mark at this point on the travel of the indicator board will be 1 in. above the zero pressure, or atmospheric pressure (line AF).

Applying this indicator to a four-cycle Diesel, as in Fig. 207, let us observe the marks made by the indicator pencil, assuming that the indicator spring S has a scale of 1 in. to 200 lb.; that is, the application of 200 lb. compresses the spring 1 in.

At the start of the suction stroke, the pencil will rest at the point F, indicating that the cylinder pressure is equal to the atmospheric pressure, 14.7 lb. absolute. When the engine piston moves down on the suction stroke, the pencil makes the line FA.

At the start of the compression stroke the pressure within the engine cylinder is atmospheric, that is, zero gage, so the indicator pencil rests on the "atmospheric line" AF at the point A. Here the diagram is much reduced, so the pressures cannot be measured by a scale.

The engine piston next starts to move toward the cylinder head, compressing the air. The indicator board of Fig. 207, of course, moves in correspondence. Compression of the cylinder air raises its pressure, and this forces the indicator piston P to move against the spring resistance. The result is that during the engine compression stroke the pencil makes a curve AB which rises as the piston moves along its stroke, appearing as in Fig. 206. This is the "compression curve." At the end of the stroke the pencil will have raised the mark to 1 in. from the zero line AF, representing 500 lb. pressure within the engine cylinder. The engine piston reverses and starts along the power stroke while oil is sprayed in. Combustion of this fuel and air holds the cylinder pressure at, say, 500 lb. for about 10 per cent of the power stroke.

Fuel introduction ceases, and during the remainder of the stroke the gases expand behind the moving piston, with the cylinder gas pressure falling. The indicator pencil then describes the curve CE on the card in Fig. 207; at E the exhaust valve in the cylinder head opens, allowing much of the cylinder gases to blow out from the cylinder. As a consequence, the cylinder pressure drops, as indicated by the line *EA*.

The piston reverses its travel and starts up on its "exhaust stroke," pushing the remainder of the burned gases from the cylinder. In a perfect engine the cylinder pressure during this exhaust stroke would be equal to atmospheric, so the exhaust line *AF* on the diagram would coincide with the original suction-stroke line *FA*.

Two-cycle Diagram.—An indicator diagram from a two-cycle Diesel does not show exhaust line *AF*, because all the exhaust action occurs from point *E* to point *A*; nor does the suction line *FA* appear, because the two-cycle Diesel has no suction stroke.

Although this indicator, with all its faults of great weight, etc., was fairly satisfactory for steam engines making 30 strokes or less per minute, it was of little use when engine speeds were increased, since the length of spring and weight of the piston were such that at the instant steam was admitted to the engine, the inertia of the indicator parts tended to cause the indicator piston to travel higher than it should. The Watt indicator was improved and developed with the idea of reducing the inertia and errors; the first step was to convert the diagram board into a drum. One of these early indicators was that of McNaught, where the paper was placed on a light drum which was rotated by a cord from the engine crosshead. The indicator did not become a serviceable instrument until Charles B. Richards, of Providence, R. I., brought out the Richards indicator. This instrument was fitted with a strong spring giving a very small travel to the piston, which eliminated the inertia of the long springs used before.

Modern Indicators.—Modern indicators followed, in general, the design brought out by Richards and until the last few years were all made with an inside spring, as shown in Fig. 208. Here the working barrel *A* is surrounded by the housing *C*, the space between the two acting as a gas jacket, with the idea of avoiding distortions of the barrel from uneven temperatures. The lower end of the barrel is free to accommodate itself to change due to varying steam temperatures. The piston *B* works inside the barrel and is provided with a spring, the resistance of which causes the piston travel to be proportionate to the gas pressure under the piston. The upper end of the spring is attached to the indicator cap. The piston is provided with a rod which carries at its upper end the swivel head F to which is attached the pencil linkage as shown. The drum, about which is wrapped the paper card, is carried on an arm projecting from the indicator housing. The cord G is attached to the engine-reducing motion, and its pull is opposed by the tension of the drum spring. This arrangement insures the cord being held taut at all times; and when the cord on the out stroke of the engine rotates the drum,

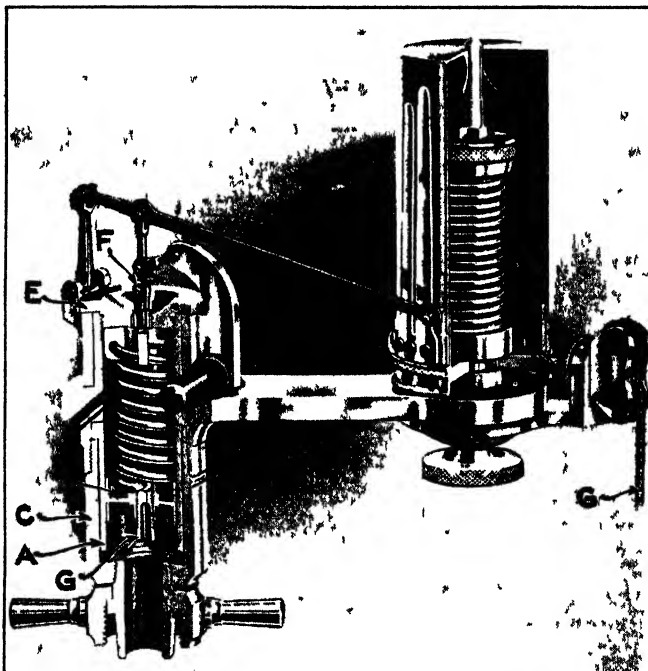


FIG. 208.—Typical inside-spring indicator. (*Crosby*)

the spring tension brings the drum back to its original position on the return stroke of the engine.

As long as pressures and temperatures were not high, the inside type of indicator proved to be perfectly satisfactory. Under modern gas- and oil-engine conditions, high temperatures altered the characteristics of the spring, giving unreliable diagrams. The spring was placed on top of the indicator, away from danger of excessive temperature changes, and the majority of instruments manufactured in recent years are of this type (Fig. 209). Owing to the high pressures in the heavy oil engine, the steam-engine-indicator piston, which has a ½-sq in. area, would require an extremely heavy indicator spring It is usual to fit the oil-engine indicator with a piston of ¼-sq. in. area. In this way the 200-lb. steam-engine spring may be used and will then allow 1 in. of the travel of the pencil when a pressure of 400 lb per square inch is exerted in the engine cylinder.

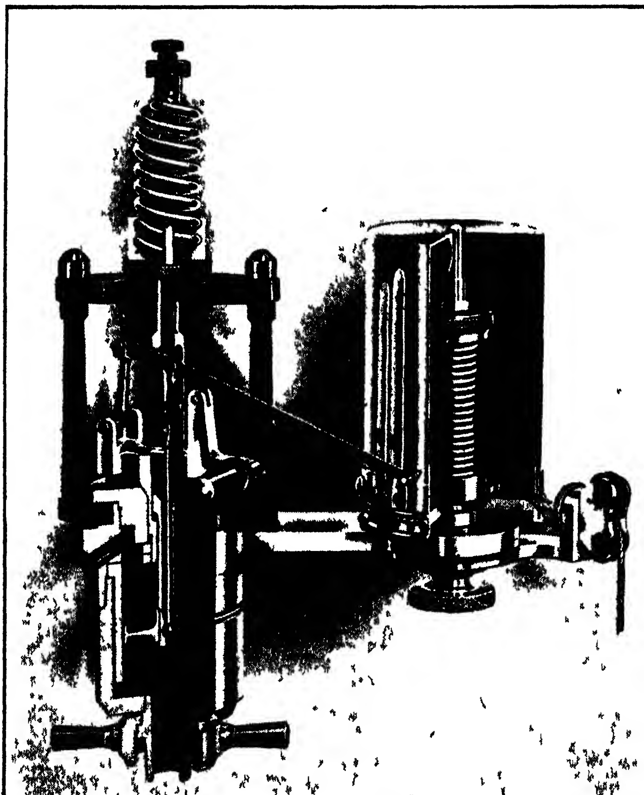


FIG. 209.—Crosby outside-spring indicator.

Hopkinson Optical Indicator.—To permit securing diagrams from engines operating at high speed, various optical indicators have been designed. The Hopkinson indicator (Fig. 210) has been more largely used than any other of this class. A barrel *A* contains a piston *F* which is held by a shackle *G* to the flat spring *D*. The spring is supported upon two arms by the screws *E*.

A mirror *H* is pivoted to the shaft *I*, which is connected by the arm *M* and the stirrup *L* to the spring. A slight deflection of the spring causes the mirror to rotate a slight amount on its axis. If a ray of light is allowed to strike the mirror, the partial rotation of the mirror will cause the image of the light to move in phase with the indicator piston, but the ratio of travel is large. The

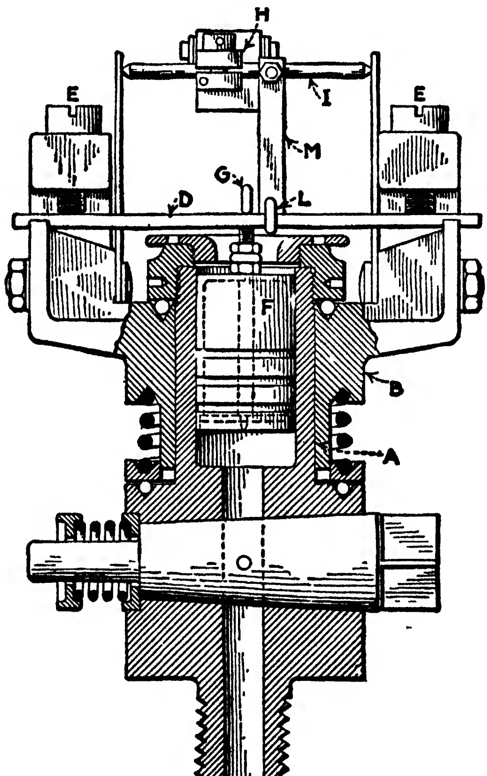


FIG. 210.—Hopkinson optical indicator.

mirror and spring are mounted on a collar *B* which is rotated by the indicator cord. The two motions of the mirror cause the image of light to trace on a plane surface a diagram representing the pressure-volume conditions within the engine cylinder. If a sensitive plate is used, a photographic diagram results.

Midgley Indicator.—A recent optical indicator designed for gas-engine work, but equally suitable for oil-engine testing, is the Midgley, illustrated in Fig. 211. In this instrument the indicator proper is fitted with a piston and a special spring. The piston rod is linked to a mirror at its top. Light rays from an electric bulb are thrown upon this mirror, which is shifted on a horizontal axis by the movement of the indicator piston. The rays are then reflected back upon an eight-sided mirror and thence upon the ground-glass screen, rising and falling with the change in pressure. The eight-sided mirror is driven from the engine in

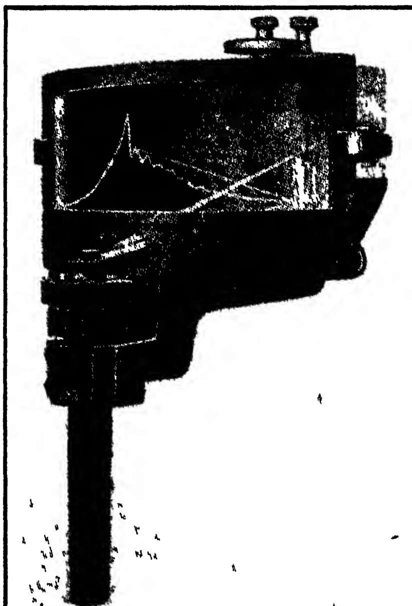


FIG. 211.—Midgley high-speed optical indicator

phase with the engine piston. This gives a horizontal movement to the ray of light. A diagram showing the cylinder events is then made on a sensitive plate or is observed by the reflection on the ground glass.

The indicators shown here do not constitute all the instruments now on the market; several are available that are more suitable for high-speed engines than are those shown, owing to the low inertia effects of these lightweight instruments.

Indicator Rigs.—The majority of Diesel engines of 700 hp. and under have trunk pistons. For this reason the indicator riggings usually employed on steam engines cannot be used on the Diesel, and even the large crosshead-type engine requires some special rigging. In Fig. 212 is shown a type of rigging often used on trunk pistons. The lever *B* is fastened to the inner wall of the piston by a screwed pin. The rocker *A* is fulcrumed on a pin *C* at the side of the crankcase door and is fastened to lever *B* by a bolt-and-kuckle joint. The indicator cord is looped over the hooked end *D* of the rocker *A*. There is a slight error in the diagram resulting from this rigging. If, however, the lever *A*

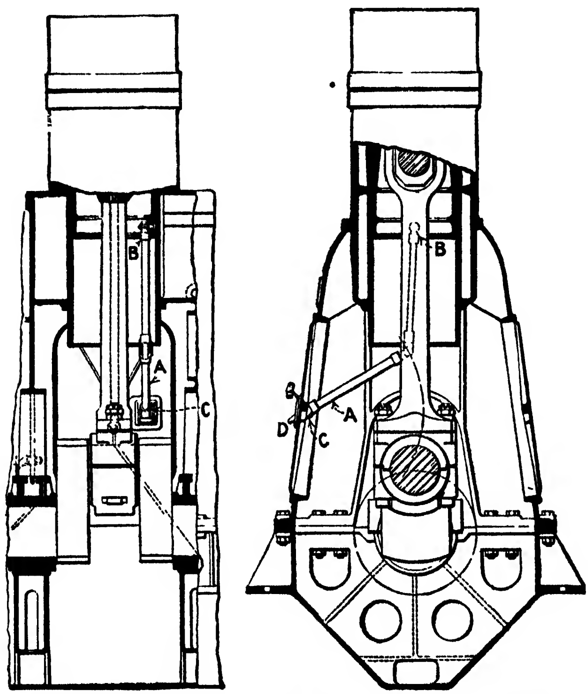


FIG. 212.—Indicator rigging for trunk-piston engines.

is made of such a length that the knuckle joining *A* to *B* moves equal distances on the two sides of the cylinder center line, the error is negligible. This rigging is easy to make if the engine builder has not supplied some arrangement.

With this rigging, when the piston moves upward the indicator-drum spring tension is increased. If the cord has a tendency to stretch, the diagrams may be distorted toward the end of the compression stroke and during the combustion period. Errors at this point are of serious moment, since the most important events in the engine cycle take place along this portion of the diagram.

Figure 213 outlines an arrangement that is quite satisfactory for crosshead-type engines. By reason of the distance the crosshead is from the face of the frame, it is seldom possible to link the rocker arm direct to the crosshead. Usually, a stud of some length carrying the lever at the outer end must be screwed into the crosshead.

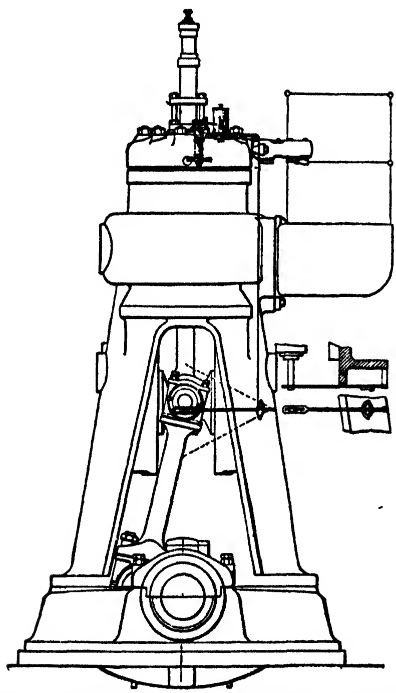


Fig. 213.—Rigging used on crosshead-type engines

It will be noticed that the diagrams are accurate if the quadrant over which the cord passes is correctly made. The cord connection is such that, as the piston moves downward, the indicator cord is drawn out against the spring tension. If the cord stretches, distortions of the diagram will occur toward the end of the expansion stroke, but this is not so serious as an equal amount of stretching at the other end of the diagram.

Many of the engines now in use were not fitted with indicator rigging, and the engineer must supply the deficiency. Either of the two arrangements discussed may be used. Simpler and more easily applied devices are often made by the engineer. In Fig. 214 appears a device that is easily made and gives a correct diagram. This consists of an eccentric B clamped to the engine shaft with the eccentric throw SA in line with the engine crank. In constructing this motion the relation of the radius of the eccentric B plus the radius of the roller C (the distance AC) to the

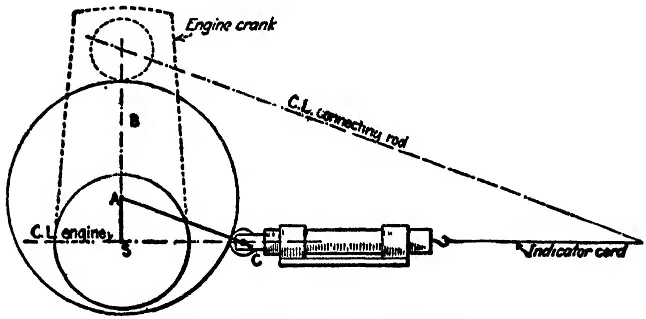


Fig. 214.—Easily constructed rigging.

engine connecting rod is equal to the ratio of the eccentric throw to the engine-crank throw, or

$$\frac{AC}{\text{Connecting rod}} = \frac{AS}{\text{crank throw}}$$

Any engineer can make this, using a piece of steel rod for the slide and a piece of pipe with a babbitt liner for the guides. The eccentric can be made of cast iron, split and drilled for clamp bolts. In an emergency a hardwood eccentric with a brass strip on the edge may be used.

Instead of an eccentric, the end of the engine shaft may be drilled for a small cap screw. A connecting rod or steel bar of a length such that the distance from the engine shaft S to the cap screw A bears the same ratio to the crank throw that the engine connecting rod bears to the rod AC will give a miniature connecting rod AC and crank SA which reproduce to a smaller scale the action of the engine crank and rod.

Figure 215 outlines the rig employed on one two-stroke-cycle, solid-injection engine. A link pinned to the engine crosshead transmits the piston motion to the lever A, which, through a short arm B, a vertical rod C, a bell-crank D, and a shaft E, pulls the indicator drum cord. This in appearance is very neat, but after a short time, if the parts are not kept snug, the slight wear at the joints causes the indicator diagram to be unreliable.

The Air-cycle Diagram.—In the construction of Fig. 206 the assumption was made that nothing but pure air exists in the cylinder and that the heat is added to the air, there being no gaseous products of combustion. It was further assumed that no heat transfer takes place between the air in the cylinder and the

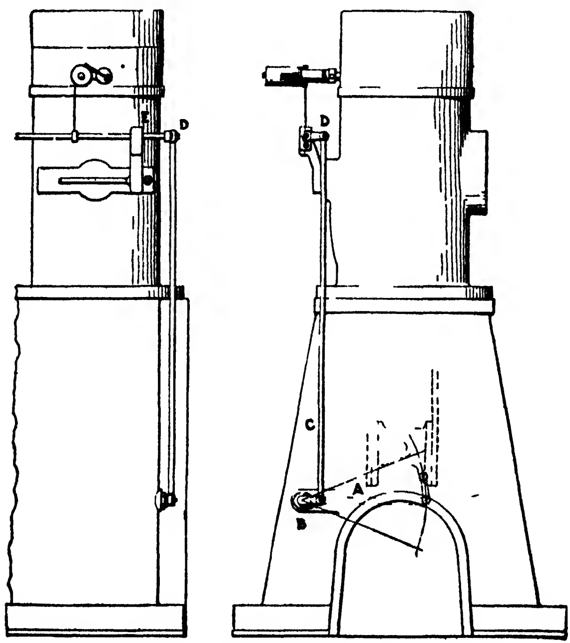


FIG. 215.—Another type of indicator rigging

cylinder walls; that is, both expansion and compression are adiabatic. The exponent in the equation for adiabatic expansion and compression $P_1V_1^n = P_2V_2^n$ was taken as 1.408 for air.

Actual Diesel Diagram.—For several reasons this ideal diagram is never obtained from a working engine. In the first place the cylinder charge of air loses more or less heat to the cylinder walls during the compression period. This causes the compression curve to fall below the adiabatic curve for air. In addition, during the expansion stroke the hot cylinder walls give off heat to the expanding gases. The actual expansion line is above the theoretical expansion line on the air-cycle diagram (Fig. 206). The specific heat of the gases does not remain constant during the expansion stroke, and the fuel is not completely burned when the fuel valve closes at C but continues to burn through a considerable portion of the expansion stroke. This also changes the slope of the expansion curve.

It is generally assumed that the exponent of the compression and expansion equation PV^n = constant is 1.35 instead of 1.408 for air. As a result of the heat loss during compression the final clearance volume of the Diesel cylinder must be less than the air-cycle clearance to attain the same final pressure. If it is assumed

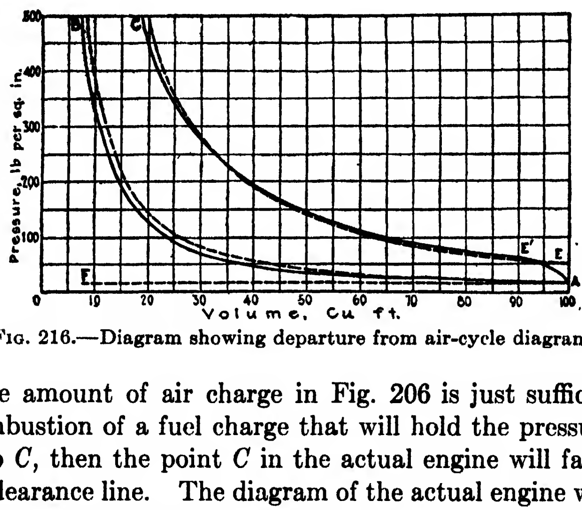


FIG. 216.—Diagram showing departure from air-cycle diagram.

that the amount of air charge in Fig. 206 is just sufficient for the combustion of a fuel charge that will hold the pressure constant to C, then the point C in the actual engine will fall closer to the clearance line. The diagram of the actual engine will then be similar to the full-line diagram in Fig. 216, the ideal diagram being shown in broken lines. The expansion in the working engine is not carried to the end of the stroke, but release occurs at E^1, this being when the crank is at an angle of 30 to 50 deg. from dead center, the piston having completed 80 to 90 per cent of its stroke. The exhaust and suction lines are not shown, as the departure from the ideal line FA will be but slight.

In addition it will be found that the diagram taken from the working engine will show other departures from the ideal diagram. The cylinder pressure during the combustion period BC is never constant; the line BC then departs somewhat from the horizontal and in some solid-injection Diesels shows a decided pressure rise. It is possible, however, by careful regulation of the injection-air pressure and the number of atomizer disks in the fuel valve to maintain constant pressure for a portion of the combustion period, as illustrated in Fig. 217. In Fig. 218 is shown a diagram where the pressure during combustion shows an increase. This may be traced to the few atomizing disks in the fuel valve together with a high injection pressure. To correct this condition the air pressure should be decreased, and one or two

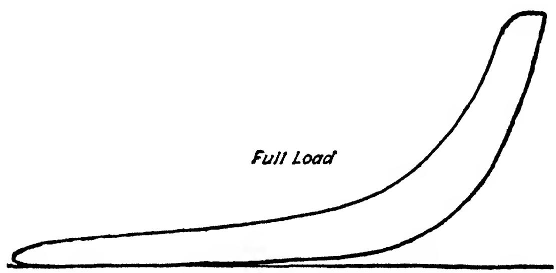


Fig. 217.—Actual diagram from an air-injection Diesel.

more atomizer disks added, or a few of the holes through the present disks closed. This will increase the resistance to the oil flow and so decrease the rate of oil injection.

Variations in the Combustion Line.—Often the diagram of an air-injection Diesel has a combustion line somewhat as in Fig. 219, the pressures falling during the period of fuel injection and combustion. It has been found that the action of the engine is improved, the efficiency is not affected, and there can be a wider

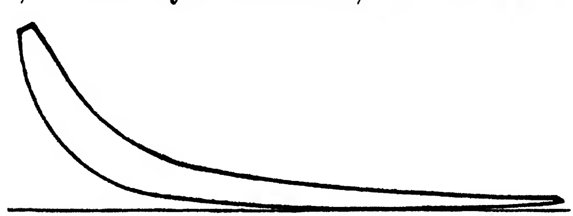


Fig. 218.—Diagram showing high injection-air pressure.

variation of air-injection pressure, oil viscosity, etc., without causing trouble.

If the fuel injection is late, the shape of the combustion line becomes similar to Fig. 220, giving to the diagram the appearance of a sharp peak. If injection is early, the same character of peaked diagram is obtained, and it is often quite impossible to tell by inspection whether the fuel cam is set to give early or late fuel injection. The only way to locate the trouble is to cut out the fuel to the cylinder in question and take a compression diagram (Fig. 221). This diagram shows the compression and reexpansion of the cylinder charge of air. The pressure at the top of the curve is the total compression pressure. Comparing

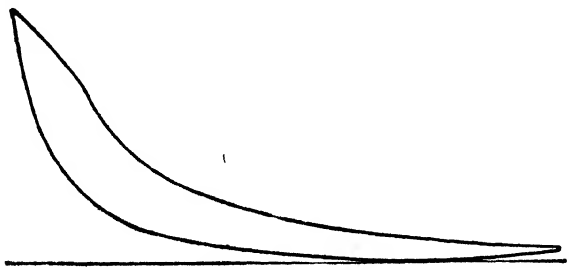


Fig. 219. Usual form of combustion line.

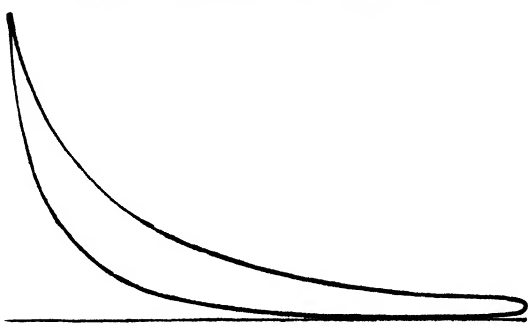


Fig. 220.—Diagram showing late fuel injection.

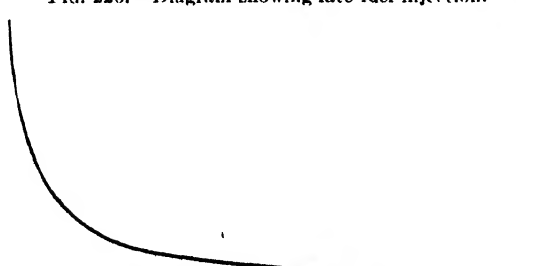


Fig. 221.—Compression line.

the height of this diagram with that of the diagram taken showing the peak will reveal whether or not the pressure at the peak is higher than the compression pressure. If it is higher, then the fuel admission is early. If no increase in pressure is found, the fuel injection must have been late to cause the peak.

Late injection does not always cause a sloping combustion line. At times the diagram appears as shown in Fig. 222, which was obtained from a two-stroke-cycle Diesel. The cylinder pressure upon the reversal of the piston dropped from *b* to *a* by reason of the slow rate of combustion, at which point the increased ignition of the fuel raised the pressure to *c*, practically as high as at *b*.

Suction and Exhaust Events.—Since the spring used in the indicator is at least heavy enough to require 200 lb. per square inch to give a pencil travel of 1 in., the suction and exhaust lines cannot be studied to any advantage. It is frequently desirable to examine these lines to detect any restriction due to long pipe lines or to improper valve setting. If a light spring is

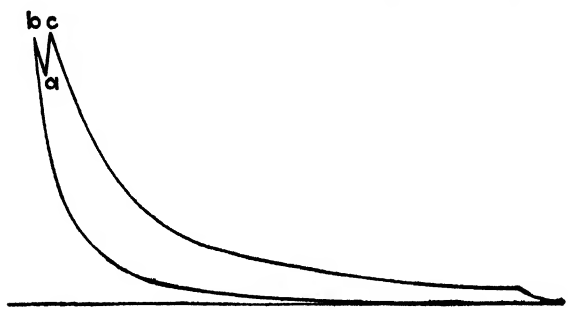


Fig. 222.—Combustion is extremely slow.

used in the indicator, so that the small variations in the suction and exhaust pressures are shown, the high pressures during combustion would cause the light spring to be jammed, probably breaking it. To avoid this, a light brass tube *C* may be slipped over the piston rod before the spring is screwed on, as shown in Fig. 223. The length of the tube should be about one-half that of the spring.

In Fig. 223 at *A* is shown an indicator diagram obtained with this arrangement. Here the suction pressure is shown to be several pounds below the atmosphere line, whereas the exhaust is probably less than 1 lb. above atmospheric. The compression line rises until the spring end touches the brass tube, whereupon no higher pressures are shown, the pencil making a horizontal line to the end of the compression stroke. On the combustion and expansion stroke, the pressure conditions are not shown until the pressure drops to that indicated by the spring when the brass tube stopped the spring compression. At this point the opening of the exhaust valve caused the cylinder pressure to drop sharply, as shown at "release." Only in installations where the exhaust

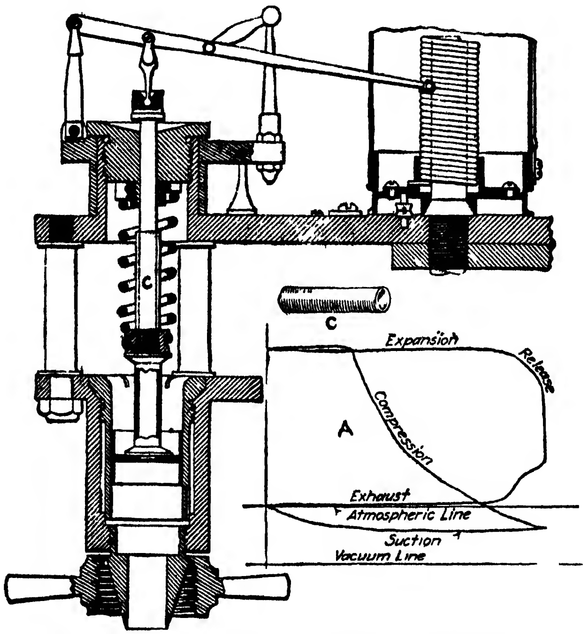


FIG. 223.—Indicator provided with a weak spring.

or suction lines are long or where the exhausts of several cylinders interfere with one another will the pressures during this part of the cycle differ much from atmospheric; a drop of more than 1 lb. should not be allowed to exist.

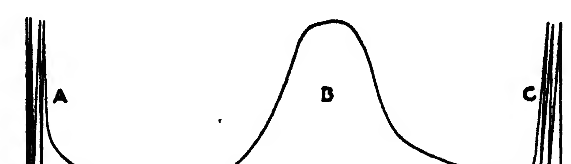


FIG. 224.—Indicator-drum cord pulled by hand.

Offset Diagrams.—It will be noticed that the combustion events are crowded together at one end of the diagram, although during this time the crank passes over a considerable percentage of its circle, about 40 deg. If this end portion of the diagram could be lengthened, it could be studied with better results. To do this, it is customary to connect the indicator rigging so that instead of the indicator drum travel being in phase with the engine piston, it is out of phase by 90 deg. This means that when the engine piston is near the end of the stroke and traveling at its slowest rate, the indicator drum is at the center of its travel and traveling at its highest speed. To secure an out-of-phase dia-

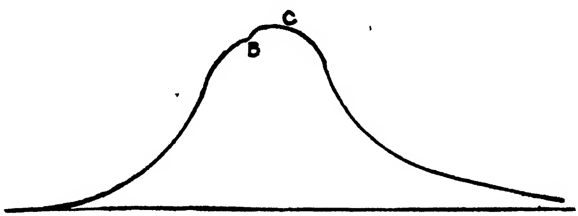


FIG. 225.—Diagram shows normal combustion.

gram, the indicator cord can be attached to another engine piston or crosshead in place of the piston of the cylinder being indicated.

It is not absolutely necessary to so connect the indicator cord. Instead, the cord may be pulled by hand, for all that is desired is a picture of the combustion events. After a few trials the engineer can easily pull the drum cord rapidly when the piston is firing, even though the speed is as high as 300 r.p.m. The diagram thus obtained, if no fuel is being injected, will be similar to Fig.

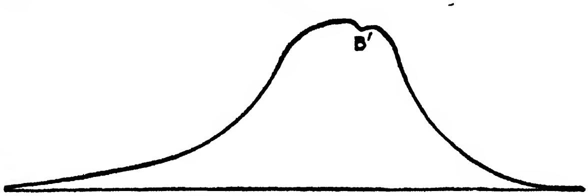


FIG. 226.—Fuel injection is late.

224. In the ordinary diagram there would be a sharp peak at the end of compression; with this out-of-phase diagram the top is rounded.

When the engine is firing, the rounded top becomes wider, and variation on pressure is made apparent. In Fig. 225 the fuel was ignited at *B* with the piston at dead center and continued to *C*. This is the normal diagram, but in Fig. 226 the fuel did not ignite until B^1, showing late fuel ignition, whereas in Fig. 227 the fuel ignited early at *B*. These distorted diagrams will reveal combustion defects and may be taken by connecting the indicator to the cylinder head without the necessity of shutting down to hook up an indicator rigging.

Indicator Diagrams from Solid-injection Diesels.—In ordinary circumstances a solid-injection Diesel gives an indicator diagram

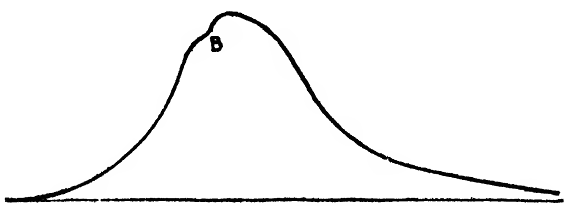


FIG. 227.—Effect of early injection.

almost identical with those from an air-injection Diesel. The chief difference is that the combustion line does not show a constant pressure; instead, the combustion of the fuel causes the cylinder pressure to rise above the final compression pressure. The reason is that the fuel introduction is not so controlled as with the air-injection engine, and so combustion is in the nature

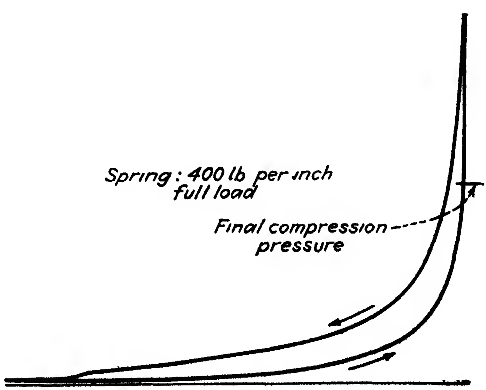


FIG. 228.—Indicator diagram from high-speed, mechanical-injection Diesel.

of a rapid explosion. In certain high-speed engines the peak pressure goes as high as 1,000 lb. per square inch.

In Fig. 228 the peak pressure is 840 lb.; the diagram was taken from an 8 by 10-in., two-cycle cylinder employing direct injection of the fuel into the cylinder.

Study of the Diagram.—The expansion and compression lines on the indicator diagram will not follow the adiabatic lines for air. This is due to the fact that the gases in the cylinder during expansion are not pure air but a mixture having a variable specific heat. In addition, there is a loss of heat to the cylinder walls during the early part of the stroke and an absorption of heat from the walls during the latter part. These combine to cause the expansion curve to follow the law PV^n = constant where n ranges from 1.25 to 1.40. On the compression stroke the air being compressed loses heat to the walls, causing the exponent n to be around 1.35, but in certain engines it may even exceed 1.41.

Regardless of what the exponent n for the expansion may be, it should be the same during the entire expansion period. If n has

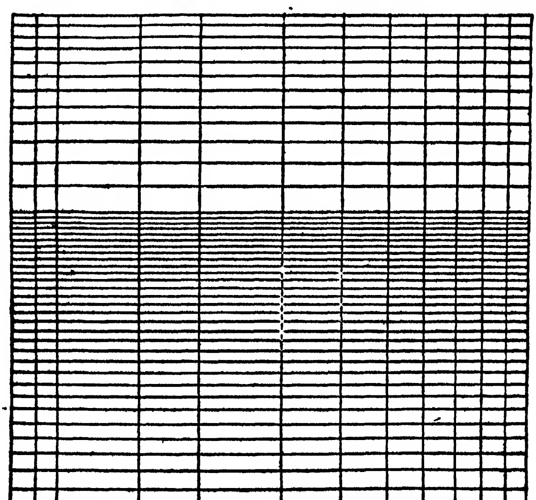


Fig. 229.—Logarithmic paper.

a lower value, say 1.25, during the early part of the expansion followed by a steeper portion of the curve with n equal to 1.3 to 1.35, it will be found, in most instances, that combustion was not completed during the period of fuel injection and "after-burning" occurred during the expansion.

Use of Logarithmic Chart.—It is quite impossible to detect a change in the curve as shown on the indicator diagram. If the diagram is transferred to logarithmic paper, however, after-burning can be detected. This is due to the fact that on logarithmic paper a curve PV^n = constant becomes a straight line, and if n varies then the line is no longer straight but is curved.

To make the transformation, logarithmic paper (Fig. 229) is purchased at an architect's supply house. If this is not possible, it may be drawn by using the logarithmic scale on a slide rule as shown in Fig. 230. The scale is laid horizontally, and division made on the paper beginning at the 0.8 point as outlined. This point may be called 0.08 or 0.8, as the engine cylinder volume may dictate, followed by 0.09 or 0.9, 0.1 or 1, 0.15 or 1.5, etc. The vertical scale is laid out in a similar manner, beginning at the

FIG. 230.—Making logarithmic paper by using slide rule.

zero point and calling the unit or 1 point 100 lb., the 2 point 200 lb., etc. Crosslines are drawn, resulting in a cross-section chart, as shown in Fig. 229. Purchase paper has the divisions made by starting at 1, then 2, 3, etc., instead of 0.8.

The indicator diagram is now divided into volume and pressure units. It is not necessary to divide the volume dimension into units representing the actual cylinder volume, but the division may be into units of tenths or of parts of one hundred, making the total diagram volume plus the clearance volume equal to unity, or 100. The system of division is shown in Fig. 231.

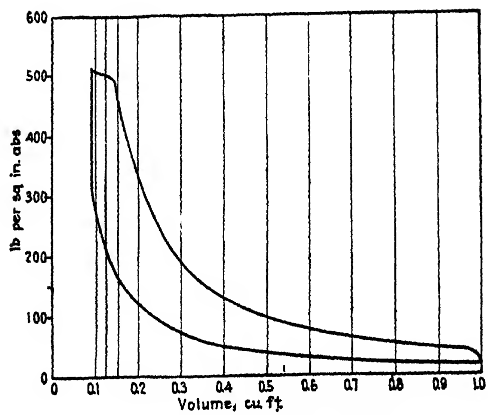


FIG. 231.—Dividing indicator diagram.

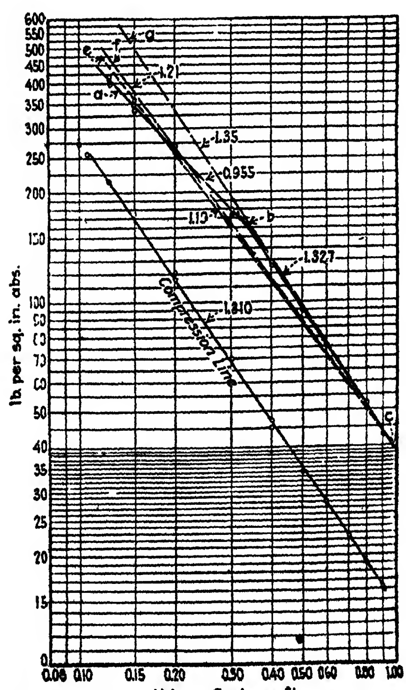


FIG. 232.—Compression and expansion lines on logarithmic paper.

It is necessary to have the clearance volume exact, or the logarithmic diagram will be worthless. In a vertical engine with a smooth clearance volume the linear dimension will represent the volume close enough. In an engine with an irregular cylinder head, as with the Price design, it will be necessary to measure the volume. This may be done by putting the engine on dead center, placing wax about the piston top to prevent leakage along the piston. The combustion chamber is then filled with water out of a weighed bucket, until the chamber is completely filled. The amount of water used is found by reweighing the bucket and finding the difference in weights.

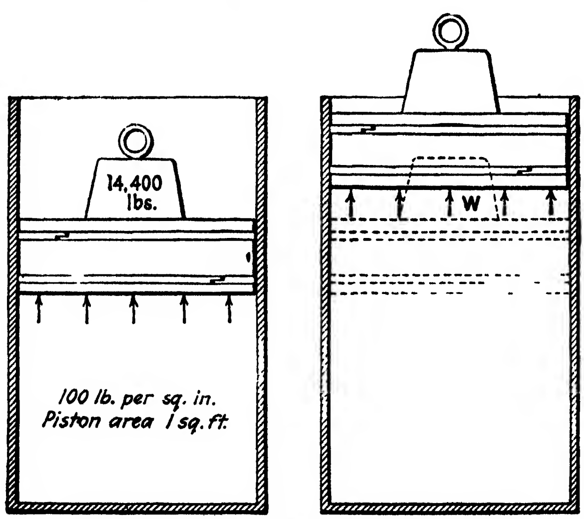


FIG. 233.—Work being done in raising weight.

Upon transferring the diagrams, if the clearance volume has been accurately determined, the compression line should be straight, as in Fig. 232. If the expansion line shows an angle as in line *abc* (Fig. 232), it may be assumed that there is some after-burning. In this chart the expansion lines *ec*, *fc*, and *gc* are fairly straight; incidentally, the diagrams shown were in some instances taken from solid-injection engines, by H. F. Shepherd.

Engineers who find that the exhaust temperatures are running high or that the exhaust valves burn will, upon such transformations to the logarithmic paper, discover that the expansion lines show after-burning. Such indicator work may appear laborious, but every engineer should use an indicator, and he will find the study of the logarithmic diagram of value.

Force.—Force may well be defined as that which either causes motion or tends either to cause or to prevent motion. Suppose we have a cylinder (Fig. 233) filled with a gas at some pressure such as 100 lb. per square inch and the cylinder fitted with a frictionless piston The gas, if not opposed by an external force, would expand and push the piston out of the cylinder. If the cylinder had a cross-sectional area of 1 sq. ft. (144 sq. in.), then to prevent the expansion of this gas we must load the piston with 144 × 100, or 14,400 lb. This weight just equals the pressure exerted against the piston by the gas. The gas pressure

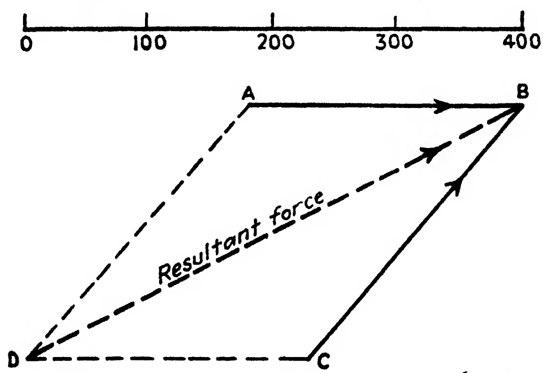


FIG. 234.—Graphical representation of two forces.

is evidently, by definition, a force, since it tends to push the piston; likewise, the attraction of gravity on the weight is also a force, since it tends to push the piston inwardly. These two forms are opposite and equal, and as long as this condition exists the piston will not move.

A force may be defined by (1) its magnitude and (2) the direction of the line along which it acts. These forces may be represented graphically by the length and direction of a line, as in Fig. 234. Here the arrow of the line *AB* indicates the direction in which the force is acting, and the length of the line represents the amount of force, say, to a scale of 100 lb. = ¾ inch.

Two forces acting upon an object may be represented by two such lines, and the combined magnitude of the resultant of the two forces and the direction in which combined forces act may be found graphically by completing a parallelogram of which the two force lines form sides and then drawing the diagonal. The diagonal represents the magnitude and direction of the resultant of the two forces. This is shown in Fig. 234 where the forces AB and BC act on B. The resultant, both in direction and in value, is DB as shown.

Power.—The element of time enters when considering the *power* developed in doing work. Power is, then, the rate of doing work. James Watt, when installing his steam engine to replace horses in the English mines, wanted some scale of comparison of the engine power with that of the draft horses. In order to have an excess of power he rated his engines on the basis of lifting 550 lb. 60 ft. in 1 min., or 33,000 lb. a vertical distance of 1 ft. in 1 min., this being somewhat more than the mine horses could do. He called this rate of doing work 1 "horsepower."

Horsepower.—The horsepower is, then, a rate of doing work equal to the lifting of 33,000 lb. 1 ft. per minute. This is likewise equal to a rate of 550 ft.-lb. per second, or 1,980,000 ft.-lb. per hour. This is the standard unit employed in English computations. When the metric system is used, obviously the unit of power is different.

Since the pressures during the suction and discharge strokes are about atmospheric, the net pressure for these two strokes is practically zero. The net average pressure exerted on the piston is obviously the difference between the average pressure on the power stroke and the average pressure of the compression stroke. If this net average pressure is known, the work done per cycle is the product of this pressure per square inch exerted against the piston times the area of the piston in inches (to obtain the total pressure) times the distance the piston is moved by that pressure. The work done per cycle in foot-pounds is then PLA, where

P = mean effective pressure.
L = stroke, feet.
A = piston area, square inches.

The power developed in one cylinder per minute is the product of the work PLA and the number of times the piston is forced forward on the power stroke, or

$$\text{Power} = \text{work per minute} = PLAN$$

where n is the number of power strokes of the piston; this for a two-stroke-cycle engine is equal to the number of r.p.m., and for a four-stroke-cycle is one-half the r.p.m.

The horsepower is then

$$\text{Hp.} = \frac{PLAN}{33{,}000}$$

Finding Mean Effective Pressure.—The indicator diagram is a closed figure, and its height at any point is proportional to the cylinder pressure at that point in the stroke less the cylinder pressure at the same point in the compression stroke. In almost all four-cycle engines, the pressures on the suction and

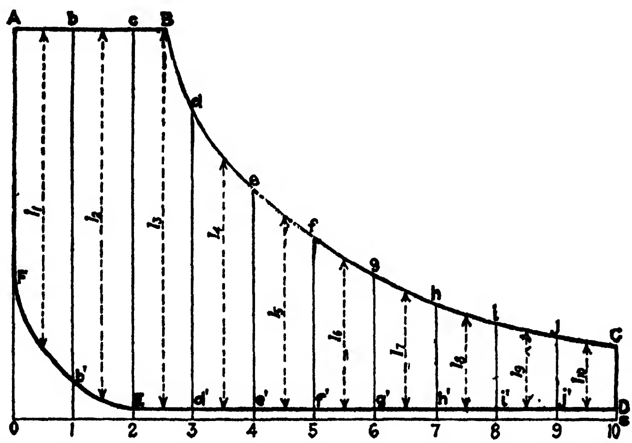


FIG. 235.—Finding mean height of a diagram. (One shown is a steam engine diagram.)

exhaust strokes are so nearly identical that usually the negative area between the two lines is not measured. The length is proportional to the engine stroke; and if the area is known, the average height can be found; this height is proportional to the mean effective pressure. The simplest way to obtain the average height of a closed figure of this kind is to divide its area by its length. If a plane figure is 4 in. long and covers 8 sq. in., it is easy to see that, whatever its shape may be, its average height is $8 \div 4 = 2$ in. This is the way that most indicator diagrams are treated. The diagram area is measured with a planimeter, and by dividing the area by the length the average height is obtained. This height multiplied by the scale of the spring with which the diagram was taken gives the mean effective pressure.

In the absence of a planimeter the mean effective pressure may be measured approximately by dividing the diagram into equal divisions, as in Fig. 235; measuring the height l of each division, and multiplying their average height by the scale of the spring. The more numerous the divisions the more accurate the result. Ten are generally used because it is easy to divide by 10 in getting the average.

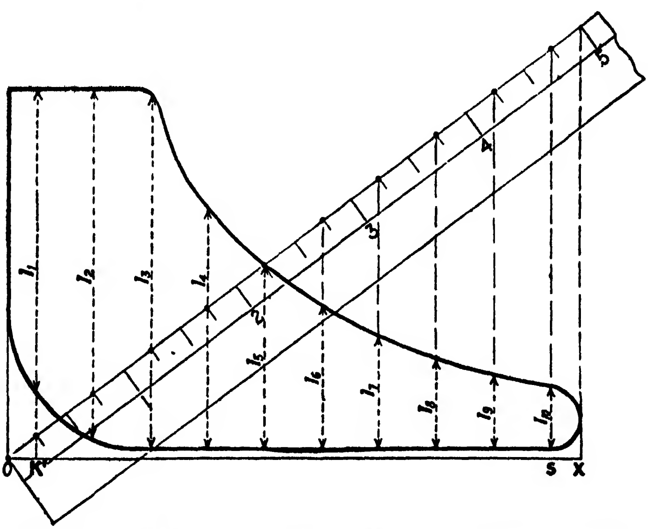


FIG. 236.—Dividing diagram length into sections to find mean height.

It is almost impossible to divide a diagram length directly into 10 even sections, since the diagram is seldom of an even measure, as 3 or 4 in. To find the heights l of the 10 divisions, which are the measurements actually desired, as shown in Fig. 235, rather than the divisions themselves, vertical lines perpendicular to the vacuum line at O and X and touching the ends of the diagram to be measured should be erected. Next, take a 5-in. rule, place it at an angle with OX that allows the zero reading on the scale to be at O and the 5 mark to cut the vertical line drawn through X, the end of the diagram, as shown in Fig. 236. Now draw a line along the rule from O, and, using the rule and a pin, start at the ¼-in. scale reading and make a pin mark; next make one opposite the ¾-in. mark, followed by one at 2¼, at 2¾,

$3\frac{1}{4}$, etc., each $\frac{1}{2}$ in. apart. By drawing through these pin marks lines perpendicular to the vacuum line OX, the diagram length is divided into 11 spaces, the first one OK' and the last one SX being just one-half the length of each of the remaining spaces. The points at which these vertical lines cut the upper and lower lines of the diagram define the limits of the several lengths l_1, l_2, l_3 . . . l_{10}.

To find the average height, the sum of l_1, l_2, l_3 . . . l_{10} is first found; and by dividing by the number of lines (10), the average height is determined. This is proportional to the *mean* effective pressure, the factor being the spring scale. If a 200-lb. spring has been used in the indicator, it is best to use a rule with a 40-lb. scale marked on it for measuring the distances l_1, etc. By then dividing by 55, the mean effective pressure in pounds per square inch is found. If such a rule is not at hand, an ordinary rule may be used and the value obtained in inches multiplied by the spring scale.

A plan followed by many engineers is to take a strip of paper and by laying it along l_1 first mark this length on the paper; then the strip is transferred to l_2, and this added to the length l_1 already marked on the paper strip. By thus adding all the ten lengths, the sum may be measured by a scale, and the error should be less than when the scale is set against each length.

The method here outlined may be placed in the form of a rule as follows:

Divide the diagram length into 11 *divisions, the first and last being one-half each of the other divisions. Erect vertical lines at each division. Add together the distances intercepted on these vertical lines by the top and bottom portions of the indicator diagram. Divide the sum by* 10, *and multiply by the spring scale. Result gives the mean effective pressure.*

Trapezoidal Rule.—*Divide the diagram length into any sufficient number of equal parts. Add half the sum of the two end ordinates to the sum of all the other ordinates and divide by the number of spaces (that is, one less than the number of ordinates). The result is the mean ordinate.*

Simpson's Rule.—*Divide the diagram length into any even number of equal parts at the common distance D apart. Erect ordinates at each point. Add together the first and last ordinate, and call the sum A. Add together the even ordinates, and call their sum B;*

add together the odd ordinates except the first and last, and call this sum C. Then the area of the figure is $\frac{A + 4B + 2C}{3} \times D$, *and the mean effective pressure is* $S\left(\frac{A + 4B + 2C}{3F} \times D\right)$, *when F is the number of parts into which the diagram length was divided and S is the spring scale.*

Actual Mean Effective Pressure.—In the Diesel engine the mean effective pressures range from 60 lb. upward, to as much as 150 lb. per square inch in high-speed, high-powered Diesels. This is the mean effective pressure obtained from the indicator diagram and is generally referred to as the "mean effective pressure referred to indicated horsepower."

A practical method of obtaining the mean effective pressure from an engine not equipped with a reducing motion for the indicator involves the recording of the pressures at the center of the power stroke and of the compression stroke. It has been found that the difference of these two readings gives a close approximation to the actual mean pressure. To obtain these readings, the cylinder should have a hole drilled into the bore at a point where the hole is just exposed by the top of the piston when the piston is in the center position of its stroke. An indicator or a pressure gage attached to the drilled passage will then register the pressure at the mid-point on the power stroke. Cutting out the fuel to the cylinder for one cycle causes the indicator to indicate the middle pressure of the compression stroke.

Brake Mean Effective Pressure.—Knowing the brake horsepower output rating of an engine, the mean effective pressure referred to brake horsepower is found by dividing the product of 33,000 times the brake horsepower by the product of the piston stroke, number of power strokes per minute of all the cylinders, and piston area in square inches, or,

$$\text{Brake m.e.p.} = \frac{\text{b. hp.} \times 33{,}000}{ALN}$$

where

A = piston area, square inches.
L = piston stroke, feet.
N = number of power strokes per minute.

Engine builders show considerable variation in the mean effective pressures used in rating their engines. In general, however, air-injection Diesels are rated at 73 to 78 brake m.e.p. Two-stroke-cycle, mechanical-injection Diesels with a 1:1 scavenging air ratio average 35 to 45 lb. and with a 1½:1 ratio go to 55 lb. Four-stroke-cycle, mechanical-injection engines of the common-rail type use high values of 80 to 95 lb., whereas those with separate combustion chambers average 63 to 67 lb. No general rule can be used in judging engine ratings; one's experience must dictate the mean effective pressure to employ. The character of service required of the engine is of importance, for an engine that is called on only occasionally to give its maximum output may be rated higher than an engine under a constant load.

Rating Diesel Engines.—It has been proposed to rate an engine at the point where its full consumption per indicated horsepower-hour starts to rise; this is at about 0.328 lb. per indicated horsepower-hour. This, however, is not a perfect guide, as many engines have a high overload capacity at this figure. Probably the best guide is the exhaust gas. As long as the exhaust does not change its color, the engine is acting normally and should be rated at, say, 10 per cent below the load at which a perceptible change in color of the exhaust gases is noted.

Air per Horsepower.—The weight of air used by a Diesel engine can be obtained only in a roundabout way, unless complicated air-measuring equipment is used.

The horsepower developed by a Diesel, as in the case of all other engines, can be found by the formula

$$\text{I.hp.} = \frac{PLAN}{33{,}000}$$

when

P = mean indicated pressure, pound per square inch.
L = length of stroke, feet.
A = piston area, square inches.
N = power strokes per minute.

The product of the three factors L, A, and N can be used to find the stroke volume if A is converted into square feet. The volume V swept by the piston per minute = $(LAN)/144$, from which we can write $144V = LAN$.

Inserting this in the horsepower formula and taking the horsepower, as unity, we have

$$1 \text{ i.hp.} = \frac{PLAN}{33{,}000} = \frac{P(144V)}{33{,}000}$$

or

$$VP = 229$$

and if P is taken as 90 lb. per square inch, $V = 2.54$ cu. ft. per minute, or 152 cu. ft. per hour.

If the engine uses 0.36 lb. of oil per indicated horsepower-hour, the volume of air per pound of fuel would be approximately 422 cu. ft. This assumes that the air entering the cylinder is equal to the stroke volume, an assumption that is practically correct.

To find the weight of air, one must know the temperature of the air charge. From examination of exhaust temperatures, apparently on a four-stroke-cycle, air-injection engine the air charge has a temperature of 150°F. after entering the cylinder.

For air, the volume, pressure, and temperature have the relations as expressed by the equation

$$PV = 53.2MT$$

when

M = air weight, pounds.
T = absolute temperature, degrees Fahrenheit = (460 + 150).
V = volume, cubic feet, = 422 in this example.
P = pressure per square foot = 14.7 × 144

Using these volume, pressure, and temperature values, we have

$$144 \times 14.7 \times 422 = 53.2 \times 610M$$

or

$$M = 27.5 \text{ lb. of air per pound of oil}$$

Horsepower at High Altitudes.—At high altitudes engine output is decreased. This drop is due to the fact that air at a high altitude weighs less per cubic foot than at sea level. The actual weight of air taken into the cylinder is then less, and the amount of fuel burned and engine output are decreased.

The Diesel Engine Manufacturers' Association has suggested the curves in Fig. 237. For elevations not exceeding 5,000 ft. above sea level no reduction in engine rating is made. The assumption is made that at such elevations there is ample weight of air to enable the engine to make its rating.

Higher Outputs by Supercharging.—A number of four-cycle Diesels have had exhaust-gas turbines added to drive rotary

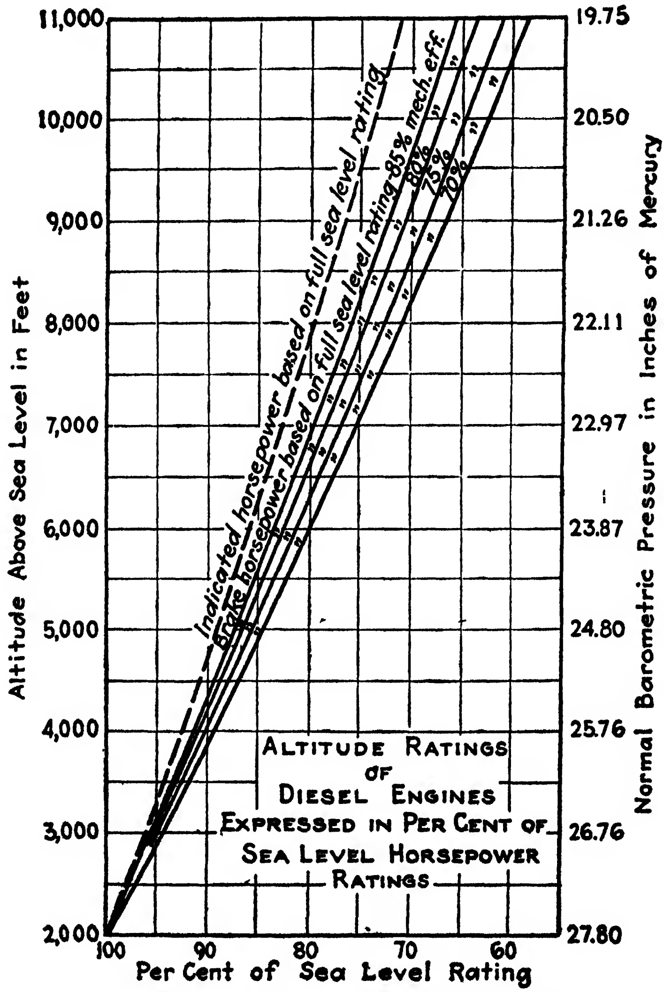


FIG. 237.—Engine rating correction for altitude.

blowers for supercharging purposes, after the Buchi patent. With the increase in output, it is claimed, comes a better fuel consumption than is obtainable with a normal engine. Undoubtedly an engine when supercharged will develop an excess rating, if its clearance volume is properly proportioned. For marine service supercharging is justified on account of reduction in engine weight, but for land service one is safe in remaining at least neutral until more data are produced. It has the advantage of reducing the cost of an engine of a given output, and a few supercharged engines have been installed in America, including several locomotive Diesels and an American Locomotive Company's 1,200-hp. four-cycle Diesel in a New York department store.

Air-cycle Efficiency.—As has been stated earlier in this chapter, theoretically nothing but pure air is compressed in the engine cylinder, to which heat is added; the air then expands and does work. This cycle of alternate compressing, heat, and cooling and air charge is called the "air-cycle."

The efficiency N of the air-cycle Diesel is

$$N = 1 - \frac{1}{r^{Y-1}} \times \frac{R^{Y} - 1}{Y(R - 1)}$$

where

$Y = 1.41$ = ratio of specific heats at constant pressure and constant volume.

R = the cut-off ratio, or ratio of volume at C to volume at B (Fig. 206).

r = compression ratio, or volume at A to volume at B (Fig. 206).

The air-cycle efficiency of a constant-volume, or Otto-cycle, gas engine is

$$N = 1 - \frac{1}{r^{Y-1}}$$

This could be derived from the Diesel cycle by recalling that in the Otto cycle the cutoff ratio R is unity, so that the second part of the expression for the Diesel efficiency disappears. No attempt will be given here to go into the derivation of the various theoretical expressions; these data appear in all textbooks on heat engineering, and those who wish to investigate this phase of the Diesel will have no difficulty in obtaining the information in the sources mentioned.

Error in Air-cycle Efficiency.—The air-cycle efficiency expression assumes that $Y = 1.41$. This causes an error in calculations, for Y varies with changes in pressure and temperatures.

Professor F. O. Ellenwood, Cornell University, has worked up a set of charts showing the actual ideal efficiency for both the Otto and the Diesel cycles. In Fig. 238 are shown sets of curves based on Ellenwood's calculation, using both the lower and the higher heat valves of the fuel. It will be noticed that the

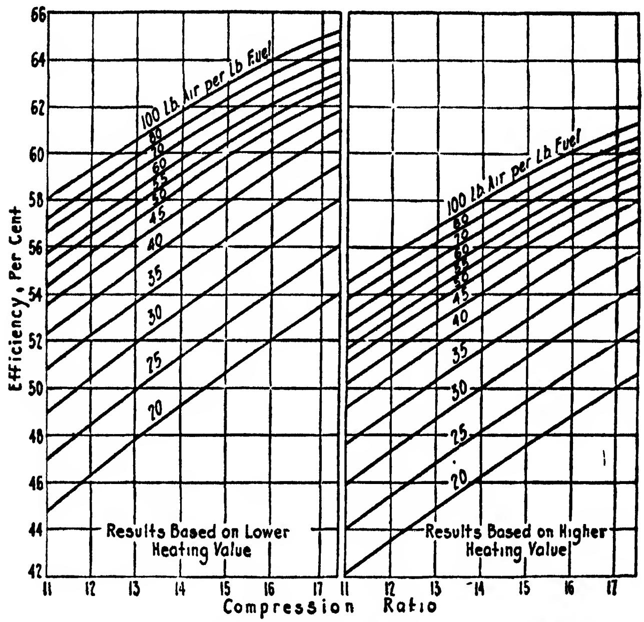


Fig. 238.—Theoretical Diesel efficiencies based on Ellenwood's calculations.

efficiency increases with the air-to-fuel ratio, but in the actual engine the net horsepower output per pound of air will decrease with an increase in the ratio, although the thermal efficiency will increase. This is due to the mechanical losses.

CHAPTER XVI

FUEL-INJECTION AND COMBUSTION SYSTEMS

General.—As has been explained in a previous chapter, fuel and air are combined and burned in a Diesel's cylinder in three general ways. These are classified by the design and action of the combustion cavity and were classified as (1) direct injection, (2) turbulence chamber, and (3) precombustion chamber. To these should be added a fourth (4) air cell, although the air cell is an adjunct to the main combustion-space designs; and a fifth, (5) energy chamber.

Direct Injection.—Direct injection involves the introduction of the fuel into a space formed by the cylinder head and piston crown. It is a logical outgrowth of the air-injection Diesel which has its combustion space identically located. Since such a clearance space has a diameter equal to the cylinder bore, its height is relatively small, for with a compression ratio of R and a piston stroke L, the clearance-volume height C is determined by the expression

$$R = \frac{C + L}{C}$$

If R has a value of 16, a typical compression ratio for high-speed solid-injection Diesels, and the piston stroke is 12 in., then

$$16 = \frac{C + 12}{C}$$

or

$$16C = C + 12, \qquad \text{or} \qquad C = 0.8 \text{ in.}$$

In a 12-in.-stroke Diesel, then, the distance between a flat-top piston and the cylinder head is but 0.8 in.; in other words, the combustion space is a disk of 12-in. diameter and 0.8-in. depth, and the oil must be sprayed into this shallow space in such a way that it meets all the air without impinging too heavily on the piston crown. Even with a piston of 24-in. stroke, the depth of the clearance space is but 1.6 in., by no means excessive if combustion is to be good.

Concave Pistons.—To permit the oil to reach the air without impingement on the piston crown, designers of direct-injection Diesels of moderate bore and stroke make the top of the piston of concave form; in some the curvature is such that a hemi-

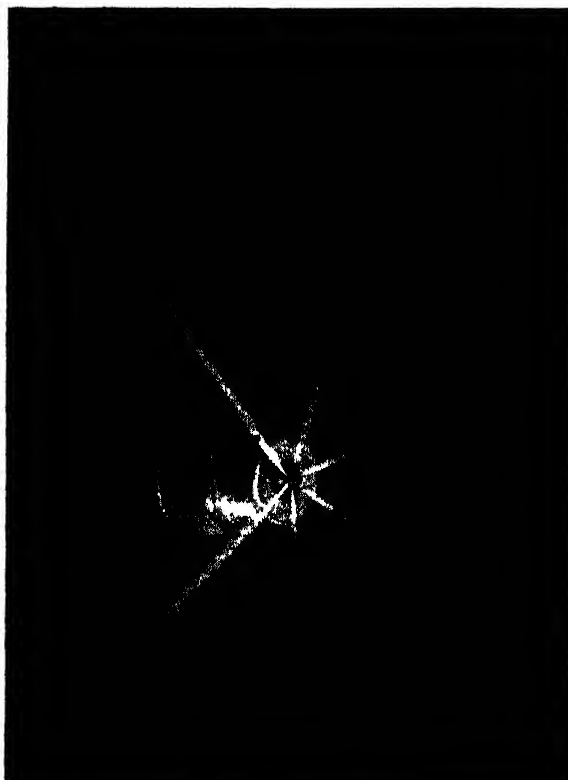


FIG. 239.—Fuel spraying out of a fuel nozzle of a General Motor Diesel.

spherical combustion space is created. With such piston shapes, the piston is allowed to approach the cylinder head so closely that the only clearance is that needed to insure that bumping will not occur.

When the cylinder bore is very small, even a concave piston crown does not give much distance for the oil to penetrate while mixing with the air, so other combustion designs are generally followed. It must be noted, however, that on the small General Motors two-cycle Diesel, direct injection is followed with apparent success.

Fuel-air Mixing.—In direct-injection Diesels, it is impossible to cause the fuel to reach the whole air charge unassisted, for, as will be seen from Fig. 239, which shows oil spraying from a General Motors fuel nozzle, the oil streams follow distinct paths, with much space between each stream.

Complete mixing is brought about by the air sweeping in circular paths around the combustion chamber, and in its travel the air cuts across the oil sprays. To create this circular air flow, the engine design attempts to create an initial circular flow by the shape of the inlet ports, etc. When the air first enters the cylinder, the assumption is that the flow is helical; and while the piston is crowding the air into the combustion space during the compression stroke, the helix is flattened into a true circular flow. In fact, however, the air is in a turbulent disorganized condition, and it is this turbulent air action that causes each oil particle to reach the necessary amount of air.

Turbulence Chamber.—To insure complete mixing of air and fuel and to have a combustion chamber of more or less spherical form, many designers have placed all the engine's clearance volume in a cavity in the cylinder head or at some point along the upper section of the cylinder wall.

The action with such a turbulence chamber is shown in Fig. 240, made up of several sketches of the Hercules engine. In Fig. 240 *A*, the retreating piston induces an air flow into the engine cylinder. At *B* the piston has passed bottom dead center and is shown approaching top dead center on the compression stroke. The entire cylinder charge of air is being forced into the spherical chamber at the right side of the cylinder.

In Fig. 240 *C*, fuel is being injected into the mass of highly heated air through the spray valve *F*. Combustion of the entire oil charge ensues; and when the crank passes top dead center, and the piston starts down on the power stroke, the hot combustion gases flow out into the cylinder space and exert pressure on the piston. As shown in Fig. 240 *E*, the burned gases are expelled through the exhaust valve during the entire exhaust stroke.

The location of the Hercules turbulence chamber is claimed to give especially good results; it is stated that as the piston approaches top dead center, the area of the throat leading to the turbulence chamber is decreased by the overtravel of the piston. In consequence, the velocity of the air is increased, improving the air-oil mixing.

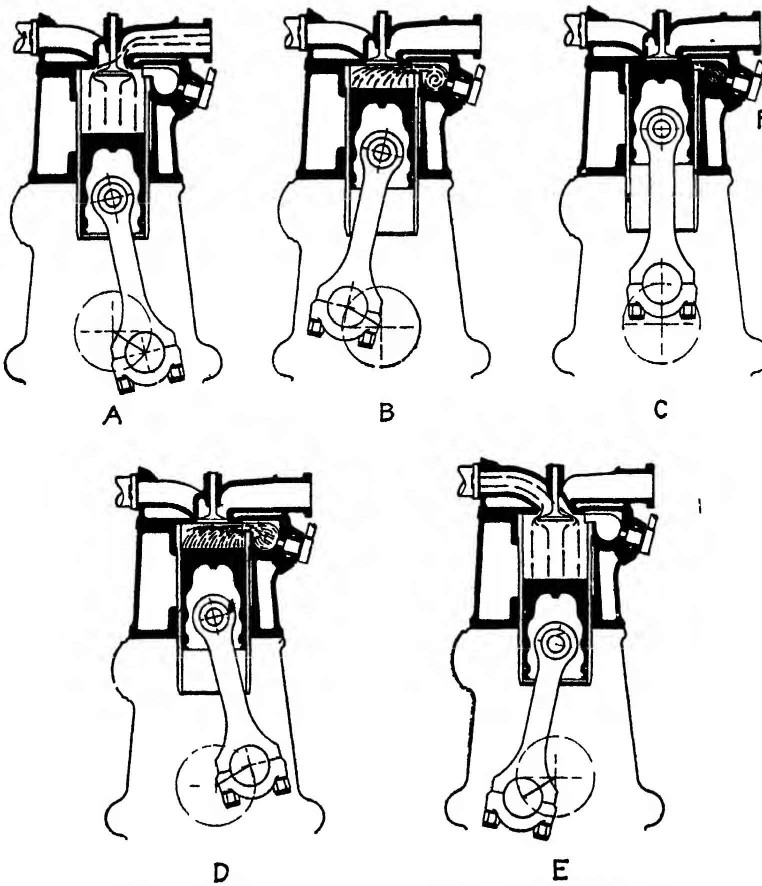


Fig. 240.—Action of turbulence chamber in a Hercules Diesel.

Ricardo Turbulence Chamber.—In Fig. 241 is shown the Ricardo chamber of the Waukesha-Comet Diesel. The general action is the same as in the Hercules. The oil-spray nozzle, however, is placed so that the oil is directed toward the throat, whereas in the Hercules the oil spray is across the air flow.

As will be seen, the turbulence chamber constitutes the entire clearance volume.

To insure that the chamber will be kept hot, it has a lining separated from the water-cooled cylinder head by a minute air space. To start the engine in cold weather, when the air passing through the throat may become chilled, an electric coil is added, as shown at the left side of the chamber.

In fact, all turbulence chambers should have ignition coils for starting, for the throat does cool the air charge.

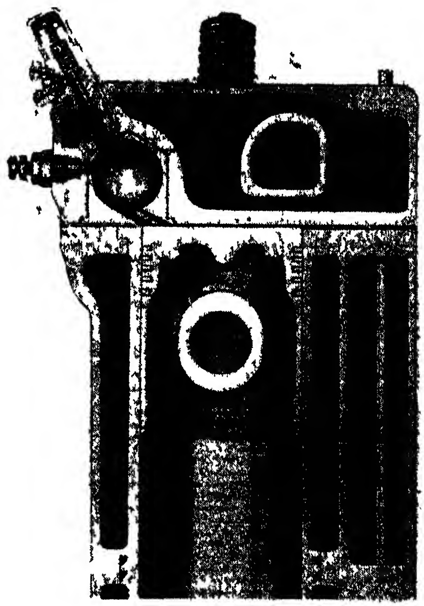


FIG. 241.—Ricardo turbulence chamber in cylinder head.

Fairbanks, Morse Turbulence Chamber.—By reference to Fig. 177 the reader will observe that the Fairbanks, Morse & Company four-cycle Diesel is provided with a turbulence chamber. This, however, is not a sphere but is a flat cylinder.

De La Vergne Combustion Chamber.—The Price combustion system employed by De La Vergne Engine Company on its larger Diesels is quite different from the turbulence chambers already mentioned. As may be seen from Fig. 242, this chamber is an inverted truncated cylinder with its flat top (the inverted base) providing room for the inlet and exhaust valves. Air is drawn into the cylinder through the inlet valve and combustion chamber. On the compression stroke the hot cylinder air is forced back into the combustion chamber. Fuel is forced through two opposing spray nozzles, into the moving mass of air. It is the claim that the location of the spray nozzles prevents the oil from striking the cold walls, as both oil jets reached only the hot central mass of air.

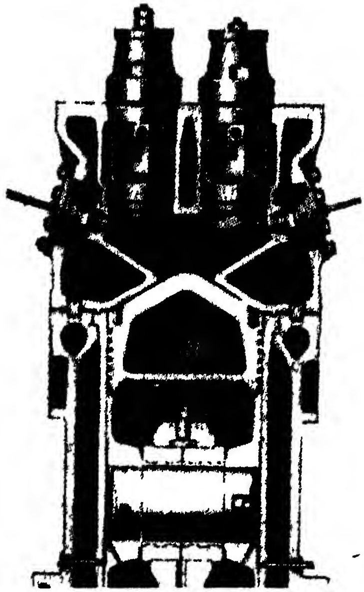


FIG. 242.—Combustion chamber of De La Vergne Diesel.

One advantage possessed by this design is that the piston is shielded from the heat of combustion; consequently, a large-diameter piston may be used without cooling of the piston crown. The disadvantage of the design when applied to medium- and high-speed Diesels is the wiredrawing of the air through the throat, so that the cylinder does not receive a full air charge, and a brake mean effective pressure of over 75 lb. cannot be obtained.

In its high-speed Diesels, De La Vergne may be said to use only one-half of the Price combustion chamber, so that the turbulence chamber becomes pear shaped. As shown in Fig. 181, the single-spray valve is horizontal, and the inlet and exhaust valves open into the main cylinder space.

Precombustion Chamber.—In engines employing the precombustion-, or outer-, chamber principle, a small chamber, with a volume equal to about 35 per cent of the total clearance volume, is connected to the working cylinder by a small throat.

Part of the cylinder's air charge is forced into this chamber during the piston's compression stroke. Oil is sprayed into this small mass of hot air; ignition occurs; and although part of the oil may burn completely to carbon dioxide, 70 per cent merely

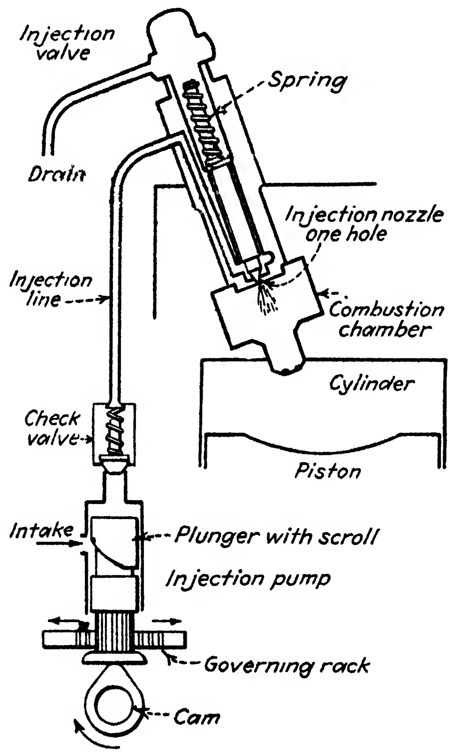


FIG. 243.—Diagram of the Caterpillar combustion system.

gasifies or becomes highly heated and is carried back into the working cylinder when the piston passes top dead center. The partial combustion in the precombustion chamber creates a high initial pressure which promptly forces the chamber's contents into the cylinder. Here the main oil charge mixes with the air charge and continues the combustion.

Caterpillar Precombustion.—The most important Diesel builder using the precombustion chamber is the Caterpillar Tractor Company. The engines of this company were developed for tractor application. It was necessary to have an engine that could be safely handled by inexperienced men, and the precombustion chamber makes an engine less sensitive to load and fuel than most other combustion devices. The thermal efficiency may suffer somewhat, although the Caterpillar will show a fuel rate as low as 0.42 lb. per horsepower-hour, but the simplicity offsets any loss in efficiency. The Caterpillar fuel system is shown in Fig. 243.

Witte Diesel.—The small, high-speed Diesel built by the Witte Engine Company employs a similar precombustion chamber.

Air-cell Systems.—Two of the most successful air-cell systems were developed by Franz Lang. Lang was associated with Diesel in the commercial development of the air-injection Diesel engine at the M.A.N. plant. He felt that there was a market for high-speed Diesels and that the air-injection Diesel could not meet the requirements.

Turning to mechanical injection, he early encountered difficulties with combustion at high speeds. Leaving M.A.N., he continued his work and finally produced the Acro system. This is basically the injection of the fuel into the main combustion space, where combustion starts, and the addition of a small cavity, or air cell, into which some air is forced by the piston during the compression stroke. When combustion progresses in the main chamber and the piston starts on its power stroke, the air in the air cell expands and flows out into the main chamber, where it creates additional turbulence and insures the complete burning of the fuel charge. The pressure in the air cell reaches a value equal to the combustion pressure; and when the latter drops sharply as soon as the piston starts downward, the energy in the air trapped in the air cell is considerable, and the turbulence effect higher than might be expected.

Stover Diesel.—The Small Diesel built by Stover Engine & Manufacturing Company employs this Acro system, as do many others in Europe.

Lanova System.—In 1927 Lang started another series of investigations in an attempt to better the performance of the Acro system. Out of this came the Lanova system.

The principal features of the Lanova system appear in Fig. 244. The main combustion chamber is shaped like a figure 8 and consists of two lobes located below the intake and exhaust valves. Usually these lobes are recesses cast in the cylinder head. The fuel is injected horizontally so that it breaks up into two main streams, each of which flows in a lobe in a circular path. Opposite the spray nozzle is the air cell, or "energy chamber," as Lang terms it. This has a volume varying from 7 to 18 per cent of the main chamber volume.

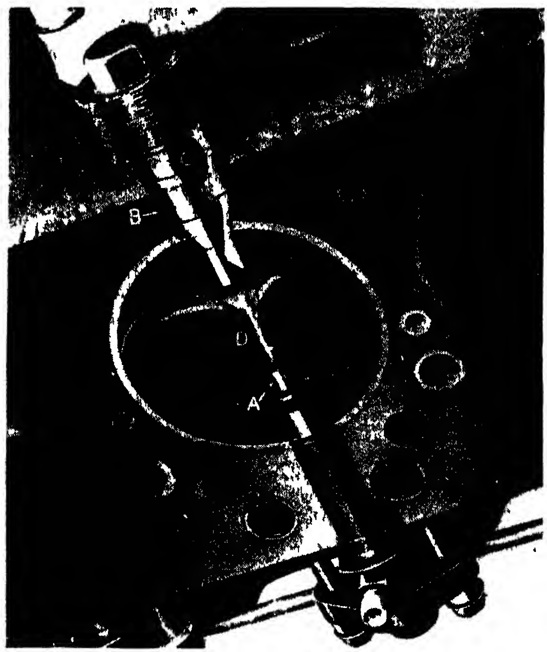


Fig. 244.—Action of the Lanova combustion system as used on the Dodge Diesel in which *A* is the spray valve, *B* is the energy cell, *C* is the interior of the cell, and *D* the double-lobe main chamber.

When fuel injection starts, air is being forced into the air cell by the movement of the piston, and this flow of air carries some of the fuel along with it into the air cell.

After the fuel and air pass into the air, or energy, cell, they are mixed by the turbulence existing within the cell.

Ignition of the fuel-air mixture occurs within the main charge in the lobes. The flame passes through the orifice into the energy chamber, and ignition of this secondary charge occurs, causing a high pressure in the cell. This pressure is 300 to 600 lb. above the cylinder pressure. The pressure difference causes the contents of the cell to backflow into the main chamber. The resulting turbulence completes the fuel-air mixing and insures complete combustion.

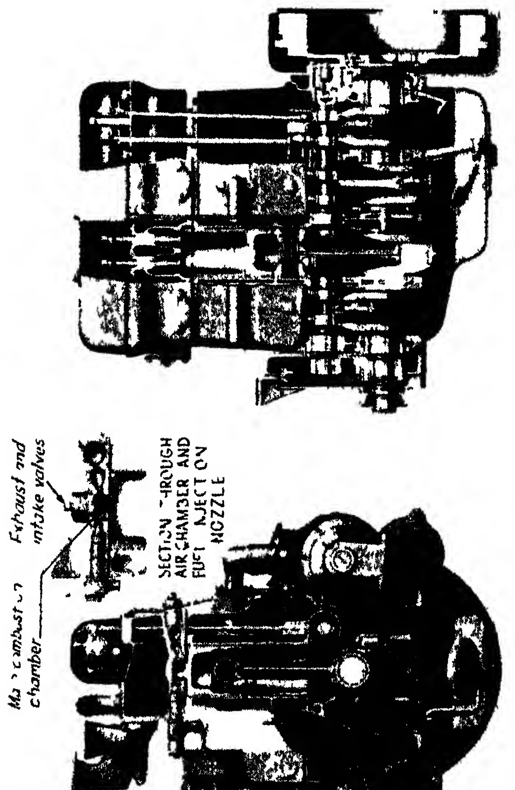


FIG 245 —Buda Diesel with Lanova combustion system

Lanova-system Engines.—Among the American Diesels using the Lanova system are Buda (Fig. 245), Mack, Dodge, and Atlas-Thornburg.

Spark-ignition Oil Engine.—To avoid the high pressures necessary to obtain oil-ignition temperature, Hesselman developed a spark-ignition engine wherein only air is compressed in the cylinder, to about 150 lb. pressure, and the fuel oil is injected by a pump and ignited by a spark plug This engine is built in America by Waukesha Motor Company and by Allis-Chalmers Manufacturing Company

A section through a Waukesha engine appears in Fig 246 The combustion space is formed in the deeply concaved piston

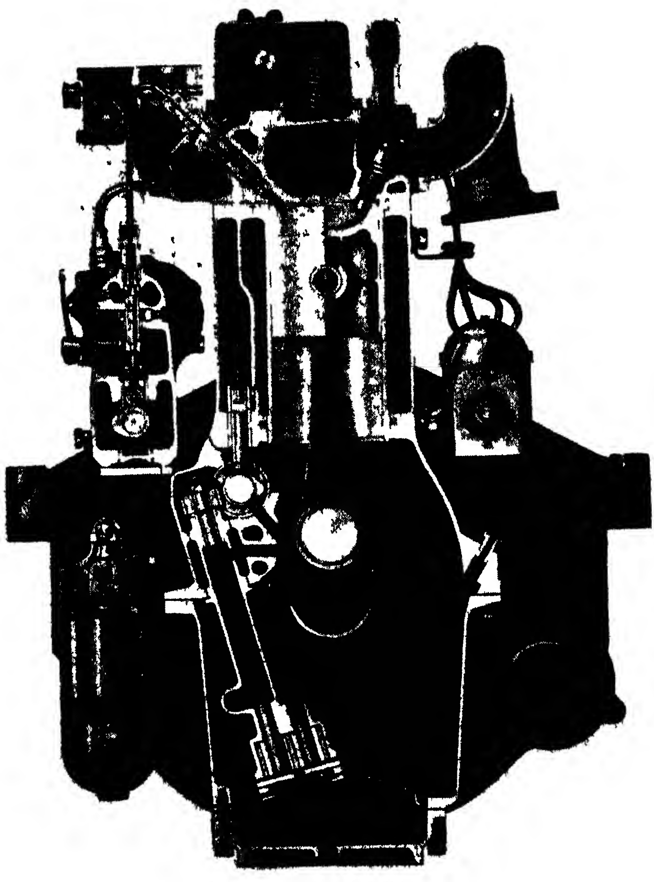


FIG. 246 —Cross section of the Waukasha-Hesselman spark-ignition oil engine

crown, into which the oil is sprayed diagonally. This engine's fuel consumption is somewhat higher than is the fuel consumption of a Diesel, but the pressures are lower, so the engine need weigh no more than a gasoline engine.

Types of Mechanical-injection Fuel Valves.—Three general types of fuel-spray valves are in use. With many engines the fuel valve is a simple orifice through which the oil is injected by the fuel pump. The injection starts as soon as the pump begins to discharge and ceases at the end of the pump delivery stroke. A simple check valve at the outlet prevents the cylinder compression air from entering the nozzle. The pump pressure is low, say 1,000 lb., just sufficient to force the oil through the relatively large orifice; Fig. 247 illustrates this design. No great attempt is made to obtain atomization of the fuel; its breaking up is the result of

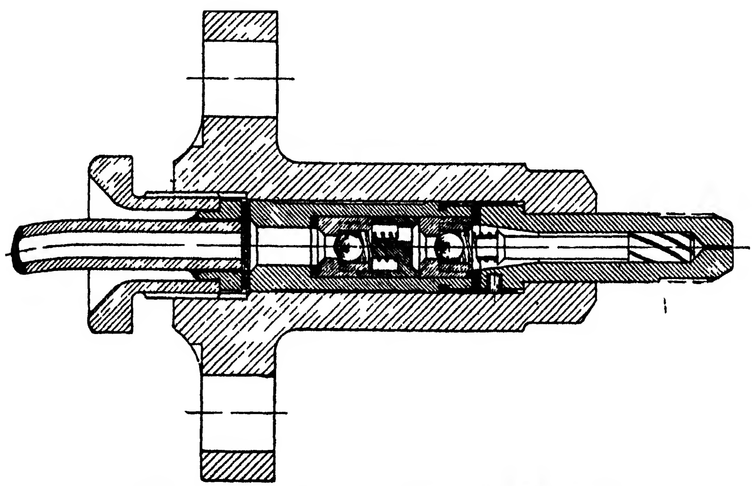


FIG. 247.—Check-valve, or open-type, fuel-spray valve.

the whirling of the air charge in the cylinder. The general type is used with many precombustion engines.

In other engines the same nozzle principle is employed, but penetration and atomization are brought about by the use of five or more smaller orifices of about 0.02 in. diameter. The valve is filled with a tip core which has a helical groove along which the oil is forced, the object being to give the oil a whirling motion. In most of these low-pressure valves the oil sprays are relatively solid and are not finely divided, as are the streams from the high-pressure nozzles.

Where high pressures are used, the valve always contains a needle valve. These needles may be lifted by a cam and rocker, as with the valve of an air-injection Diesel, or may be designed so that the needle has differential sections. This allows the needle to be lifted when the oil pressure reaches a definite high value and be closed by spring pressure when the pressure of the oil drops at the end of the pump's delivery stroke. The mechanically operated valve is used with the common-rail, or constant oil-pressure, systems.

Much might be written on oil sprays, atomization, etc.; but as this volume is intended for the operating engineer, the subject is passed over as being one of interest to the designer, who will find the Langley Field reports of value.

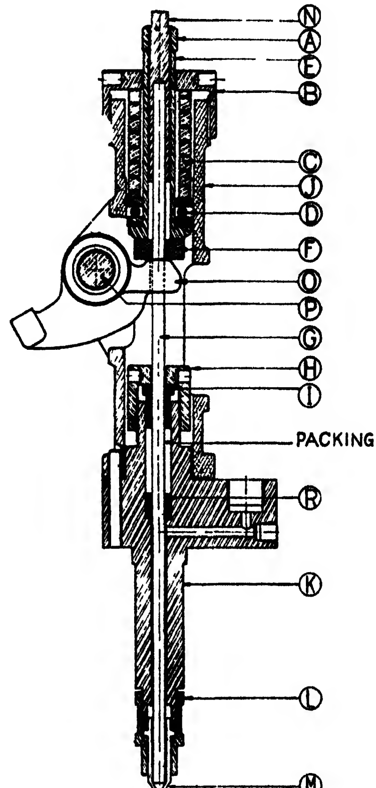


FIG. 248.—Cam-operated spray valve of Atlas-Imperial Diesel, with the parts labeled as follows: A = spindle-adjustment lock nut; B = spring nut; C = spring; D = ball thrust bearing; E = spindle extension sleeve; F = spray-valve spindle collar; G = spray-valve stem; H = gland nut; I = packing gland; J = spray-valve spring casing; K = spray-valve body; L = spray-valve seat nut; M = spray tip; N = spindle extension; O = rocker; P = rocker pin; R = lower gland.

Mechanical-lift Fuel Valves. When the common-rail fuel system is applied to an engine, the fuel is introduced into the cylinder by a spray valve almost identical with those employed on air-injection Diesels. As shown in Fig. 248, a casing contains a needle, or rod, whose lower end is conical and seats in the end of the case. A rocker, actuated by a cam (not shown), lifts this needle, allowing oil which fills the cavity around the needle to flow through the opening at the end of the case, into the cylinder. The various parts are listed in Fig. 248.

Engines employing cam-operated spray valves include the Superior four-cycle and Atlas-Imperial four-cycle Diesels.

Differential Spray Valve.—When the spray valve is provided with a spring-loaded needle as in Fig. 249 but is without a cam action, some other method must be adopted to cause the needle to lift and permit the oil to spray into the cylinder at the proper time.

This is accomplished by timing the pump so that it starts to deliver fuel at the correct time. To cause the needle to lift, to afford passage of the fuel into the cylinder, the needle is made with two diameters, so that the oil exerted upon the annular area created by the two diameters forces the needle upward.

As shown in Fig. 249, after the needle M is lifted, by the oil pressure exerted on the area C, the oil pressure exerts an upward force not only on the annular area but also upon the area of the valve tip. The result is that, although pressure of, say, 4,000 lb. per square inch may be necessary to lift the valve against the spring, the increased area after lift enables the oil to hold the needle open even if the pressure drops to, say, 3,000 lb. This, then, counteracts the drop in oil pressure in the valve when the oil starts to flow into the cylinder.

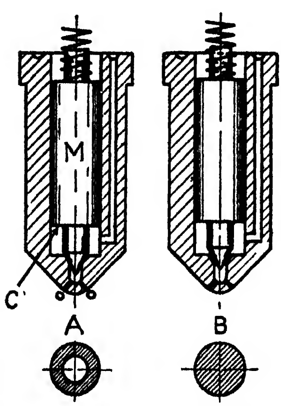


FIG. 249.- Action of differential-needle spray valve.

In action, this pressure condition should hold the needle wide open until pump delivery ceases, whereupon it should snap to the closed position. Pressure surges in the line between the pump and spray valve may prevent this desirable action.

Bosch Spray Valve.—The Bosch spray valve is shown in Fig. 250.

The valve stem is lapped in the guide bushing to a fit that makes any form of packing unnecessary; as a matter of fact, packing cannot be used without danger of the stem becoming too tight when the packing is compressed. An air vent should be provided at the uppermost part of the nozzle, to dispose of any air that may accumulate in the fuel line or in the nozzle, thereby avoiding "air binding."

In the Bosch design, the nozzle and guide bushing are integral. The nozzle holder is secured to the cylinder head by the usual yoke and studs. It contains the loading spring and the device for adjusting the spring pressure and is threaded at its lower end for the holding nut that secures the nozzle proper. The faces of the nozzle and its holder are ground and lapped to assure a good fit and freedom from leakage at any pressure. At the side of the holder are two connections, one for the fuel line, and the other to carry off, usually to the service tank, any

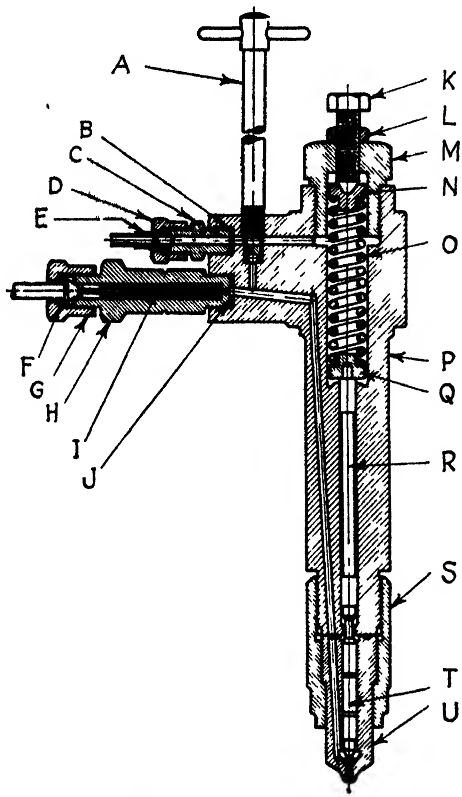


FIG. 250.—Parts of a Bosch spray valve, with parts as listed: *A* = bleed or by-pass valve; *B* = gasket; *C* = connector; *D* = packing gland; *E* = bleed or vent line; *F* = oil line from pump; *G* = gland; *H* = connector; *I* = oil filter; *J* = gasket; *K* = spring adjuster; *L* = locknut; *M* = valve cap; *N* = spring cap; *O* = spring; *P* = valve body; *Q* = spring collar; *R* = upper stem; *S* = clamp; *T* = differential needle; *U* = tip.

fuel that may leak past the valve stem. The loading spring is housed at the upper end of the nozzle holder, which is naturally the coolest portion. The lower end of the spring rests on a cap which in turn rests on the valve-stem proper, and the upper end abuts on the adjusting screw. A protecting cap covers the adjusting device. The holder is also equipped with a feeler pin which passes through a central hole in the adjusting screw. If this pin is depressed by the finger, the motion of the nozzle stem can be felt. This provides a convenient way to determine whether or not the nozzle is functioning.

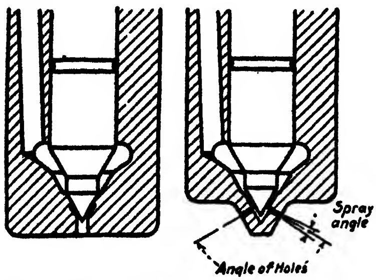


FIG. 251.—Single-orifice nozzle and multi-hole nozzle.

Nozzle Seats.—Valve seats of most nozzles are conical; but flat-faced seats are used occasionally, the advantage being that exact centering of the valve-stem is unnecessary. In certain forms of combustion chamber, the fuel jets must be delivered in definite directions. Nozzles and holders must be located by dowels in the cylinder heads of such engines.

Nozzle Orifices.—Nozzles may be classified as to their orifices as follows: single hole, multiple hole, circumferential, and pin.

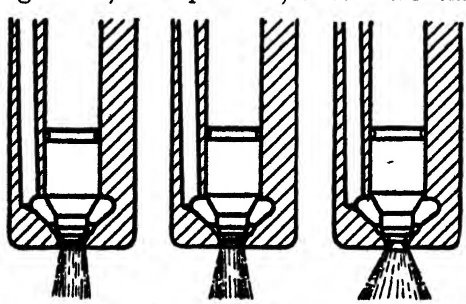


FIG. 252.—Pintle-type orifice.

Single-hole nozzles, like that shown at *A* in Fig. 251, are those in which the fuel jet is formed by a circular hole in the mouth of the nozzle. The axis of this hole can be the same as that of the nozzle or at an angle to it. The jet of such a nozzle is of conical shape, having an angle of 4 to 15 deg. At angles greater than 12 deg. the uniformity of the cone jet is easily disturbed by any slight inaccuracy in manufacture; and the wider the angle the more likely is the major part of the fuel to be ejected unevenly. Wider spray angles are obtained by the use of spiral groove to impart a centrifugal motion to the fuel particles.

Single-hole nozzles, with and without spiral grooves, are used in precombustion-chamber engines and those direct-ignition engines in which the fuel is distributed in the combustion chamber by strong air whirls.

Pintle nozzles are fitted with valves whose ends are furnished with a thin shank, or pin, as shown in Fig. 252, the shape of which is made according to the spray angle desired. This pintle, or pin, reaches into the nozzle orifice, so that an annular space is formed. By suitably shaping this pin, the designer will obtain either a hollow cylindrical jet of high power of penetration or a tapered spray with an angle varying from a few to approximately 60 deg. The wider the angle the better the atomization, but at a sacrifice of penetration. This arrangement permits fine variations in the spray characteristics; for instance, the cross section of the orifice can be opened gradually during the lifting of the valve stem if the pin is tapered or in steps if the pin is made in two cylindrical steps. In this way only a small quantity of fuel is injected at first, followed by the major part of the fuel. This procedure prevents a sudden pressure increase in the cylinder and gives smoothness to the running of the engine.

The double-throttling effect of the nozzle, first at the valve seat and then at the nozzle orifice, is responsible for the fact that the pintle nozzle works very uniformly and accurately. In addition, the motion of the pin in the nozzle orifice prevents the formation of a carbon crust. The pintle nozzle can be used with advantage to replace the single-hole nozzle previously described.

Bosch Fuel Pump.—The Bosch pump (Fig. 253) is employed by many Diesel builders. It may embrace all the pump plungers for the several engine cylinders in a single casing, or a separate casing may be employed for each pump plunger.

The general design of a Bosch pump arranged to deliver fuel to one cylinder can be readily understood by reference to Fig. 253.

The pump consists of a body containing a barrel *G* and a plunger *B*. The plunger has a recess section whose upper edge is helix.

The pump cylinder is enclosed by a bushing *H* to the upper end of which a gear segment *I* is fastened. This segment, in turn, engages with a toothed rack *R* which is actuated manually or by a governor. At its lower end, this bushing *H* has two opposite slots *K* in which a crossarm *J* of the piston is guided;

the angular motion of the bushing, caused by sliding the control rod, is thereby transmitted to the plunger *B*. No fuel is delivered by the pump when the control rod is at one extreme position; in the opposite position the maximum quantity of fuel is delivered.

When the plunger is at the bottom of its stroke, fuel flows into the plunger barrel, or cylinder, through the holes, or ports, *P*

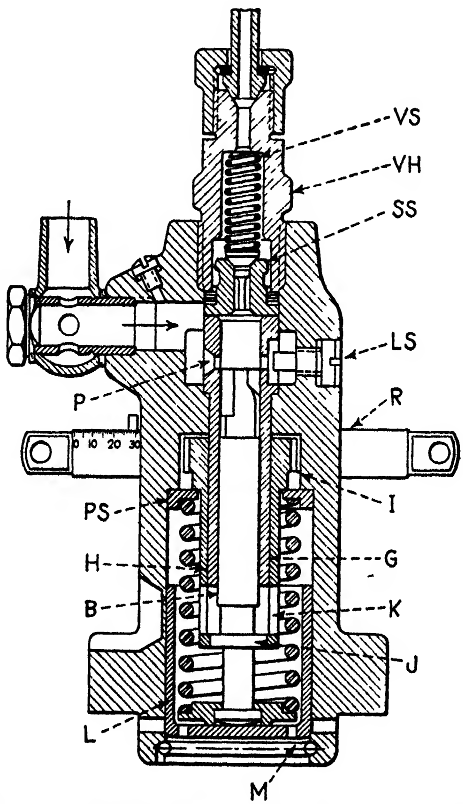


Fig. 253.—Section through a Bosch pump where *B* = pump plunger; *G* = pump barrel; *I* = control pinion; *J* = yoke; *K* = slot in *G*; *L* = plunger guide; *LS* = locking screw; *P* = suction port; *PS* = spring plate; *R* = control rack; *SS* = delivery-valve seat; *VS* = delivery-valve spring; *VH* = valve holder.

in the side of the barrel shown in Fig. 253. On the return stroke, as soon as the piston has passed the top edge of the holes *P* in the barrel, oil delivery begins, and any continued movement of the plunger, naturally, results in continued delivery of fuel. (The actual starting point of the injection period in the engine's cycle, however, is determined by the spring in the nozzle holder and by the timing of the injection pump.) The amount of fuel oil delivered on each stroke of the plunger is determined by the position of the plunger, or piston, a part that can be rotated inside the barrel, or cylinder. The manner in which this controls the amount of fuel oil injection is shown in Fig. 253.

A fuel-transfer pump keeps the fuel-pump suction line filled with oil under a slight pressure.

The pump-plunger barrel ports *P* are always submerged in this fuel oil, so that when the top edge of the pump plunger uncovers the top edge of the ports *P* and during the rest of its downward stroke and until the top edge of plunger closes the top edges of these ports on its upward stroke, fuel rushes in through these two ports, filling the chamber above the plunger in the barrel *G*. As this chamber is connected by a vertical groove with the recess in the plunger formed by the helical edge and the lower edge, fuel also flows down the groove, filling this recess.

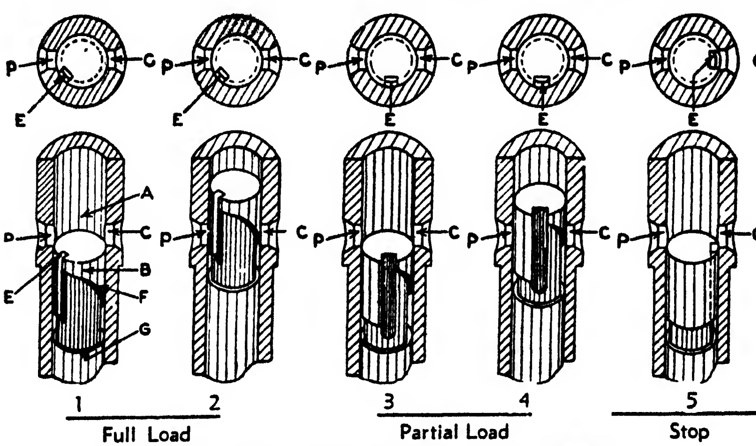


Fig. 253 *A*.—Action of fuel-quantity control on Bosch pump.

The location of the various grooves and edges in relation to the intake-port opening *P*, when the pump plunger has been rotated to the extreme left, giving the maximum effective pump-plunger stroke, is shown at 1 (Fig. 253 *A*). This is the position of the plunger at the beginning of its upward stroke when starting the engine or when and while maximum power is demanded.

When the plunger travels upward, at some exact point the top edge of the plunger just closes the two intake ports *P* and *C*.

At this exact point the fuel that has filled the chamber *A*, formed above plunger *B* in barrel *G* and the groove *E* and the recess formed between helical edge *F* and lower edge (Fig. 253 *A*), is trapped in this chamber, groove, and recess and cannot escape, as the vertical groove *E* does not register with any port opening, and helical edge *F* does not register with port opening *C*. The only way fuel can escape is through these two port openings, and therefore one of these edges or the top edge of the plunger must index with one or both of the suction-port openings *P* before the fuel can escape except by injection through the pump discharge valve *V* (Fig. 253) and the nozzle in the engine cylinder. A spring set at a predetermined pressure holds the nozzle valve located in the nozzle-holder assembly on its seat until the pressure in the pump and in the fuel lines equals this spring pressure, at which time the valve is lifted off its seat by this fuel pressure, and fuel is sprayed into the combustion chamber of the engine cylinder. Injection continues until either the helical edge *F* or the vertical groove *E* uncovers the port opening *C* which causes instantaneous release of pressure. This is the position shown at 2 (Fig. 253 *A*) and indicates the point at which fuel injection closes. For the arrangement of spray nozzle, piping, and pump examine the several illustrations.

In the release position shown at 2 (Fig. 253 *A*) the plunger has just completed its upward stroke and is at the exact point when helical edge *F* uncovers port opening *C*, giving the maximum effective plunger stroke. This is the position of the plunger at the end of the injection stroke when the engine is being started or maximum power is demanded. At the instant helical groove *F* uncovers the bottom edge of port opening *C*, the fuel in the chamber, groove, and recess is released through this port opening into the pump suction reservoir, and the pressure is thereby also instantly released. The spring pressure applied on the spray-nozzle valve in the nozzle-holder assembly at the engine cylinder is then greater than the pressure in the fuel line, so the nozzle valve snaps back onto its seat, which cuts off injection into the combustion chamber.

Just above the chamber in the pump barrel is located the check valve *V*, called "fuel-pump delivery valve" (see Fig. 253). When the pressure on the fuel in the pump barrel is raised, the valve is lifted off its seat and allows the fuel to be forced into and through the fuel line to the nozzle. When the fuel pressure is released by the helical edge *F* of the pump plunger uncovering the port opening *C*, the fuel in the fuel line closes this discharge valve *V* in the fuel pump. In this way the fuel oil in the fuel piping is trapped between the discharge valve in the pump and the nozzle valve in the nozzle-holder assembly at a pressure just slightly less than the opening pressure of the spray-nozzle valve. This eliminates the necessity of building up the pressure in the fuel lines for each plunger stroke.

The sketch in Fig. 253 *A* at 3 shows the pump plunger at the end of its downward stroke and beginning its upward stroke, assuming that the engine is carrying only a partial load. The operation of the plunger is the same as described under position 1, the only difference being the location of the vertical groove *E* and helical edge *F* in relation to the port opening *C*. The control rod has rotated the partial-load position. Compare location of groove *E* with position 1

At position 4 (Fig. 253 *A*) the pump plunger is at the end of its effective stroke when partial load is demanded of the engine Helical edge *F* has uncovered the lower edge of port opening *C*, releasing the fuel and pressure; therefore, injection has just stopped It will be noticed that the top edge of the pump plunger is considerably below the top edge of the plunger shown at 2 in Fig. 253 *A*, but in each case injection has just stopped owing to the helical edge uncovering the port opening *C*. Fuel was, therefore, injected for longer periods of time at 2 in Fig. 253 *A* than at 4 in Fig. 253, so more fuel was injected during the longer effective stroke shown at 3 and 4 in Fig. 253 *A* than in the shorter effective stroke shown at 1 and 2 in Fig. 253 *A*. The plunger, however, in both cases must make the same length of upward and downward stroke, owing to the cam action that operates it. In position 4 (Fig. 253 *A*) the rest of the upward plunger stroke will not deliver any more fuel to the spray nozzle, for there is no fuel under pressure left in the chamber, groove or recess, since it has escaped through the port opening *C* back into the fuel suction reservoir in the pump housing. The engine cannot carry so much load or, if carrying the same load speed, will be reduced with the shorter effective stroke shown at 4 in Fig. 253 *A* as compared with the longer effective stroke shown at 2.

Position 5 (Fig. 253 *A*) shows the pump plunger just beginning its upward stroke when it has been rotated manually to the extreme right, to stop the engine. Notice in this case that the vertical groove E is registered with port opening C and that as this slot is vertical it will register with port opening C during the whole stroke of the plunger. Fuel rushes in through both port openings P and C as long as the top edge of the plunger leaves them uncovered, filling the chamber, groove and recess. When the top edge of the plunger covers the top edge of the port P, the pressure is not raised, however, for in position 5 (Fig. 253 A) the fuel is not trapped but can and does escape through vertical groove E back into the pump reservoir because groove E is registered with port opening C during the complete plunger stroke. As no fuel is delivered to the nozzle, the engine must stop.

These illustrations represent any one plunger and show the locations of the various edges and grooves under different engine-load conditions and are not shown as for engine-cylinder Nos. 1, 2, 3, 4, and 5. All plungers in the pump are rotated to exactly the same position simultaneously by the fuel-injection-pump control rod, so any one figure shown represents the location of all six plungers at some point in the engine cycle.

Timing Bosch Pump.—To facilitate the adjustment of the fuel-injection pump and the timing of the fuel injection, an inspection window is provided in the pump casting, through which a mark on the plunger guide should always be visible when the pump plunger is at top dead center or bottom dead center of its stroke. The beginning of injection is indicated when the mark on the plunger guide coincides with the mark on the side of the window This adjustment is, however, correct only for a certain opening pressure of the nozzle and for a certain speed of the pump. The mark on the plunger guide should coincide with the mark on the side of the window when the engine crankshaft stands at the number of degrees ahead of top dead center, as called for by the engine builder's timing data. Top dead center for the engine crankshaft is indicated when a line that is marked on the fly-wheel comes even with a line marked on the end plate. A tappet screw is provided directly under the fuel-injection pump. This screw is held with a locknut. By loosening the locknut the tappet screw can be adjusted either to raise or to lower the line on the fuel-pump plunger guide at the will of the operator. Great care should be exercised when moving this screw to see that the mark on the plunger guide is still visible in the inspection window at both the top and the bottom of the pump-plunger travel. Serious injury to the pump may result if the pump plunger is raised or lowered too far by the tappet-adjusting screw. The engineer must be cautious.

On some engines these marks are absent from the pump, so it is necessary to find the point at which the pump starts delivering oil by the flow method already outlined or by direct measuring of the lift (if the engine builder supplies these data).

Flow Method of Timing.—Read the description of the plunger operation carefully, for when timing the fuel injection pump by the "flow" method, the point at which the pump is timed is when the No. 1 engine piston is at a position before top center as recommended by the engine builder as the time at which injection should start. At this point the pump plunger should exactly close the two port openings *P* and *C* on its upward travel. This point must be exact, or the pump timing will not be correct. That is why it is necessary that less than $\frac{1}{64}$ in. movement of the outside diameter of the pump coupling is the difference between fuel flowing and not flowing. The pump must be timed at the exact point of closing, not just before or just after, to be timed correctly by the "flowing" method.

Usually flywheels have a line marked *DC* (dead center), and on both sides of this line are graduations designating each 2 deg. of crankshaft travel, and each 10 deg. up to 40 deg. is marked with numerals; on some engines these marks may not appear on the flywheel.

The steps to take in "flow timing" are as follows:

1. Rotate the flywheel by means of the hand crank until the *DC* mark on the flywheel appears in the timing hole of the bellhousing or in line with a timing mark on the engine. Be sure that No. 1 piston is just completing the compression stroke and beginning the expansion which can be determined by observing that the No. 6 cylinder exhaust valve is nearly closed in a six-cylinder engine. The engineer should be extremely careful that the correct top dead center is used.

2. Rotate the engine in the direction of the degree graduation marks, that is, counter engine-wise, until the injection-point mark on the flywheel is directly in line with the mark in the center of the timing hole in the bellhousing.

Install the pump assembly, tightening all attaching screws but leaving the rear half of the coupling loose from the front half, so that the pump shaft can be rotated while the drive shaft remains stationary.

Connect all fuel-suction and -discharge pipes from fuel tank to pump. Install all fuel lines except to the No. 1 cylinder.

With the governor lever in the wide-open, or full-load, position, proceed to prime the pump.

Put the governor lever in the "stop" position, and remove the pump delivery-valve holder. Remove the delivery valve and spring but not the valve seat. Replace the delivery-valve holder, finger tight.

Place the governor lever in wide-open, or full-load, position. The fuel should now rush out of the delivery-valve holder. Rotate the pump shaft over the top and away from the engine by means of the rear half of the coupling until the fuel flow stops. If the fuel did not flow when the governor lever was first opened, rotate the shaft until it does, then back to where it is just off.

Very carefully rotate the shaft until the fuel just barely flows, then back to the point at which the flow is just barely shut off. Repeat this two or three times until a movement of less than $\frac{1}{64}$ in. on the circumference of the pump-drive coupling is the difference between fuel flowing and not flowing. This determines when the pump plunger just closes the fuel port and begins the period of building up pressure in the lines and nozzles so that injection can start. It is very necessary that this adjustment be extremely accurate.

With capscrews provided, connect the front and rear half of the coupling. Be sure that these screws are tight so that no slippage can occur, and yet do not strip the threads. It is not advisable to use a wrench over 6 in. long for tightening. Also observe if any slight movement, which might occur when tightening the screws, has started the fuel flowing again from the delivery-valve holder. When these screws are tight, no fuel should flow.

The fuel pump is now timed to close the ports at the proper point of injection before top center.

Put the governor lever in the stop position again. Remove the delivery-valve holder, and replace the delivery valve and spring. Install the delivery-valve holder, tightening it firmly. Be careful not to get any dirt, water, or any other foreign matter in or on any of these parts.

Connect the fuel line to pump No. 1 cylinder. Prime the fuel lines as explained heretofore, being sure the fuel pump, strainer, and all lines are full of fuel with no air.

Start the engine.

If after checking all points the engine still runs "ragged," stop, and recheck the timing.

After the engine is operating smoothly and has been properly warmed up, stop it.

With a light chisel and hammer enlarge the single mark on the front coupling hub, and put a corresponding mark on the other hub of the coupling so that these two parts can be lined up together at any future time without the necessity of "flowing" the pump.

Ex-Cell-O Pump.—The Ex-Cell-O Type A fuel-injection pump has been developed for application to high-speed Diesels. It combines a "hydraulic unit," made up of the parts that handle the fuel oil; a "drive unit," embracing those members which operate the hydraulic unit; the "governor"; and the "transfer pump."

The Ex-Cell-O model-A fuel pump (Fig. 254) is novel in that it employs a wobble plate *B* to give a reciprocating motion to pushrods *D*. The plate *B* is driven by gears from the engine shaft. Against the plate the shoes *C* bear, each having a recess to carry the pushrods *D*. These rods, in turn, contact with the ends of pump plungers *P*, of which there is one for each engine cylinder. In this manner, the plungers are given a reciprocating motion by the rotation of the wobble plate *B*. A central valve *H* is rotated by the pump shaft *F* through a bayonet point. This valve has a cut-away section *O* which is brought into communication with the port leading to the barrel of the pump plunger. When this alignment occurs, oil passes through the valve and port to charge the pump barrel. The amount of this charge depends upon the time at which the port and valve are open to the flow of oil. The governor, acting through a linkage *L*, shifts arm *K*, which, in turn, moves the valve to the left or right. This varies the time during the plunger's discharge stroke so that the oil can flow back through the valve *H*.

Each outlet *J* connects to its respective spray valve in the engine-cylinder head. The transfer pump delivers the oil through the filter to the suction line, which is shown at the top of the upper drawing.

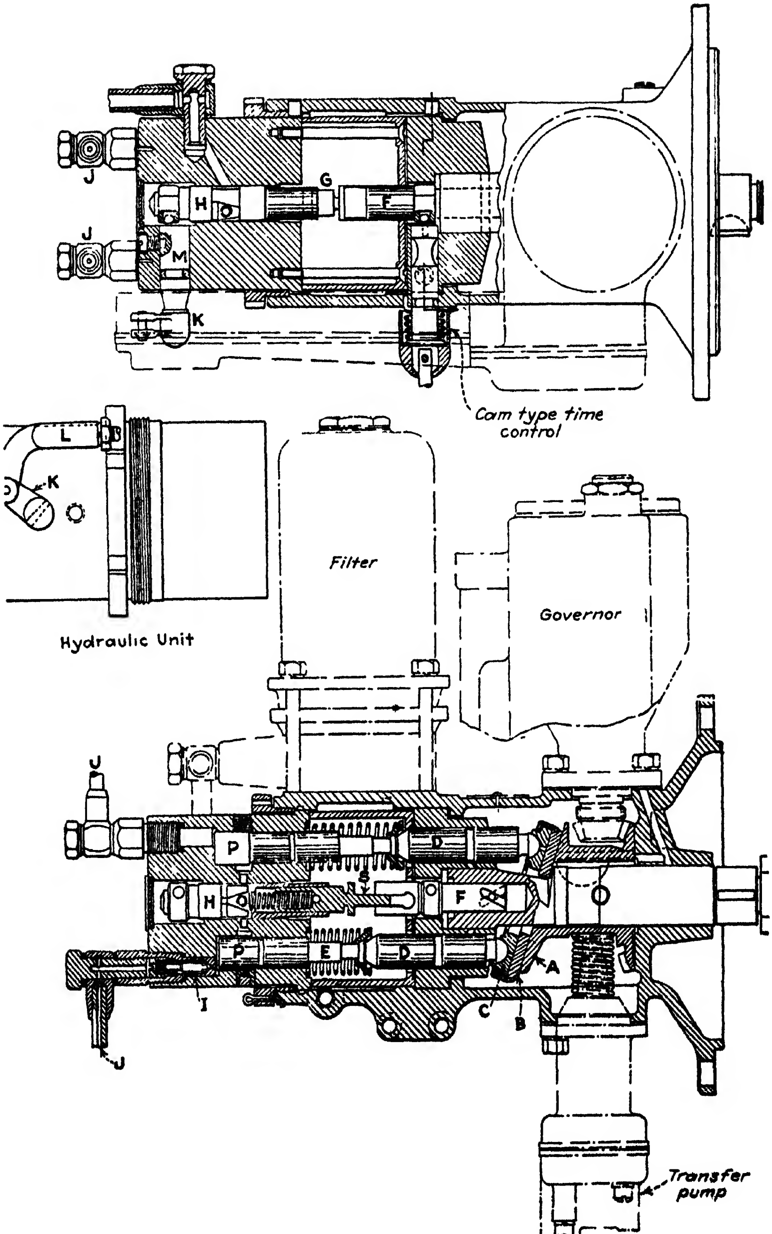


FIG. 254.—Section through Ex-Cell-O Type A fuel pump.

Ex-Cell-O Pump Adjustment.—If hard starting is experienced, other possible difficulties should be eliminated before suspecting the hydraulic unit of wear. The procedure is:

Make sure that there is a supply of fuel in the tank and that there are no closed valves between the tank and the strainer just ahead of the fuel-transfer pump.

Make sure that the strainer just mentioned is clean and is emptied of any sediment or water, or ice if below freezing. At the entry to the fuel-transfer pump is a screen which must also be clean.

Make sure that the transfer pump is delivering fuel.

Make sure that both service and safety filters are clean and have been filled with fuel, bled of air, and primed as outlined.

Make sure that there are no air leaks in the supply line. This is especially important if the transfer pump must lift the fuel by suction.

If the foregoing five items are correct and the engine refuses to start, then remove all nozzles from the engine, turn them end for end so that the spray will shoot away from the engine and not into the cylinder, and tighten the nozzle-tubing nuts. With the throttle open (accelerator depressed) and the engine cranking, the nozzles should inject, or spray, fuel into the air. Cranking the engine in this instance means spinning it with the electric starter or turning it by other means at a speed equal to that attained by the engine when the starter is cranking against normal compression. If the nozzles are injecting, there is no fault in the fuel pump, supply system, or nozzles. If one or more nozzles do not inject, examine the associated check valves. If the nozzles do not inject, and the check valves, throttle control, or drive unit is not at fault, then the hydraulic unit must be replaced; but before deciding, examine the throttle control as follows:

Remove the throttle-control cover. The throttle-control lever should stand in the opening position, which is vertical or at an inclination rearward of its vertical position. If the throttle control does not stand in the opening position, then the governor is at fault in connection with failure to start and should be inspected as outlined.

Next remove the hydraulic unit to permit inspection of the drive unit. This inspection should be made before replacing the hydraulic unit.

Look in the end from which the hydraulic unit was removed. The central slot, which drives the tang of the rotary valve of the hydraulic unit, should rotate when the engine is turned. This slot is the timing member, replacement of which is described.

Next reach into the end from which the hydraulic unit was removed, and push in on the tappets. These should reciprocate (move in and out) as the engine is turned. Before reassembling, make sure that the ball ends of the tappets enter the spherical sockets of the tappet shoes.

If the drive unit is in good order, the fault is in the hydraulic unit, which should be exchanged.

Even if the drive unit is found to be at fault because the central slot does not turn, the hydraulic unit removed from such a pump should be returned for inspection.

De La Vergne Fuel Pumps and Spray Valve.—The De La Vergne Diesel employs the Price separate combustion chamber,

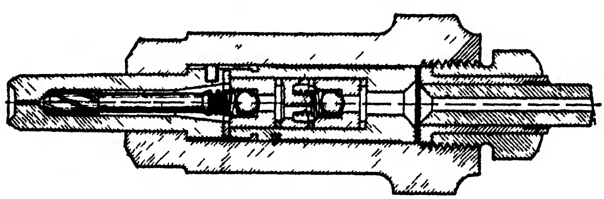


FIG. 255.—De La Vergne spray nozzle

as shown in Fig. 242. Two nozzles are placed at opposite ends of the truncated cavity. As a low oil pressure is used, the valve is a simple nozzle with a double-ball valve and a helically fluted core, as shown in Fig. 255.

The De La Vergne fuel pump is shown in Fig. 256. The pump plunger is of a size that it handles about 100 per cent more oil than the engine needs at full load. The amount not needed to maintain engine speed is by-passed through a valve that is opened at the proper time by a linkage from the pump plunger. The governor, as will be seen, controls the fulcrum part of the by-pass valve lever and so regulates the point in the plunger's stroke at which the by-pass valve closes. Each engine cylinder has its individual fuel pump, which is mounted on a level with the cylinder bases. The pump cams are set so that the fuel injection starts at about 19 deg. before top dead center of the working piston.

The fuel-pump cams are individually adjustable. When a cam is initially set, a toolmark at the nose of the cam should be directly under the center of the pump roller when the crank of the corresponding cylinder is approximately 10 to 15 deg. after top dead center on the firing stroke. After this initial

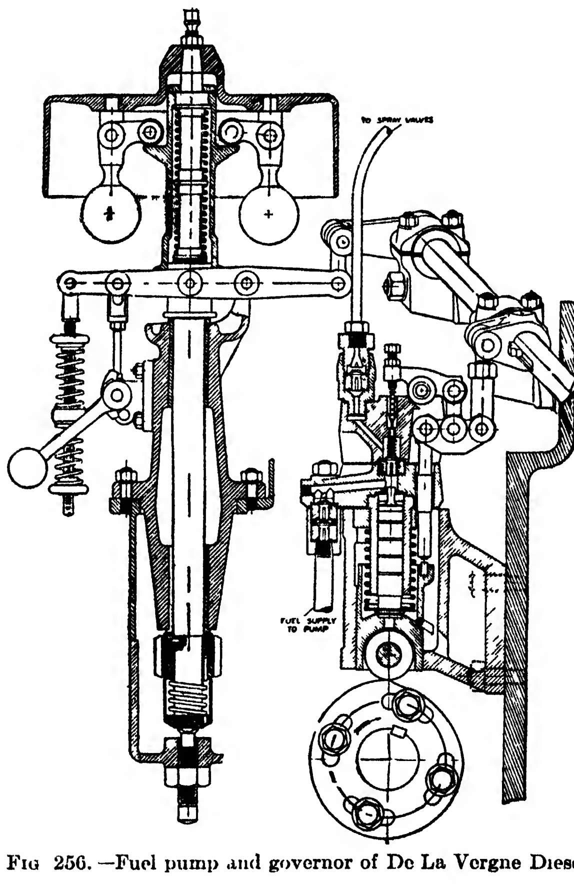


FIG 256.—Fuel pump and governor of De La Vergne Diesel.

setting has been made, the engine should be operated, and maximum combustion pressures at approximately full load determined for all cylinders, by means of an accurate cylinder indicator. If the pressures are too high, retard the fuel-pump cams; conversely, if too low, advance the cams. When properly adjusted, maximum pressures for all cylinders should be equal within 15 lb. The maximum combustion pressure at full-load operation should be from 550 to 565 lb. for VG engines previous to serial 1523, VA engines previous to serial 1436, and VB engines previous to serial 1454. For all engines above these serial numbers, the maximum combustion pressures should be from 580 to 600 lb.

Fuel pumps should be inspected regularly for wear and general performance. Bar the engine over until the roller of the pump to be inspected is at low position on the cam. With the pump full of fuel, insert a solid gasket in the discharge connection of the fuel pump. Pull the priming lever by hand, but do not jerk it. If there is no leakage, the plunger will not move. Leakage may result from a worn plunger and cylinder, suction valves, or by-pass valve. Recondition or install new parts as required. The by-pass-valve seat should not have any recess as the result of wear or regrinding. The by-pass push rod should work freely in its bushing and without undue clearance. When reassembling or installing new fuel-pump parts, wash them all in clean kerosene or fuel oil. Plungers and cylinders should be reassembled and worked a number of times while submerged in oil to assure free movement and freedom from foreign matter.

At intervals, the fuel-injection pressures should be determined, using a Bosch maximum-pressure indicator with adaptor connected into the fuel-pump discharge and while the engine is operating at approximately full load. If pressures are not within specified limits, the difficulty may be caused by restrictions in spray valves or fuel pipes or by improper-sized spray nozzles.

Ingersoll-Rand Fuel Valve.—Models of this engine using the Price system, built up to about 1935, had a simple check-valve fuel nozzle. The checks are of the poppet type, slightly spring loaded. As the Price system does not need a finely atomized oil spray, the design of fuel valve is satisfactory. Two sprays are used for each cylinder, being supplied with oil from a distributor serving all the cylinders.

It is imperative that where two nozzles per cylinder are employed, the two fuel lines should be of the same length to insure a balanced flow of the oil to the two nozzles.

Ingersoll-Rand Fuel Pump.—This Price engine uses a single fuel pump for the entire engine, regardless of the number of cylinders. To do this a distributor, to be described later, is used, and the cam operating the pump plunger has a nose for each engine cylinder, as shown in Fig 257. The pump has two suction valves *G* and a by-pass valve *F*. The reach rod *B* transmits motion from the pump plunger to the crosshead *C*. The by-pass locker *D* is lifted by *C*; and as it fulcrums at *E*,

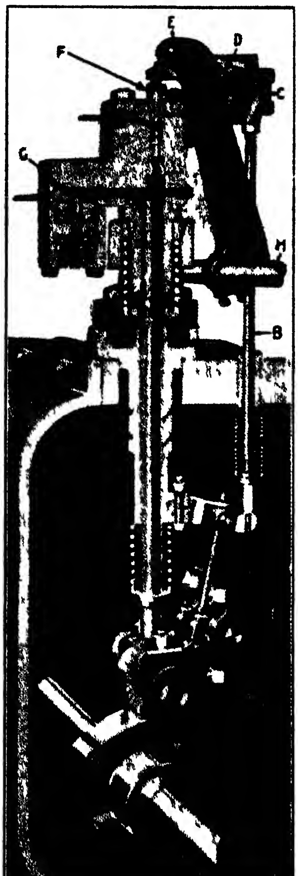


FIG 257.—Fuel pump of Ingersoll-Rand engine.

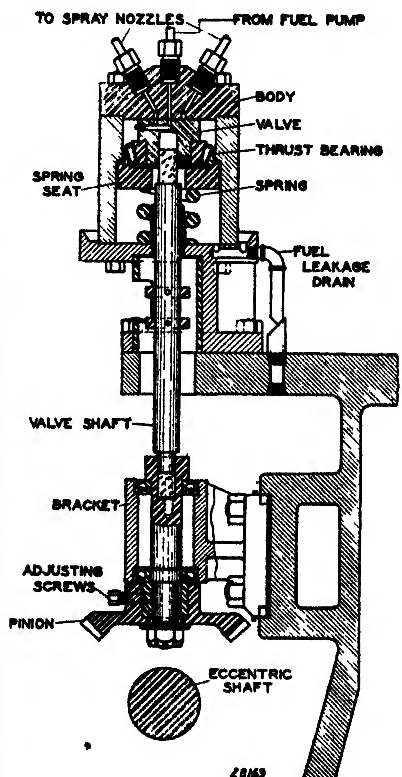


FIG 258—Continuous-rotation distributor of Ingersoll-Rand engine.

the other end of *D* opens the by-pass valve *F*. At a change in load the governor shifts the fulcrum *E*, so that the rocker *D* opens the valve earlier or later, as the case may be.

The pump is set to start injection at 20 deg. ahead of top dead center. To alter the turning, a shim is added to or removed from behind the roller rocker *A*.

The distributors used differ in that on the older engines the distributor plate stopped at each time of injection but on the later engines the distributor rotates continuously.

Continuous-rotation Distributor.—This distributor is driven from the eccentric shaft by a bevel gear. The feed pipe from the fuel-injection pump enters the body of the distributor at the center, and the feed pipes to the cylinders leave the body about its conical surface as shown in Fig. 258. A port leads from each feed-pipe connection to the under surface of the distributor body against which the valve bears and rotates.

This valve contains a central port connected by a passage in the valve to an offset elongated port on its face. The central port mates with the port from the fuel-injection feed pipe at all times, whereas the elongated port mates successively with each of the ports to the spray-nozzle feed pipes. It remains mated to one port during a complete injection period for one cylinder. The valve is held against the body by a strong spring, thrust being taken by the roller thrust bearing. Pressure of the spring is adjusted by inserting washers under the spring seat. While running, the fuel pressure on top of the valve partially counterbalances the spring pressure.

Present Ingersoll-Rand Fuel Pump.—On its new design the Ingersoll-Rand employs the Bosch fuel pump, with two spray valves delivering oil into the cylinder clearance space. The Price combustion chamber has been abandoned.

Fairbanks, Morse Model 32B Fuel Nozzle.—Originally, this engine employed the precombustion principle. Consequently, no high degree of atomization was needed. A simple check-valve nozzle was used, the oil spray discharging toward the opening into the cylinder. The meeting of the oil and air forced into the precombustion chamber by the advancing piston gave enough turbulence to insure satisfactory combustion. Because of the restrictions set up by the precombustion principle the horsepower developed per unit of cylinder volume was low, but the thermal efficiency was reasonably good.

At present this engine employs direct injection of the fuel into the cylinder clearance space. The spray valve, located in the center of the head, is of the differential type, as shown in Fig. 259. Since tle pump pressure is higher than in the pre-

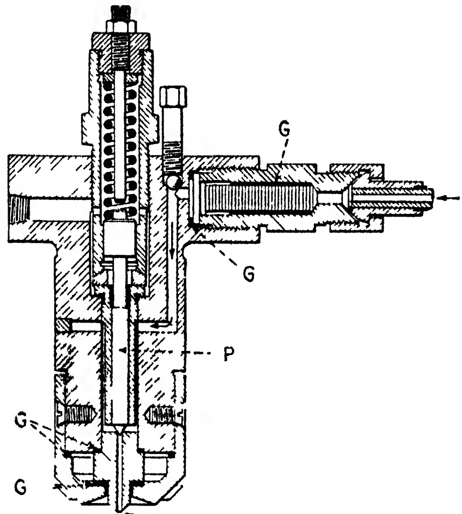


FIG. 259.—Section through Fairbanks, Morse spray valve.

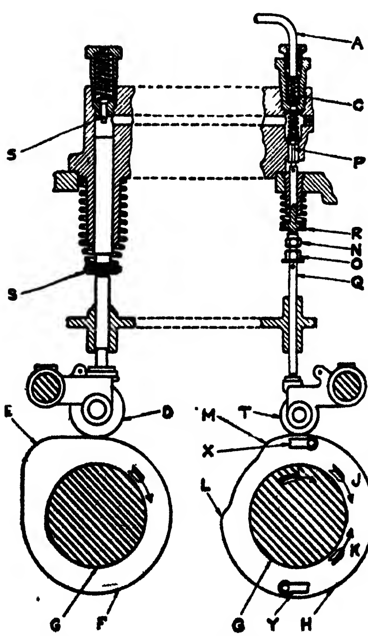


FIG. 260.—Schematic layout of Fairbanks, Morse pump.

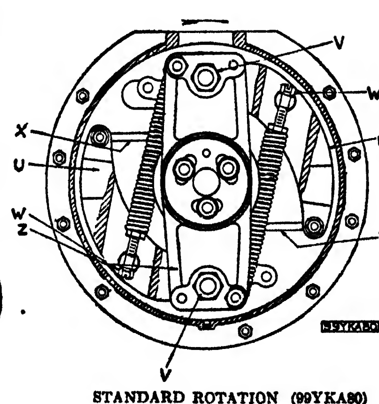


FIG. 260 A.—Governor used with Fig. 260 pump.

combustion model, the fuel-injection pump has been made heavier, although the design remains unaltered.

Fairbanks, Morse Model 32B Fuel Pump.—This pump is shown schematically in Fig. 260. The pump plunger is operated by a cam fixed on the crankshaft, whereas the suction valve is opened and closed by a movable cam under governor control. As the plunger *B* descends, fuel is drawn in through the suction valve *P* which is held open by the governor cam *H*. When the plunger starts on its upward stroke, oil is discharged out through the open suction valve until the low portion of the cam *H* comes under the roller *T*, whereupon the suction valve is closed by the spring *R*, and the pump pressure lifts the discharge valve *C* and forces oil to the spray nozzle. To insure that oil will not dribble after the pump delivery stroke ends, the plunger strikes the relief valve *S*, which allows the fuel to by-pass. Figure 261 shows the actual pump.

In order that the fuel may enter the combustion chamber at the proper time, injection must occur in a certain relation to the position of the power piston. In other words, the injection cam must be set in a certain relation to the crankshaft.

The method of timing the injection is to clamp the governor mechanism to the crankshaft in such a position that the injection-pump plunger *B* is at high point a certain number of degrees after the corresponding piston has passed upper dead center. The following tabulation gives the settings for the different engines:

Size of engine, inches	Single cylinder, after top center, degrees	Multicylinder, after top center, degrees
12 by 15	3 to 5	8 to 12
14 by 17	3 to 5	10 to 14

The later setting on multicylinder engines is due to the use of the relief valve, by which injection is completed before the plunger is at high point. Injection is completed on all engines at approximately the same time.

Timing Fairbanks, Morse Fuel Pump.—To check the time of injection, set the No. 1 piston on the upper dead center as follows: Set the piston approximately on the upper dead center as indicated by the flywheel keyway. Remove the injection nozzle, and lower a rod through the opening thus exposed, until it rests on the end of the piston Mark the rod about 6 in. above the top of the combustion chamber. Turn the flywheel in the direction of rotation until the mark comes down flush with the top of the combustion chamber. Now mark the flywheel rim approxi-

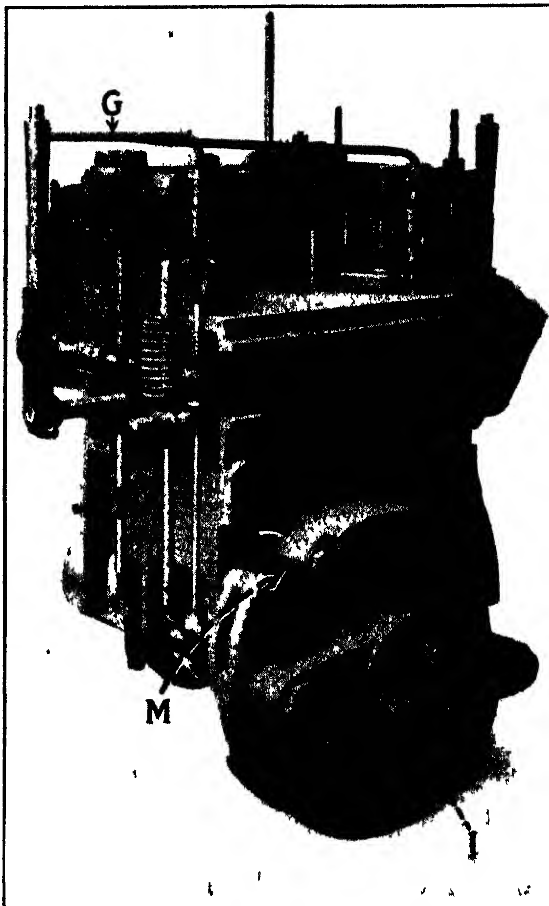


FIG. 261.—Fuel pump of Fairbanks, Morse engine, with section cut away

mately on its horizontal center line, and measure the distance from this mark to the floor. Next turn the flywheel backward, moving the piston up to upper dead center and down again, until the mark on the rod is once more flush with the top of the combustion chamber. Make a new mark on the flywheel rim, the same distance up from the floor as the first mark was made. Make a third mark halfway between marks 1 and 2; then turn the flywheel so as to bring this mark the same height above the floor as were marks 1 and 2 when made. Be sure to measure all marks from the same point on the floor to insure accuracy. This last setting gives the accurate upper dead center of the piston.

With the engine in this position, the long mark designated by the letter *C* on the governor spider should be directly under the governor-case timing pointer. If the cam has moved from this position, the engine will be correctly retimed when the spider is turned until the long mark *C* coincides with the pointer. The letter *O* on the spider indicates the high point of the cam.

Changing the injection-cam setting will seldom be necessary, as the factory setting covers a wide range of approved fuels. Very heavy fuels may require a slight change which can readily be made as follows: Remove the governor case end plate, and note the position of the governor spider indicated by the graduated dial and pointer. Then loosen the three nuts on the dial, and move the spider, but not more than one degree, in the direction of the engine rotation to secure earlier injection. Note the effect of the changing in the performance of the engine before making further adjustment.

In addition, the time of closing is affected by the amount of clearance between the upper pushrod stem and the suction valve, measured with the cam in low position. With a small clearance, the suction valve will be farther off its seat and will close later. With a large clearance, the opposite is true. Thus, if less fuel is to be injected into the cylinder, the clearance must be decreased; and if more fuel is to injected into the cylinders, the clearance must be increased.

The clearance between the suction valve and pushrod should be from 0.010 to 0.025 in. with the governor cam in low position. To measure the clearance, remove the pump-case-housing cover, the fuel-reservoir cover, and the injection-pump discharge valve *C* with its spring. With the governor cam in low position, hold the suction valve down on its seat, and lift the upper pushrod stem against the valve. Then insert the thickness gage between the pushrod stem and the adjusting screw.

This clearance is adjusted correctly at the factory, and the adjusting screw for No. 1 cylinder is sealed. On single-cylinder engines no further adjustment should be made, but on the multi-cylinder engines occasional adjustment is required to keep the load balanced on all cylinders. Adjustment should always be made on the unsealed pushrods. In this way, No. 1 cylinder is used as the key cylinder, and all other cylinders must be adjusted to it.

The adjustment is made by either lengthening or shortening the pushrod. Lengthening it results in decreased clearance; the suction valve closes later, and the amount of fuel injected is decreased. To make the adjustment, loosen the adjusting screw nut, and then, while holding the pushrod with a stiff wire or nail inserted through the hole in the pushrod, turn the adjusting screw in the desired direction. Start the engine and observe the exhaust temperatures under full load. If one of the cylinders has more load than the others, its pyrometer reading will be higher. To balance the load, adjust as above, until the pyrometer indicates exhaust temperatures all within a limit of 30°F.

The compression pressure is dependent upon the clearance volume, while the firing pressure is dependent on the load and on the time of injection. These pressures should be checked periodically in order to take care of slow changes in adjustment due to wear. An indicator should be used to take the pressures, an opening being provided in the combustion chamber for attaching it. This opening is closed with a pipe plug when the indicator is not in use. The pressure should be as follows:

Engine	Compression (hot) pressure, lb.	Firing pressure, lb.
Single cylinder.	520–540	600
Multicylinder......................	490–510	600

Fairbanks, Morse Type M Fuel Pump.—The fuel pump used on the 16 by 20 and larger Fairbanks, Morse Diesels is one with a constant-stroke plunger and a suction valve where closure is under governor control, through the eccentric fulcrum. The details are shown in Fig. 262.

In addition, the air-starting valve *A* is shown. To time the pump or change the gears, place the No. 1 piston on its upper dead center. (No. 1 piston is next to the scavenging-air cylinder.)

This brings the drive-gear keyway on the upper side of the crankshaft on its vertical center line.

Mesh the drive, intermediate, and camshaft gears so that the letter *O* on the drive gear comes between the two letters *O* on the intermediate gear, and, while these gears are so meshed, set the camshaft gear so that the letter *O* on the opposite side of

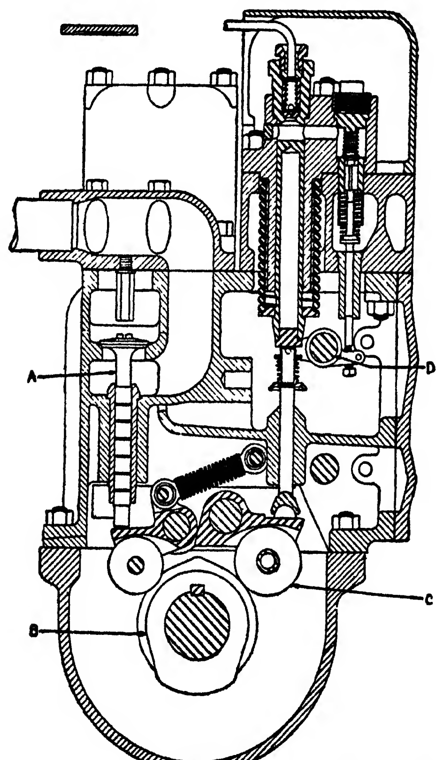


FIG. 262.—Fuel pump used in larger Fairbanks, Morse engines.

the intermediate gear meshes between the two letters *O* on the camshaft gear. When all the gears are meshed as described, the injection-pump plunger should be at the upper end of its stroke, and the prick-punch mark on the movable cam gear at the center of its adjustment.

Turn the camshaft gear hub until its prick-punch mark coincides with the long chisel mark on the camshaft gear; then clamp the gear to its hub by means of the six bolts in the gear. With the two marks coinciding, the end of injection should occur when the crankshaft is 4 to 6 deg. past its upper dead center and the injection-pump plunger is at the upper end of its stroke.

The fuel injection is timed at the factory to give the best running conditions and is correct for a considerable range of fuels. Therefore it seldom will be necessary to make any change in the injection setting; but should the use of an unusual grade of fuel necessitate an adjustment, only a competent mechanic, thoroughly familiar with the operation of the engine, should undertake to make the change.

Under normal conditions, it is necessary to inspect and regrind the injection-pump valves from time to time, depending upon the condition of service and the fuel used. When the valves become leaky, grind them into their seats, using a fine grade of carborundum paste, flour of glass, or pumice stone; but do not use emery, as even the finest grades are too coarse for this purpose. Some of the valves are provided with screw-driver slots; for cup-shaped valves a round stick may be fitted and used for rotating. After grinding make sure that absolutely no grit is left on the valve or valve seat, as the smallest particle might ruin the valve in operation. Be careful not to overgrind the valve, but grind only enough to secure a perfect seat.

Cummins Fuel Valve and Pump.—There is some question as to just how to classify the Cummins engines in respect to the combustion principle. As will be seen from Fig. 263, oil is delivered into a cup *B* through a passage *A* from the fuel pump. At the proper time the plunger is forced downward by a rocker, and the oil is jetted into the cylinder. This design has been criticized by many as being crude. Nevertheless, its performance is excellent, and it is one of the few high-speed engines that runs satisfactorily.

On the late high-speed engines of the automotive type the company uses the method shown in Fig. 264. To handle the oil a single-plunger pump is used for all the cylinders, operating in conjunction with the rotary distributor *C*. The latter successively connects the metering pump *B* with the injectors on the several cylinders in the proper firing order. As the same pump is used for all the injectors, there is no likelihood of their receiving unequal amounts. The pump is of the simple, cam-actuated, spring-returned type and works with a variable-ratio, cam-roller guide link on which the connecting rod of the pump is positioned by the accelerator pedal. At high power the rod is placed

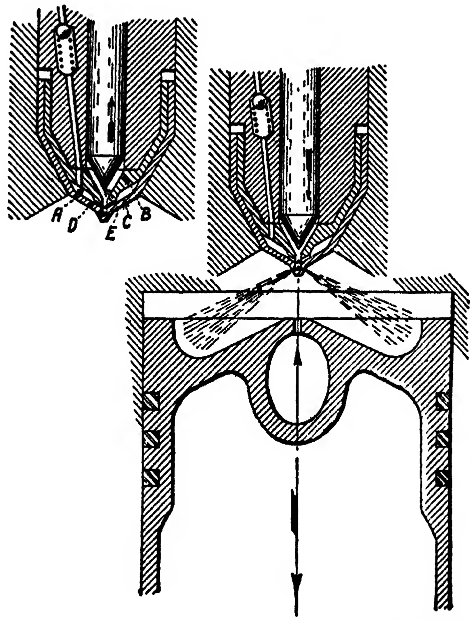


FIG. 263.—Injection device of the Cummins engine.

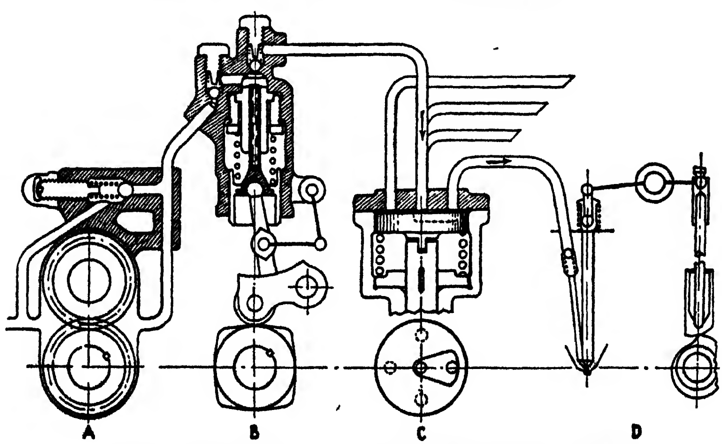


FIG. 264.—Fuel system of the Cummins automotive Diesel.

further to the left and transmits a greater proportion of the roller travel to the plunger. There are, of course, as many cam noses as there are cylinders, and all are ground to a master cam in such a manner as to rule out measurable variations in the lifts. The distributor is of the simple disk type with a central opening communicating with the pump discharge. The cavity in the rotating disk is always open to the pump delivery and connects the latter to the lines leading to the various injectors in sequence. Its arrangement is different from that of the high-pressure distributors sometimes used for direct-injection pumps. The latter must be so arranged that the pump pressure urges together the valve faces, which are, therefore, subject to heavy forces

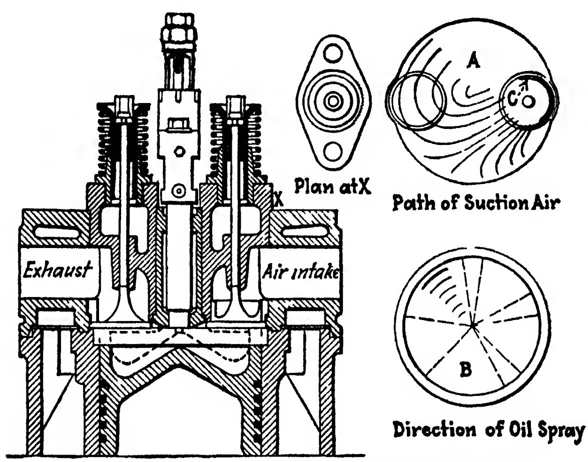


FIG. 265.—The Hesselmann solid-injection system.

and an amount of friction that sooner or later leads to cutting and destruction of the surfaces. The Cummins distributor valve, on the other hand, is held to its seat by spring pressure only and readily lifts away from it when abnormal conditions cause an excessive pressure rise. Both the regular idling of the engine in the automobile and its generally smooth action under all conditions of slipping the clutch and changing of the gears are in a large measure attributable to the good distribution inherent in this fuel-supply system.

A low-pressure fuel-supply pump is indicated at *A*. It further adds to the positive action of the fuel system by relieving the metering pump of suction duty. It will be noted that a spring-loaded by-pass is provided and that it is set to maintain a uniform pressure of 50 lb. per square inch. The effect of the supply pump is particularly evident when the engine starts from cold.

Busch-Sulzer Solid-injection Spray Valve.—The Busch-Sulzer Brothers Diesel Engine Company, late in 1930, acquired an A.E.G. license to build solid-injection Diesels with the Hesselmann combustion system. The engine in all other respect is the Busch-Sulzer, two-cycle, crosshead-type Diesel.

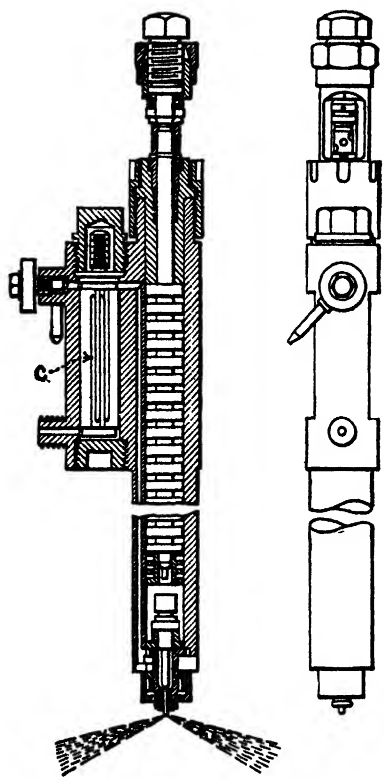


Fig. 266.—Spray valve of the Hesselmann system, including an oil filter *c*.

As shown in Fig. 265, the Hesselmann system includes a piston crown so shaped that the oil spray issuing from the nozzle comes into contact with the entire air charge. The nozzle is a differential needle valve with a multi-hole orifice. The spring of the valve is composed of a series of light washers and rings so assembled that each washer, or disk, acts as a diaphragm spring. The inertia of the apparatus is exceedingly small. The spray valve is shown in Fig. 266. The Busch-Sulzer Company has sold several 10-cylinder, 3,000-hp., 240-r.p.m. and 4,000-hp., six-cylinder 30 by 42-in., 125-r.p.m. units of this type, these being the largest solid-injection engines sold in this country.

Busch-Sulzer Fuel Pump.—Several designs of plunger-type pumps control the quantity of fuel discharged to the injection nozzle by holding a suction valve open during a portion of the plunger's discharge stroke. This permits fuel to by-pass back through the suction port instead of discharging to the nozzle and combustion chamber. The Busch-Sulzer fuel pump (Fig. 267) is typical of this type. An individual cam-operated plunger serves each engine cylinder. Fuel is drawn in by the pump during the plunger's downward stroke through the cam-operated suction valve *S*. It is discharged through the valve *D* directly above the pump bore. This valve is held closed by a light spring, its purpose being to prevent oil from flowing from the injection-nozzle piping back to the pump during its suction stroke.

The tappet *E* directly under the suction valve is arranged so that, when raised by the lever *F*, it will hold open the suction valve *S*. The lever *F* is fulcrumed on eccentric *G*, and its ball-shaped end is held in a yoke formed in the pump plunger. Thus, this lever moves up and down with each stroke of the plunger *P*. Position of the eccentric *G* is controlled by the lever *H*, which is connected to the governor.

When lever *H* is moved to the extreme left, the center of the eccentric *G* is raised to such an extent that lever *F* raises tappet *E* and holds the suction valve *S* off its seat during the entire upward stroke of the plunger. Thus, no fuel is delivered from the pump. When lever *H* is shifted to the extreme right, which is the full-load position, the center of *G*, about which *F* rotates, is lowered so that lever *F* does not contact the suction valve tappet *E* until the plunger *P* has completed more than half of its upward stroke. During this period the suction valve is held closed by its spring. Thus, when the plunger starts on its upward stroke, fuel is trapped in the space above it, and its only escape is through the discharge valve *D* and piping to the injection nozzle. Injection starts with the upward stroke of the plunger and continues until lever *F* raises tappet *E* and the suction valve off its seat. Pressure is immediately released, and fuel flows back through the suction valve for the remainder of the plunger's upward stroke. Intermediate positions of the lever *H*, as at the left of Fig. 267, hold the suction valve open for a larger portion of the plunger's stroke; injection takes place during a shorter portion

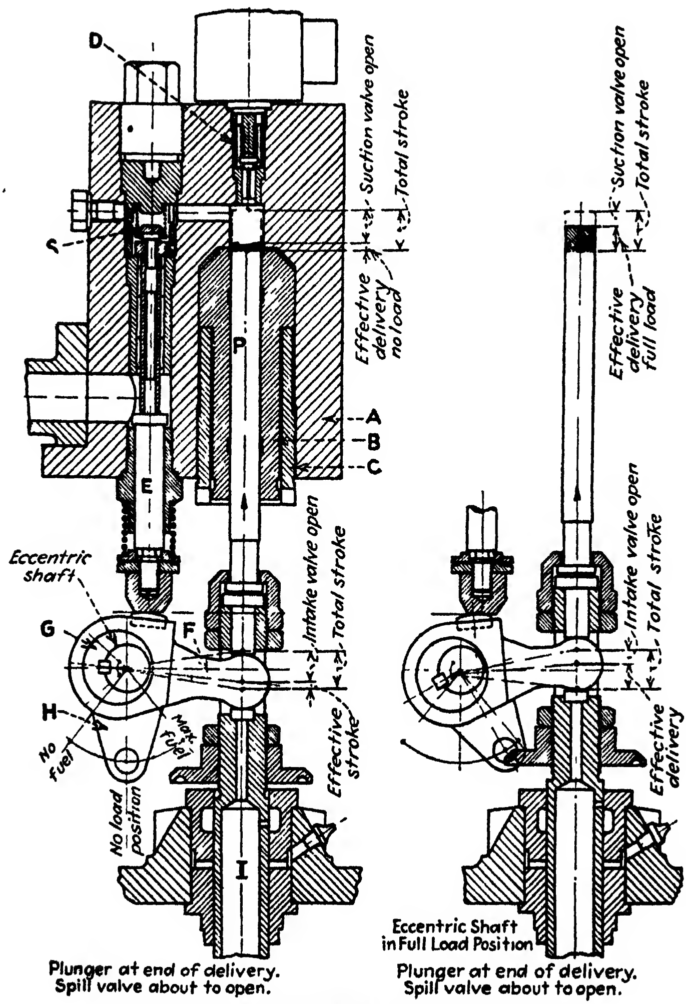


FIG. 267.—Busch-Sulzer fuel-injection pump.

of the stroke, and less fuel is discharged into the engine's combustion chamber.

General Motors Injection System.—On the General Motors two-cycle, high-speed Diesel a combined fuel pump and spray valve (Fig. 268) is used. The plunger *P* is operated by a rocker whose roller contacts with a cam. As will be seen from the drawing (Fig. 268), the end of the plunger *P* carries two scrolls for fuel-discharge control. The rack *R* meshes with a pinion which

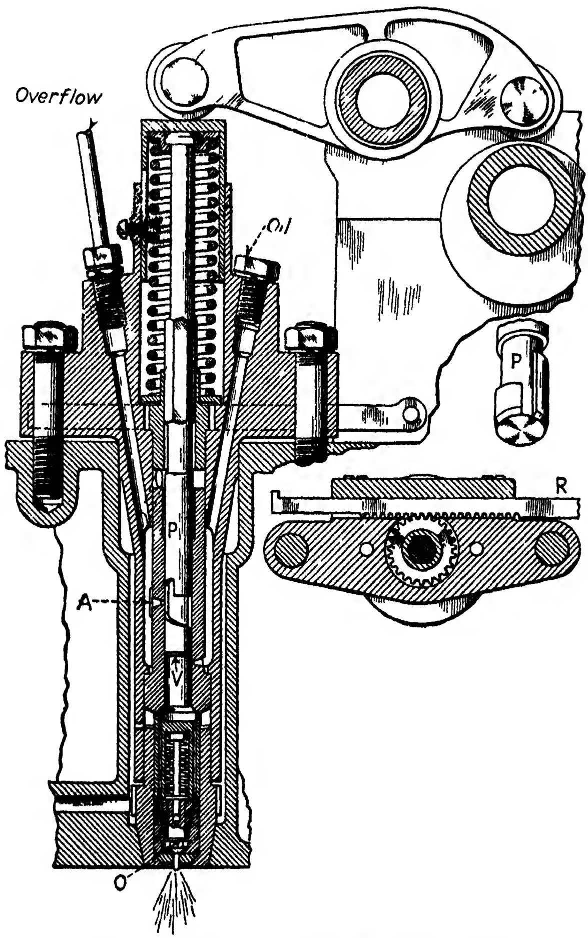


FIG. 268 —Diagram of General Motors fuel system

is a part of a bushing. This bushing has a hexagonal bore in which the hexagonal shank of plunger *P* slides. As a result, the motion of the control rack *R* rotates the plunger through an angle at a change in engine load, with the result that the helix on the plunger uncovers the suction port earlier or later, thereby changing the point in the plunger's discharge stroke at which the oil can by-pass back through the suction port.

The spray valve is spring-loaded and is the only valve in the entire injection equipment.

The plunger has two motions, a reciprocating motion to pump the fuel, and a governor-controlled angular motion to vary the amount of fuel according to the load. The sectional diagrams of

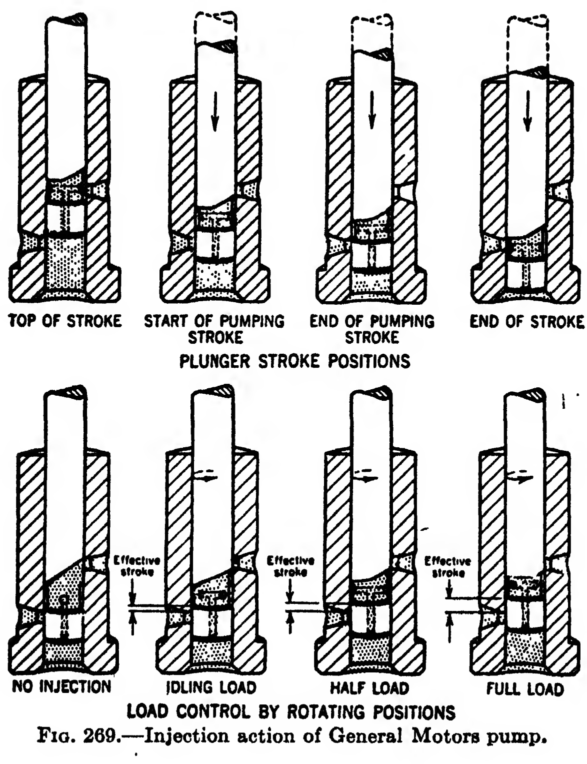


FIG. 269.—Injection action of General Motors pump.

Fig. 269 indicate these motions. The upper diagrams show the reciprocating, or plunger-stroke, positions, and the lower diagrams show the plunger rotary, or load-control, positions.

The upper series of diagrams of Fig. 269 show in detail how the start and end of the pumping stroke are controlled by the upper and lower lips on the plunger. These diagrams make it evident that the total downward stroke of the plunger is divided into three parts. The first, or upper, part of the stroke allows the plunger to be accelerated to the required velocity before pumping begins. The second, or pumping, part of the stroke controls the quantity of fuel injected. The third, or overtravel, part of the stroke slows down the plunger velocity to the end of the stroke.

The illustrations show that the pumping stroke is controlled by the upper and lower helix on the plunger passing over the ports. The lower set of diagrams in Fig. 269 shows how the distance from the lower helix to the lower port is varied by the rotation of the plunger and the resulting variation in the effective pumping stroke.

The pumping part of the stroke begins when the upper helix on the plunger covers the lower edge of the upper port *A*. Rotating the plunger, so as to increase the effective pumping stroke, causes injection to start earlier during the cycle. When the load is increased, the plunger rotation automatically advances the beginning of injection.

Fuel enters the annular space around the spray-valve seat; and when the fuel pressure, acting on the differential area of the valve needle, has been built up high enough to compress the spring, the needle is lifted from its seat. Fuel is admitted to the spray tip *O* and is forced through the spray orifices into the cylinder. Owing to the high velocity of the plunger at the full speed of the engines, and because of the restrictive action of the small spray holes, the final fuel-oil pressure is built up to a much higher value than the pressure at which the needle valve opens.

Deco Fuel-injection Pump.—The Deco fuel-injection system employs an individual, cam-operated, variable-stroke plunger pump for each cylinder. These pumps are mounted in a single-housing body, a cross section of which is shown in Fig. 270. Each pump consists of a barrel and plunger, tappet assembly, control rod, and suction and discharge valves.

The barrel fits securely in the body, as shown in Fig. 270. Directly above the barrel is the suction-valve body. The lower portion of the valve body is accurately ground as a valve seat, and the upper portion acts as a guide for the suction valve. A small spring insures a quick closure of the valves when the suction force is released. Several small holes drilled in the suction-valve body, just above the seat, permit fuel to pass from the fuel manifold to the valve. The entire valve assembly is held securely in position by the sleeve nut, which is screwed into the pump head.

The plunger and barrel, which are located directly beneath the suction-valve assembly, are accurately ground to secure a perfect fit of the plungers in the barrels. The lower end of the plunger is in contact with the tappet-adjusting screw located in the upper end of the tappet-body assembly. The lower end of the tappet assembly is provided with a roller which engages

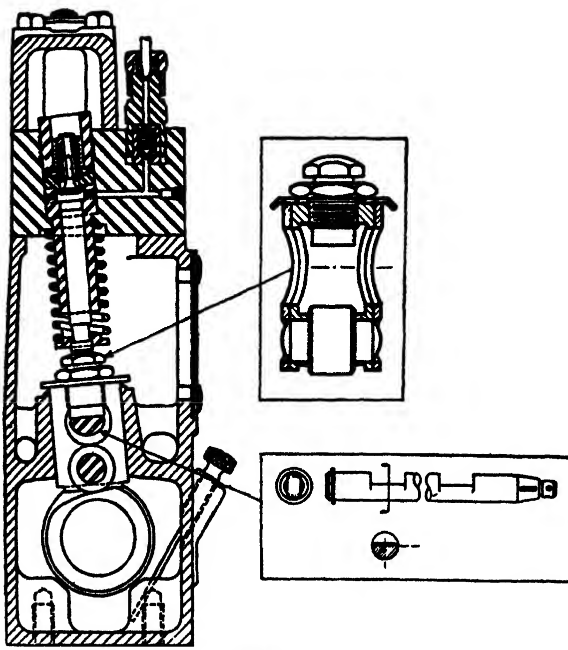


FIG. 270.—Cross section of Deco pump.

the camshaft, and the entire tappet assembly is installed with a sliding fit in the pump housing and rests on the flat cross section of the control rod. It is held in place by the heavy plunger spring.

The control rod, which extends the entire length of the pump housing, passes through all of the tappet bodies between the rollers and tappet screws. The section of the control rod within the tappet bodies is machined flat, and its rotation controls the plunger stroke and, therefore, the amount of the fuel handled by the pump.

Deco-pump Action.—When the plunger moves downward, it creates a suction that opens the inlet valve, and the barrel fills with fuel. When the plunger reaches its lowest position, it remains stationary for a moment, thus relieving the suction, and the spring closes the suction valve before the plunger starts upward. As the plunger starts upward on its delivery stroke, the pressure of the fuel within the barrel is increased, which lifts the discharge valve. Fuel then flows into the injection line to the injector. The injector line carries the fuel to the spray

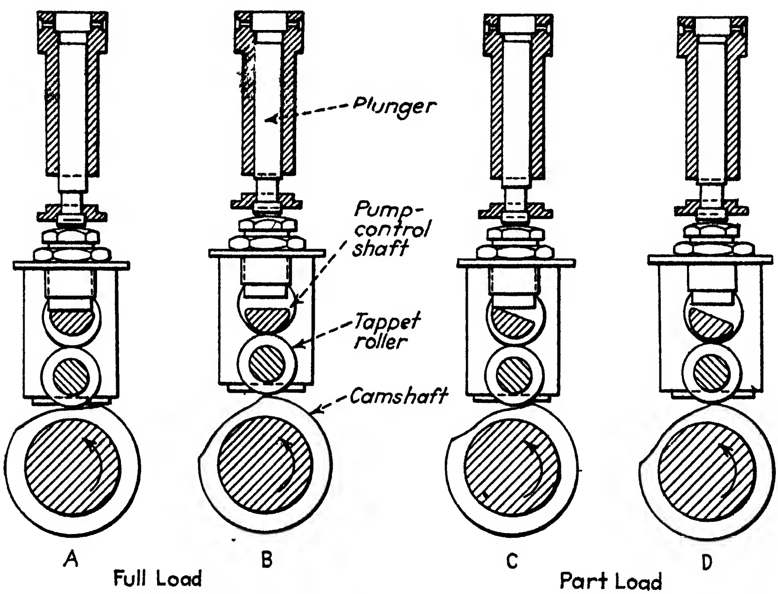


Fig. 271.—Deco pump action.

nozzle located in the cylinder head of the engine through which it is sprayed into the combustion chamber.

As soon as the plunger reaches the end of its upward stroke, the discharge-line pressure falls below the injector closing pressure, terminating injection.

The amount of fuel injected by the pump is metered by varying the length of the plunger stroke. Rotation of the control rod a few degrees causes the high edge of the control rod to act on the adjusting screw and thus raise the entire tappet assembly. The effect of this movement is to decrease the stroke of the plunger, because the roller will be raised and, therefore, will be contacted by the cam at a higher point on the cam. By rotating the control rod to the maximum, the roller in the tappet assembly may be raised so that it will never be contacted by the cam. In this position the pump becomes ineffective, which serves to stop the injection. The action for full load and part load is shown in Fig. 271.

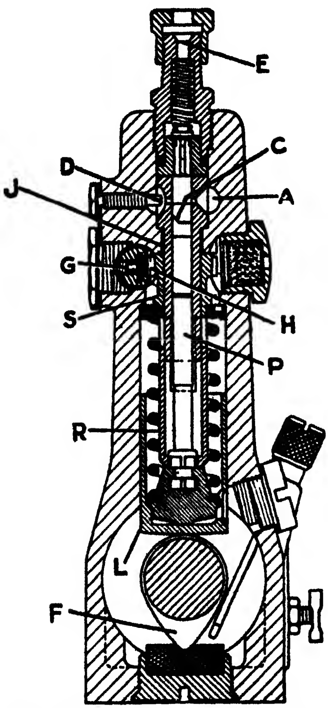


FIG. 272.—Cross section of Timken fuel pump.

Timken Injection Pump.—As will be seen from the illustrations, the Timken pump is of the cam-operated, helical-plunger type. When the plunger is in its lowest position, as shown in Fig. 272, the cylinder receives a charge of fuel through the suction port *A* from a feed header extending the length of the pump housing. This header is supplied with filtered oil by a transfer pump. Delivery of fuel to the injection nozzle starts as soon as the piston *P* covers the inlet port *A* and ends when the upper helical edge *C* of the annular groove in the piston opens the overflow, or by-pass, port *D*, releasing pressure in the discharge line *E*. The effective delivery stroke of the plunger is regulated by rotating the latter in its cylinder, thus varying the point in the stroke at which the by-pass port is uncovered. This action is similar to the Bosch pump action already discussed.

The pump design is such that by removing two nuts any of the pump assemblies can be removed from the housing for repair or replacement without disturbing any of the other pumps. This is an outstanding feature of this pump. The piston-control gear *H* has one wide tooth which must mesh with a similar gear root in the rack rod, making it impossible to replace a pump assembly in any but the correct position.

Another feature of this pump is that the pistons are driven by constant-velocity cams *F*. Thus, the delivery rate of fuel entering the combustion chamber is maintained constant at a speed adapted to the rate of combustion.

Fulton Pump and Fuel Valve.—The mechanical-injection four-cycle Fulton Diesel has the pump and spray valve shown in Figs. 273 to 275.

The fuel pump (Fig. 274) is mounted on the front end of the engine and is driven by a silent roller chain. The chain has hardened pins and rollers throughout, ground within very close limits. The pump body is made of open-hearth forging, with removable and hardened valve seats and hardened valves. Pump plungers and liners are ground and lapped into each other, but the liners are held to the pump body by gland nuts and can be removed easily. Valves, seats, plungers, and liners are all made of stainless steel. Each cylinder has its individual pump body, which can be conveniently and quickly removed without loosening any pipe connections with the exception of the small fuel pipe leading from the pump to the fuel-spray valve.

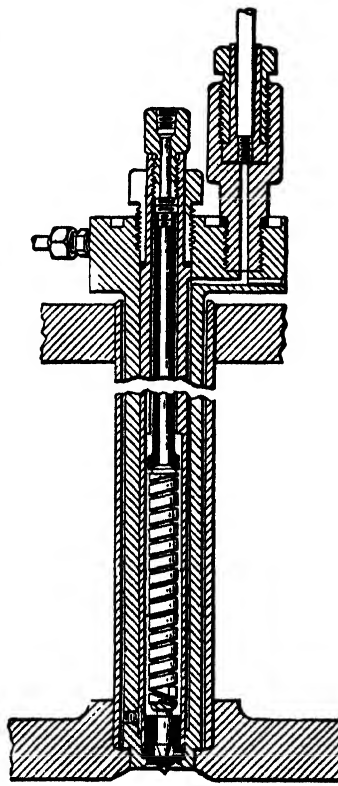


FIG. 273.—Section through Fulton spray valve, which consists, essentially, of a spring-loaded needle valve and a five-hole orifice.

Cams and cam rollers, both hardened and ground, drive the pump plungers. The motion from the cam roller to the plunger is effected in such a way that the latter cannot bind and stick. Surplus fuel oil furnished by the pump is by-passed by a valve which is under the control of the governor. The opening and closing of each by-pass valve can be adjusted individually, so that each cylinder gets its proper share of fuel and load. This adjustment is very simple and can be accomplished while the engine is running.

The pump housing has a horizontal partition, the lower compartment being completely filled with lubricating oil, submerging the camshaft, bearings, cams, cam rolls, and their guide. This not only secures effective lubrication of the moving parts but aims to prevent their heating and sticking, as the oil in the lower compartment of the pump housing is in connection with the forced-feed lubricating system and is continuously renewed.

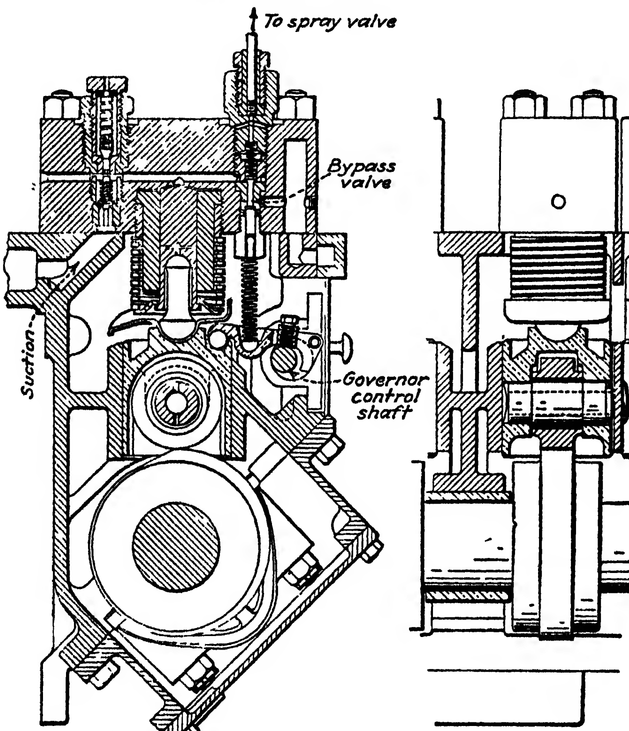


FIG. 274.—Cross section of Fulton fuel pump.

The upper part of the camshaft housing is designed so that any dripping or leakage from the fuel pump is caught and prevented from mixing with the lubricating oil of the pump.

The camshaft of the fuel pump is extended and at its forward end drives the rotary pump (Fig. 275) which supplies the main fuel pump. Fuel passes through a strainer having openings not exceeding 0.0035 in. before entering the fuel pumps. This prevents grit or other impurities from coming in contact with the plungers and valves.

Cooper-Bessemer Fuel System.—On some of its Diesels the Cooper-Bessemer Corporation employs a fuel system of its own design, which contains some interesting features. This consists of a duplex fuel-oil pressure pump with hardened and lapped plungers, an adjustable pressure-regulating valve and an atmospheric-relief timing valve for each cylinder, built in blocks of three or four. These timing valves relieve the fuel pressure to the atmosphere in the line to each nozzle after each injection. This eliminates any possibility of dribble at the nozzle and results

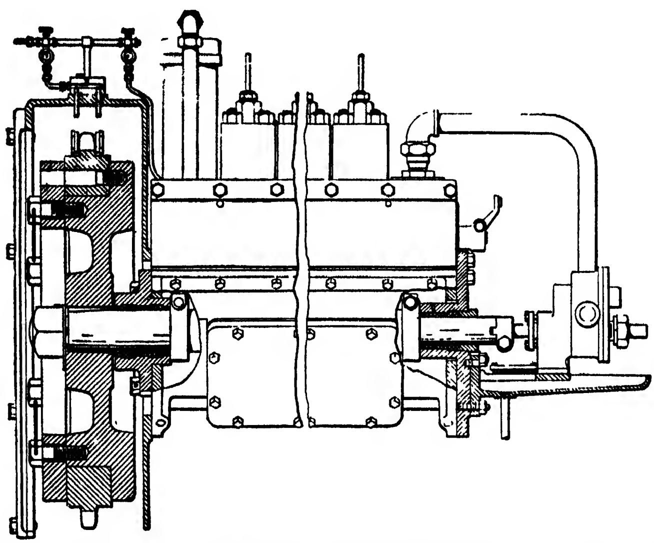


FIG. 275.—Fulton fuel-pump assembly, showing gear drive and rotary oil-lifting pump.

in unusually clean combustion. A separate shutoff valve for each cylinder is provided.

Diagram of Fuel System.—Figure 276 shows the fuel-handling system. Starting with the fuel in the storage or bunker tank, the transfer pump draws the fuel through the strainer from the storage tank and delivers it to the day tank. Excess fuel is returned to the storage tank through the overflow pipe in the event that the transfer pump continues to operate.

Fuel flows by gravity from the day tank through a duplex strainer to an untimed pressure pump capable of maintaining a working pressure up to 7,000 lb. per square inch. The desired pressure is controlled by a spring-loaded pressure regulator located in the discharge line from the pressure pump. The tension at the spring of the pressure regulator determines the pressure and is changed manually to suit the load or the speed. Excess pressure opens this regulator, allowing the fuel to return to the day tank, as shown in Fig. 276. Fuel at the desired pressure goes to the distributing system, which contains a pressure tank, or accumulator, and an isolating, or shutoff, valve for each cylinder. The purpose of the isolating valve is to shut off fuel from any cylinder, if for any reason it is desired to do so.

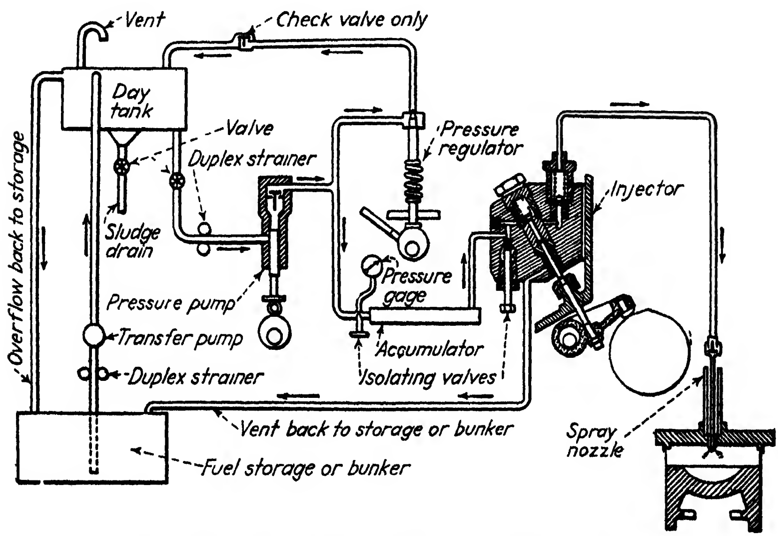


FIG. 276.—Diagram of Cooper-Bessemer fuel system.

Assuming that the isolating valve is open, the fuel goes to the injector valve, a poppet valve which remains closed most of the time, being opened by a cam at the proper time for injection fuel into its particular cylinder. When the cam, shown below the injector valve in Fig. 276, opens the injector valve, fuel flows through the tubing to the spray nozzle located in the engine-cylinder head. This spray nozzle is equipped with a spring-loaded valve which opens and closes at a predetermined pressure, usually set at 1,500 to 2,000 lb. per square inch. The fuel flows past this valve, then through small holes at high velocity into the combustion chamber, where it is ignited by the heat of compression and continues to burn as long as injection lasts.

Injection lasts as long as the cam holds the injector valve open, after which the needle valve in the fuel-spray nozzle closes, with a snap, thereby preventing after-drip of fuel into the combustion chamber. The amount of fuel entering the cylinder depends upon the engine load and speed and is regulated by varying the duration of the injector-valve opening, which also varies the injector-valve lift or travel. Moving the control one way decreases the clearance between the tappets and thereby increases the lift of the valve and the injection period. Moving the control in the opposite direction does the reverse; and if it is moved far enough, it will stop injection

Fuel-injector Valve.—The function of this valve is to measure and time the fuel to its respective cylinder. There is one valve per engine cylinder, and they are mounted in groups of three or four en bloc, depending upon the number of cylinders the engine has. Six- and eight-cylinder engines have two blocs, of three and four injector valves each, respectively. Each bloc has its own camshaft, driven by a gear, which is, in turn, driven from the engine camshaft by a gear.

Atlas-Imperial Fuel System.—The industrial and large Diesels manufactured by Atlas-Imperial Diesel Engine Company employ a common-rail fuel system. The arrangement is shown in Fig. 277. In this system a constant pressure is maintained on the fuel line, with provision for varying it, by means of a fuel-relief valve, to suit the conditions of operation. Oil is delivered from the fuel tank through a fuel filter by means of a low-pressure (sometimes called a "day-tank") pump to the high-pressure injection pumps. From the high-pressure fuel pumps the oil is led to a common rail, from which the system gets its name, with branches leading to the spray valves in each cylinder. A spring-loaded, adjustable relief valve, called the "fuel-pressure relief valve," is connected into the discharge side of the high-pressure fuel pumps, to by-pass any excessive oil over that required to maintain the constant pressure. The overflow from this relief valve is led back to the main fuel tanks.

Late models of engines are equipped with a gear-type pump driven from the crankshaft. These pumps require little, if any, attention, except to keep the packing glands reasonably tight to avoid leakage. Screwing the packing gland excessively will cause an undue amount of wear on the pump shaft and should be avoided. The discharge from the low-pressure fuel pump is connected with the fuel filter.

The fuel oil is delivered to the suction valve of the high-pressure pump, from the fuel filter. The high-pressure pump discharges into the fuel rail, maintaining a pressure as determined by the setting of the high-pressure relief valve. Under operating conditions this pressure will vary from 3,500 to 5,000 lb. per square inch. For proper functioning the valve in the high-pressure fuel pump should be kept well seated by grinding when necessary.

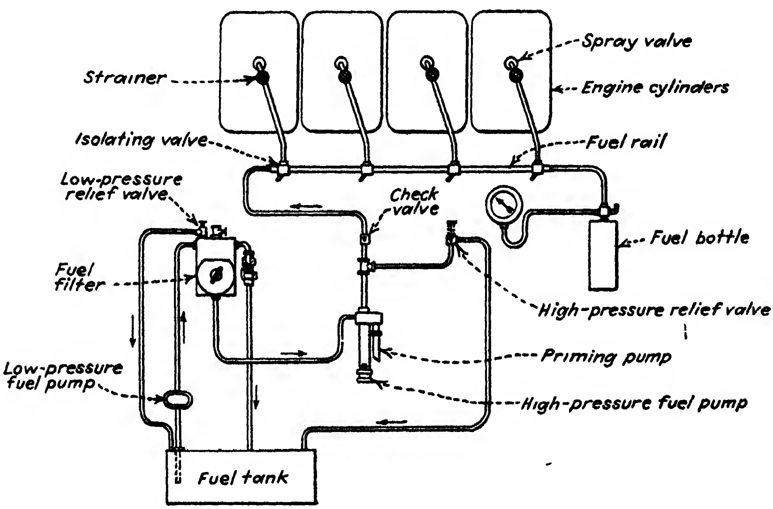


FIG. 277.—Diagram of Atlas-Imperial fuel system.

These pumps, of which there are two, are detachable from the pump block. The plungers are lapped to a precision fit in the barrels and use no packing. The pump block is held in place by three nuts, two on top of the block and one on a stud, through a cross bar inside the pump housing. When removing the pump block for any reason, first close the valve in the pump suction line at the day tank to prevent draining by the day tank.

The fuel-relief valve automatically maintains the pressure on the fuel rail by by-passing the surplus fuel pumped by the high-pressure fuel pumps. The valve and stem are removable for regrinding or replacement, and it is essential that they be kept tight for proper functioning of the engine.

This valve regulates the pressure maintained to the spray valves. Use between 1,500 and 5,000 lb., as the load requires.

Fuel-spray Valve.—The function of the spray valve is to admit the fuel to the cylinders at the proper time and at the same time deliver the fuel oil in the form of a very finely atomized spray so that the fuel is readily ignited and properly burned within the cylinder. The spray valve is located in the center of the cylinder head and extends through the head, with the spray-valve tip projecting slightly below the bottom of the cylinder head. It is provided with a needle valve held in position by a strong spring, the tension of which is slightly adjustable (see Fig. 248). The fuel oil is admitted below the stuffing box at a pressure anywhere from 1,500 to 5,000 lb. per square inch (depending on the adjustment of the pressure-regulating valve). The spray valve is held in position by a clamp arrangement having a single stud; this makes it easily removable for inspection or cleaning. It is operated by a lifting rocker arm which lifts the valve only a few hundredths of an inch when the engine is pulling its full load. The governor controls the lift of these valves by shifting a wedge between the two sections of the valve pushrod, so that when the engine is running idle the lift is very small and only sufficient to admit enough fuel to turn over the engine. As the load increases, the engine governor causes these valves to be lifted higher and consequently additional fuel is admitted in direct proportion to the power developed.

Spray Tips.—The size of spray tips varies with the class of service, with maximum engine speed, etc., and it may be necessary to change tips after the engine is put into service. The size tips with which the engine is equipped when it leaves the factory is marked on the engine-timing name plate and is usually as follows:

For all 6-in. bore engines, use five 0.008-in. tips.
For all 7-in. bore engines, use five 0.008-in. tips.
For all 8-in. bore engines, use five 0.009-in. tips.
For all 9-in. bore engines, use five 0.010-in. tips.

Conclusion.—It should be understood that the fuel systems covered in this chapter are not all the fuel-injection devices on the market. They do include, however, all the basic designs.

CHAPTER XVII

LUBRICATION

General.—With the high pressures, both cylinder-gas and unit-bearing pressures, lubrication of the reciprocating and rotating parts of a Diesel is one of the serious problems of the engine designer, operator, and oil refiner.

It must be confessed that the factors that determine the success or failure of a specific oil are still open to argument; most refinery men set chief dependence upon viscosity, which in turn is indicative of the body of the oil, or its ability to maintain a film under working conditions. Another characteristic, the *oiliness* of the oil, is now receiving some attention, and it is true that an oil slightly compounded to increase this characteristic seems to give best results. Another factor is the tendency of the oil to break down and form a residue.

Before discussing oil, however, it is advisable to discuss the system of lubrication.

Lubricating Systems.—Diesel engines usually employ two distinct systems for the lubrication of the cylinders and bearings. These may be classified as shown in the tabulation.

Cylinder Lubrication	Bearing Lubrication
Oil fog	Pressure feed Ring oiling Splash Mechanical force feed
Mechanical force feed	Mechanical force feed Gravity feed Pressure feed Ring oiling

The oil-fog system, created by the throwing of oil from the crankpin bearing, to lubricate the cylinder walls, is found on small engines of 14 in. and less cylinder diameter. It is obvious that with large cylinder diameters, the oil mist, or fog, so created would not be positive enough. In fact even for small engines with ring-oiling bearings it is satisfactory only when an oil level is carried in the crankcase sufficiently high to permit the crankpin bearing to scoop up oil to fling up into the cylinder.

For positively coating the cylinder walls with an oil film a mechanical force-feed lubrication is necessary. Formerly, it was thought desirable to employ a timed pump and a closed oil nozzle to inject the oil at a fixed time when the piston on its upstroke covered the lubricating-oil opening. This may have been possible with a slow-speed engine, but with engines turning at 200 r.p.m. and above, the lubricating-oil feed cannot be timed accurately. The better plan is to have the oil feed placed low enough on the cylinder wall to be sealed by a few piston rings. This will prevent the high gas temperature from reaching the oil film.

It should be understood at once that any oil deposited on the extreme upper part of the liner by the wiping action of the piston will burn when exposed to the flames of combustion. The oil on the walls below this part will remain as a film, but the piston by its wiping action must serve as a swab to place the oil on the cylinder walls. The oil enters in drops and starts flowing downward until the piston picks it up.

An engineer should bear in mind that the mechanical oil pump is subjected to stoppage of one or more oil lines due to the presence of bits of waste. Consequently, as much attention must be given to the pump as to other parts of the engine.

It is of the greatest importance that, having installed a mechanical lubricator, the operator should be certain that the lubricator employed is capable of delivering the oil at the proper time and in the correct amount for long periods of time and over a wide range of load variations and temperatures. It is also essential that the lubricator should be capable of a wide range of adjustment, enabling the feeds to be cut down to a very low point without stopping the flow. There are many designs of mechanical lubricators, which, although very compact, leave much to be desired with regard to accessibility of the various parts.

Some forms of sight feeds show oil that is *not* going to the engine, whereas what is wanted is to see the actual oil and the amount that is actually being fed.

The best type of machine is undoubtedly a pump that depends entirely for the amount of oil discharged on the displacement of the ram, the pump chamber being always completely flooded.

The action of the lubricator should be independent of the viscosity of the oil and of the head level of the oil in the tank, and, being independent of viscosity, it will then be independent of changes in temperature within wide limits. An efficient strainer should always be supplied regardless of the type of lubricator.

Cylinder Lubrication.—Considering first the lubrication of the power cylinder, the lubricating oil must do two things: lubricate the cylinder walls, piston, and piston rings; and assist the piston rings to form a seal. As the greater proportion of power-plant engines operate on the four-stroke cycle, let us first investigate what happens to the oil on each of these four strokes.

On the charging stroke, the cylinder walls, as far as the piston travels, have been covered with a film of oil from the previous stroke, which, together with the oil on the piston and between the rings, acts as a lubricant for the piston and rings. The oil in the clearance space between the rings assists them in forming a seal to prevent leakage of air or vapor from the crankcase into the cylinder.

The film of oil left on the cylinder walls during the suction stroke, together with the new film picked up by the piston from the cylinder walls, lubricates the piston on the compression stroke; and as the piston moves toward the cylinder head, it smears a fresh film of oil on the walls. The supply of oil collecting on the advancing side of the piston rings and around the upper edge of the piston serves to prevent leakage of the gases being compressed. The degree of compression will depend, therefore, somewhat upon the seal-forming properties of the oil and on the condition of the cylinder, piston, and rings.

During the compression stroke the temperature of the gases gradually increases but does not rise high enough to do any damage to the oil film until near the end of the stroke, when most of the oil film is covered by the piston.

The film of oil placed on the cylinder walls by the piston of the compression stroke lubricates the piston on the working stroke. As the piston moves toward the crank, exposing the walls to the high temperature of the burning gases, the flame comes in contact with the oil film but only after it has served its purpose of lubricating the piston on the instroke. The greater part of the damage to the oil film occurs on this stroke. The oil between the rings and between the piston and cylinder walls is subjected to the pressure of the burning gases and must assist the piston rings in preventing loss of power through leakage of the gases out of the open end of the cylinder.

If there is any trouble with lubrication, it will be encountered on the exhaust stroke, as the oil film on the cylinder walls has just been exposed to the high temperatures of the burning gases and has been more or less damaged. If the oil possesses the proper characteristics, however, and has been applied in sufficient amount, some lubricating value remains, which, together with the oil film on the piston itself, lubricates the piston for this stroke.

Although it is a difficult matter to make accurate determinations of the actual temperatures within the cylinders, estimates made by various authorities are close enough for our purpose. The maximum temperature obtained immediately after firing is about 2700°F., and the minimum temperature during suction stroke, 250°F. The average temperature for complete cycle is 950°F. It should be remembered that these are temperatures of the gases and not the temperatures of the cylinder walls or of the oil film.

The inner surface of the cylinder wall probably ranges from 100 to 300°F. hotter than the cooling water, depending on the thickness of the metal and the efficiency of the cooling system. As long as the water is not boiling, the cylinder-wall temperatures will hardly be more than 260 to 450°F.; and with a normal cooling-water temperature of 110 to 120°F., at which many power-plant engines operate, the cylinder-wall temperature is below 375°F. The oil film itself is hotter than this, at least on that side of the film exposed to the burning gases. The maximum temperature of 2700°F. occurs only when the piston moves toward the crank, uncovering the oil film and exposing it to the flame until near the end of the stroke. The inner surface of the oil film in contact with the cylinder wall at the comparatively low temperature of 175 to 400°F. is affected very little by the high temperatures, but the outer surface of the oil film directly exposed to the flame is undoubtedly damaged.

As it is impossible to produce a petroleum lubricating oil of any kind having a flash point over 700°F., it would seem that the oil film exposed to an average temperature of 950°F. would be destroyed promptly. That such is not the case the engineer can readily prove to himself by pouring a few drops of oil on a red-hot plate. Several seconds will elapse before the oil is completely evaporated. If the engine is running 200 r.p.m., the oil film will be replaced three times a second. So it is a comparatively simple matter to maintain the oil film if oils of suitable characteristics are used.

Influence of Completeness and Rapidity of Combustion.—The completeness of combustion has a great influence on the kind of lubricant to be used and on its life. If combustion is completed very soon after the fuel-spray valve closes, maximum temperatures are obtained when the piston covers the working surfaces of the cylinder, thus protecting the lubricating film from destruction. If the fuel is not completely vaporized and intimately mixed with air in the right proportions, it burns slowly; and as the piston uncovers the oil film, the burning fuel destroys it. An excessively lean mixture, such as exists on low loads by reason of the small amount of fuel injected into the cylinder, will allow, by reason of the large excess of air, the oil film to be completely burned up or at least so badly coked as to form carbon deposits. The bad effects of slow burning can be overcome sometimes by using a heavy-bodied oil.

Piston Seal.—When there is a film of oil on the cylinder, if it is of the right thickness and the oil possesses the proper characteristics, the friction of a tight-fitting ring can be lessened, and the seal of the loose-fitting ring improved. Taking as an example the vertical cylinder of a four-stroke-cycle engine, as the piston moves downward on the suction stroke, the lower edge of the ring scrapes a portion of the oil off the cylinder wall, forming a considerable body of oil on the advancing side of the ring. This oil "wave" reduces the passage of air and oil vapor from the crankcase into the cylinder past the rings and allows the piston to draw in a full charge of air.

On the compression stroke the oil builds up on the top edge of the ring and so prevents the blowing of the compressed vapor past the rings. As the piston is forced down on the working stroke, the pressure is exerted upon the oil resting on the top side of the ring. If there is only one ring, most of this oil would probably be blown out of the piston clearance space by the high-pressure gases; but when there are two or more rings, any oil blown away from the top ring packs up against the second ring and tends to prevent the further passage of the gases.

Another place for leakage is through the clearance space back of the rings, between the ring and its groove. If the ring is a loose fit in the groove, particularly with respect to width, the vapor will find its way around it, and power will be lost. The oil is of benefit here, too, as it fills this clearance space, and its resistance helps to prevent the gas leakage. The value of the oil in assisting the rings is particularly noticeable where the cylinder is not perfectly round, so that it is impossible for the rings to prevent leakage of gases without the assistance of the oil.

Effect of Viscosity.—The physical properties of the oil that influence its seal-forming value are viscosity, surface tension with respect to metal, and cohesion. There may also be some chemical reaction between certain constituents of some kinds of oils and the metal of the cylinder and the piston which increases its adhesion to the metal and prevents the oil from being scraped off.

Other things being equal, one might expect that a high viscosity would cause the oil to offer a greater resistance to being blown out of the clearance space and would also permit the accumulation of a larger body of oil on the advancing side of the rings than could be secured by a lower viscosity oil. It is found by actual experience that the piston seal can be greatly improved in many instances by raising the viscosity of the oil; better compression is secured and more power developed by the engine at a lower consumption of fuel per horsepower. This is especially true where the piston, cylinder, and rings have become worn.

From this statement the conclusion might be drawn that if a high-viscosity oil is good, a higher viscosity would be better. This is not true, as it is sometimes found that if the viscosity is too high the oil will not work its way between the piston and the cylinder fast enough, the cylinder becomes dry, and many troubles follow the lack of sufficient oil.

Just what the viscosity should be depends upon the design of the engine. In general, a viscosity of 500 to 700 sec. Saybolt viscometer at 100°F. covers the oils ordinarily used.

Should the viscosity be a little low, there is not likely to be lack of lubrication but simply loss of power through leakage and troubles from excessive oil in the cylinder and carbon deposits which gum the piston rings. The oil must be of just the right viscosity; and since the viscosity varies with the temperature, it must have the right viscosity at the working temperature of the oil film. Some engines run hotter than others and need a heavy oil. Others run cool, and a viscous oil could not feed fast enough to give proper lubrication.

An oil with a low surface tension with respect to the metals of which the cylinder and piston are composed will adhere more tightly to them than an oil with a higher surface tension; in other words, its adhesiveness will be greater. A high adhesive quality will also prevent the oil from being blown off the cylinder wall and out of the clearance space and therefore will improve the piston seal. Unfortunately, it is a difficult matter at present to measure the adhesion of oil with respect to different metals and to depend entirely upon actual running tests to determine the value of the various lubricants.

Bearing Lubrication.—Just how to lubricate the main and crankpin bearings is a problem to which various designers offer differing solutions. The three systems in general use are the splash, the gravity stream, and the pressure feed.

Stream Lubrication.—As has been mentioned, an engine having a mechanical oiler for the cylinders, as a rule, employs drop, stream, or pressure lubrication for the bearings. The stream system used to be preferred by some builders, since a sufficient amount of oil was thereby assured at all times, but is no longer used.

The lubrication of the crankpin is usually accomplished, where stream lubrication is used, by fitting banjo oil rings to one of the throws of each crank. A drilled passage in the crankpin leads the oil from the banjo to the crankpin bearing.

The piston pin, with such a system, is oiled in one of a number of ways. Some engines have the piston pin lubricated through a groove cut in the side of the piston with a drilled passage leading to the pin. Oil from a timed mechanical lubricator is forced onto the cylinder wall at the time that this groove registers with the oil-line nozzle.

Pressure-feed Lubrication System.—A pressure-feed system for crankshaft and connecting-rod bearings is employed on the Busch-Sulzer, Type B and most other Diesels. This system, as used on the Busch-Sulzer, is outlined in Fig. 278, showing the drilled crankshaft and connecting rods, as well as the pump and filtering mechanism. The oil is forced by the pump, which is a rotary geared to the air-compressor crank disk, into the lower half of the main bearings as indicated. It flows around the journal, lubricating the entire surface. The diagonally drilled passages to the crankpins once each revolution register with the oil-inlet opening in the lower bearing shell. This allows a stream of oil to enter the passage and to flow to the crankpin box. The connecting rod also is drilled, and at each revolution this passage

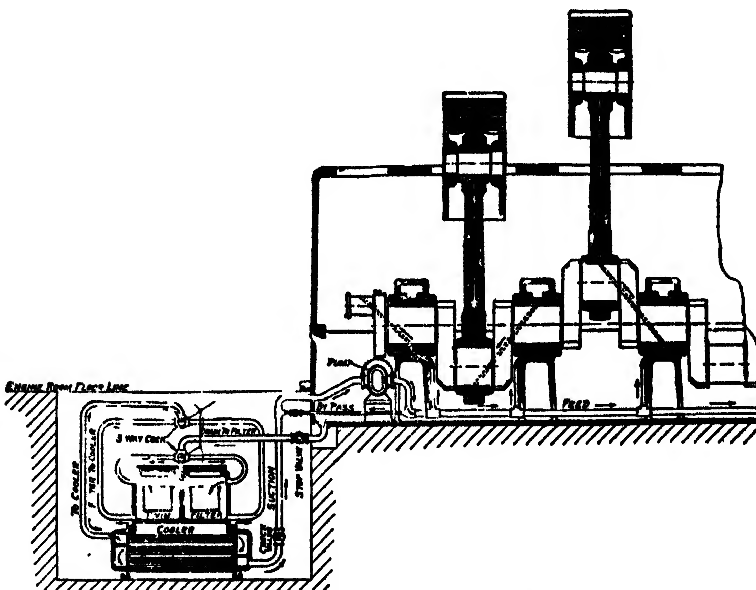


Fig. 278.—Busch-Sulzer lubrication system.

aligns with the crank end of the diagonal passage, which causes a supply of oil to pass up the rod to the wristpin bearing. The drip from the various parts is caught in the crankcase and from there flows through the filter into the cooler. The pump then draws the oil from the cooler to be reused. In order to seal the various parts of the system, the oil level in the crankcase should be around 3 in. before the engine is started and should always be carried at least 1 in. in depth while the engine is running. The pump discharge has a pressure gage attached; the pressure should average around 25 lb. The gage pressure is a fair indication of the condition of the system. If the pressure shows a decided

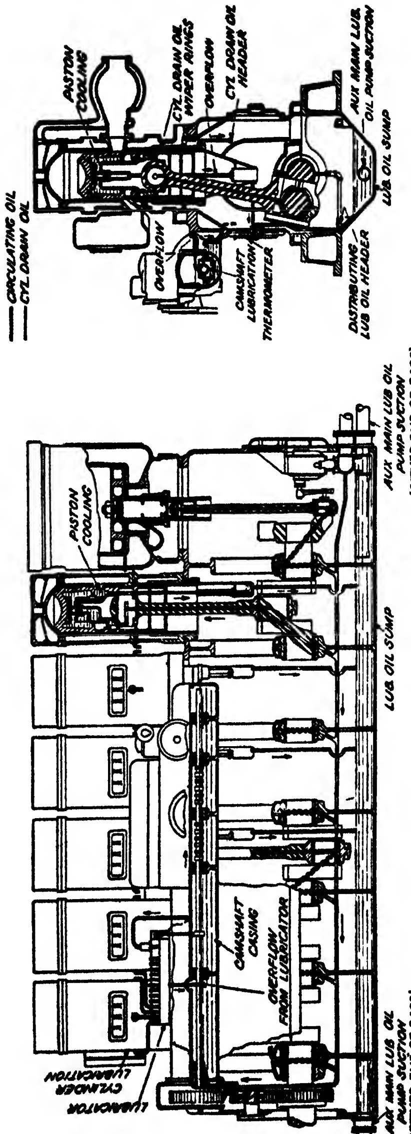


Fig. 279.—Combined pressure oiling and piston cooling system used on Fairbanks, Morse Diesel.

increase, it is probable that part of the discharge lines are clogged with dirt or waste.

The presence of water in the lubricating oil will cause the pressure to rise. Where the water is fresh or pure, the oil containing it should be drawn off, and the water settled out by heating. As to sea water getting into the lubrication system, every marine engineer avoids this as he would the plague; the effect of sea water is so serious that it is frequently cheapest to take no risks and entirely refill the system with new oil. Sometimes annoying leaks occur from the water-cooled exhaust valves, or cages, and the water finds its way into the oil system. Trouble also arises from leakage of water at the point where the oil passes to the pistons through the jacket, although such cases are not frequent. Where the pistons also are water cooled, a close lookout should be kept for leakage of water; and where the pistons are oil cooled internally, the oil selected should be of high quality. Care should also be exercised when dismantling the valves and cylinder covers, and when the water connections are broken, to ensure that water is excluded from the lubrication system.

Should there be a considerable fall of pressure, as shown by the gage, frequent examinations of the lubricants from the filters should be made; and if it is certain that the pumps are in order, make sure by observation and smell that no fuel oil has found its way to the crankcase or passed the pistons unburned.

Worn bearings will, likewise, cause a drop in pressure due to the escape of oil at the bearing ends.

Where the pistons are water cooled, at times water mixes with the oil in the crankcase; if in any considerable amount, there is danger of bearing trouble. An oil separator, such as the De Laval, will remove any entrained water. It should be connected between the crankcase and the oil filter.

In Fig. 279 is shown the pressure oiling system of the 16 by 20-in. Fairbanks, Morse Diesel. The pistons of this engine are cooled by a part of the oil feed.

Splash-oiling Systems.—The splash-oiling system is almost abandoned in Diesel-engine practice. It was, however, the method employed on the first American Diesels for lubricating the bearings and connecting-rod brasses as well as the piston. The American Diesel Company used a mechanical oil pump for the purpose of lubricating the cylinder walls but, in fact, depended on the splash from the crankcase to meet most of the cylinder-lubrication demands.

The great objection to the splash system is the difficulty of preventing excessive oil deposits forming on the cylinder walls.

Reduction of Bearing Impact.—Even if other methods provide a satisfactory oil film, there is no doubt that the pressure provides

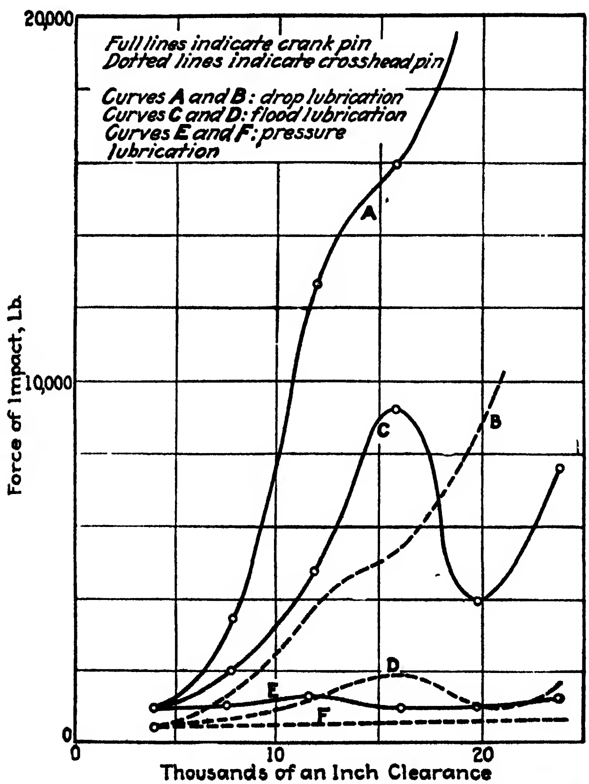


FIG. 280.—Relative bearing impact blows with different oiling systems.

a method of filling the bearing clearance with a fluid that acts as a cushion to absorb the shocks when the pressure on the bearing is reversed; and even a two-cycle engine has a reversal of pressure on the rod bearing at some load or speed point. Just what this cushioning means is indicated by the curves in Fig. 280, which shows the relative impact on a bearing observed during a test. The impact, or hammer blow, with pressure feed is low, as will be seen. In fact speeds of 1,000 and even 500 r.p.m. would be impossible with any other method.

Cylinder-lubrication Oil Specifications.—Although oils of widely varying characteristics give excellent service in many plants, the following specifications, if adhered to, will guarantee the procurement of an oil that will give satisfactory results. Practically all refineries have a lubricant that will meet these requirements.

Lubricating-oil Specifications

	°F.
Boiling point	600 to 700
Flash point	325 to 500
Fire point	400 to 600
Viscosity at 100° F Saybolt	550 to 800
Specific gravity, Baumé	18 to 24
Carbon content, per cent	0.05 to 0.2
Sulphur	None

If the lubricating oil is to be used on the compressor cylinders, the flash point must be above 450°.

One should purchase an oil from a responsible refiner who can show that his oil has given satisfaction in other engines of the same make.

Bearing Lubrication.—The viscosity of the oil used on the bearing may vary over a wide range and still be satisfactory. The oil with the lower viscosity, all other things being equal, will cause the bearings to run at a lower temperature; but even with a high-viscosity oil after a time the bearing temperature becomes constant.

Testing Lubricating Oils.—The method to be followed in testing internal-combustion cylinder oils depends a good deal upon what sort of information is desired. To compare the relative suitability of two oils for a power-plant engine, we must test them under identical conditions. The average operating load and speed should be selected or, if the load is variable, the test should be made long enough to average up the loads on both tests. Although it will not be feasible in some instances to secure all the information suggested, the following log recommended by W. F. Osborne of the Texas Company gives a list of the data usually needed when comparing oils.

A. Time:
1. Time of starting the test.
2. Time of each reading.
3. Time of ending the test.
4. Total number of hours in operation.

B. Oil consumption:
1. Gallons of oil put into lubricating system or lubricator at start.
2. Gallons of oil added to system or lubricator during test.
3. Gallons in system or lubricator at end of test.
4. Calculate gallons consumed per brake horsepower per hour or per kilowatt-hour.

C. Fuel consumption:
1. Fuel in tank or reservoir at start of test.
2. Fuel added during test (use meters for gas).
3. Fuel in tank or reservoir at end of test.
4. Calculate pounds of liquid fuel or cubic feet of gas per brake horsepower hour or per kilowatt-hour output.

D. Load conditions:
1. Revolutions per minute.
2. Brake horsepower by prony brake or dynamometer at ¼, ½, ¾, and full load.
3. Horsepower by indicator diagrams at loads as in D2 and also for no load.
4. Calculate mechanical efficiency at various loads, and plot curve.
5. Calculate frictional losses at various loads, and plot curve.

E. Influencing operating conditions:
1. Temperature of engine room.
2. Temperature of circulating water to engine.
3. Temperature of circulating water from engine.
4. Gallons of water circulated per hour.
5. Temperature of oil in crankcase (if crankcase oiling is used).
6. Temperature of oil to bearings in case of mechanical lubricator.
7. Temperature of air to cylinder.
8. Temperature of exhaust gas.
9. Time of ignition.
10. Analysis of samples of exhaust gas to determine efficiency of combustion.
11. Analysis of samples of fuel to determine constancy of fuel.

F. Condition of engine at end of test:
1. Condition of cylinder head, piston rings, and valves, with respect to wear, polish, and carbon deposits.
2. Character and amount of carbon deposits.
3. Analysis of crankcase oils to show deterioration of lubricant. (Samples should be taken at intervals during the actual running of the test.)
4. Analysis of new oil for all physical and chemical properties.

If a test is made to compare the lubricating performance of one oil in two or more engines, in addition to the data herein indicated complete information concerning the mechanical factors influencing lubrication should also be secured. Some of these items are as follows.

1. Cylinder diameter and length.
2. Piston stroke.
3. Piston diameter and length.
4. Area of piston in contact with cylinder wall.
5. Piston clearance. If piston is tapered, clearance at all points of variation should be ascertained.
6. Number and type of piston rings.
7. Width of piston rings.
8. Piston-ring pressure.
9. Character of piston metal.
10. Diameter and length of all bearings—connecting-rod, wristpin, and main-shaft bearings.
11. Bearing clearance.
12. Type of bearing metal.
13. Method of lubrication (full description including lubricators, oil grooves, pipes, leads, pumps, filters, capacity of system, rate of circulation, etc.).
14. Method of fuel admission (carburetor, air injection, solid injection).
15. Method of fuel ignition (spark, hot plate, compression).
16. Type of cycle (two stroke, four stroke).
17. Method of cooling engine (air, forced-water circulation, siphon, oil, etc.).

Amount of Lubrication.—The lubrication consumption varies over wide limits in various makes of engines. Table XI gives values that follow very closely the amounts necessary on engines from 200- to 600-hp. capacity.

TABLE XI.—DIESEL LUBRICATION REQUIREMENTS

	Drops per Min.
Air-compressor cylinder	3 to 4
Engine cylinder	25 to 30
Wristpin	10 to 15
Crankpin	15 to 25
Helical gears	5 to 10
Governor	4 to 8
Exhaust-valve stem	2 to 5
Admission	2 to 5

The engine as a whole should use about 1 gal. of oil for each 3,000 or 4,000 hp.-hr. of engine rating. This means that a 1,000-hp. engine should run 3 hr. on 1 gal. of oil even though the load is only 500 hp. In Fig. 281 are curves showing the lubricating-oil consumption of 36 plants reporting to the Sub-committee on Oil Engine Power Costs, of the American Society of Mechanical Engineers

Reclaiming Lubricating Oils.—The methods of reclaiming the oil from the Diesel lubricating system may be classified as follows:

Type I Tanks in which the oil is allowed to settle

Type II Reclaimers in which the oil is heated above the boiling point of water in order to remove the latter; the oil after this is filtered through a bed of activated clay

Type III Those systems where the oil is treated by chemicals

Type IV. Centrifugal separators

The only advantage possessed by Type I is that of cheapness, for all that is needed is a covered tank or even a barrel. The

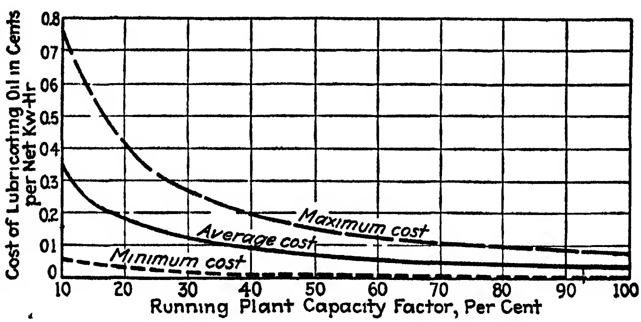


FIG. 281 —Lubricating-oil costs reported to A S M E Diesel Cost Committee.

time required for the water to separate from the oil, however, and for the dirt, grit, and carbon to settle to the bottom is considerable If the oil is disturbed by the pouring of an additional supply of used oil, the separation will be prolonged. Only in places where a more modern arrangement cannot be procured should this system be used

Type II has come into wide use since 1935. The lubricating oil to be cleaned usually contains a certain amount of water. In the typical reclaimer this water is driven off by heating. The hot oil is then allowed to flow into a tank in which is a layer of activated clay. In percolating through this clay, the oil is freed of all dirt and emerges with its color restored. In some clay systems the oil is mixed with a charge of clay and is finally forced through a filter press. Other designs have the clay molded in the form of a hollow cylinder, and the oil flows through the wall.

Systems using activated clay to remove foreign matter from lubricating oil may be operated continuously on a by-pass, or they may be arranged to batch-treat the oil. The system shown by the cross section (Fig. 282) filters the oil through activated clay, evaporates off any water or light fuel oil, and returns the oil to practically its original color.

The motor *Q* drives two plunger pumps *C* and *M* and a vacuum pump *G*. Pump *C* draws dirty oil from a storage tank or from

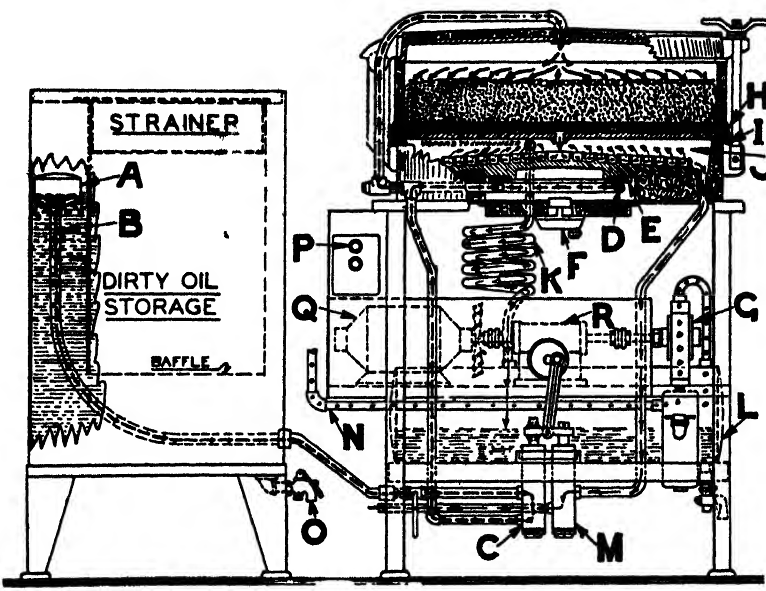


Fig. 282.—Hilliard oil purifier uses fuller's earth.

the engine crankcase; pumps it through a coil *D* under the heating element *E*, where the oil temperature is raised; and delivers it to the special activated-clay filter element at the top of the unit. The oil is forced under pressure through the filter element, which consists of Hiltite, filter paper *H*, and canvas *I*, and falls on to the plate *J*, which is heated by the electric heating element *E*. It flows in the spiral channels formed in the plate, finally reaching the suction of plunger pump *M*, which returns the cleaned oil either to the engine or to a clean-oil storage tank. The pump *G* maintains a suction in the distillate tank *L* and draws water and light-oil vapors from the space above the heated plate through the condenser coil *K*.

Type III covers those systems wherein the oil is placed in a tank and after being heated is dosed with a certain amount of chemicals. The chemicals cause the dirt and carbon to settle at the bottom. The oil is then drawn off to a second tank where further heating decreases its viscosity enough to allow the entrained water to settle out.

In Fig. 283 is shown the De La Vergne oil reclaimer using Oilite.

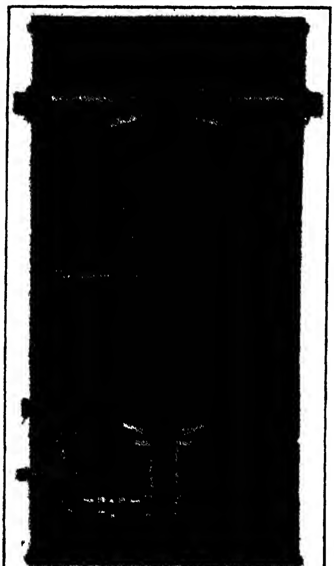


FIG. 283.—De La Vergne chemical oil reclaimer.

On analysis Oilite compound gave the following results:

Sodium carbonate (Ha_2CO_3) 84 per cent.

Sodium hydroxide (HaOH) 3 per cent together with traces of potassium, nickel chlorine, and other minerals. The compound is merely a commercial soda ash and as such may be purchased at any drugstore.

Type IV, the centrifugal filter, is nothing more or less than a cream separator. The dirt and carbon, as well as the water, are thrown out of the oil by centrifugal force. The results are absolutely satisfactory, and the oil is restored to its original grade. If the service is particularly difficult, the oil may subsequently be put through a filter press or terry-cloth filter.

Centrifugal Purifiers.—Centrifuges have been in use for a number of years for the purification of used lubricating oils, and it is hardly necessary to justify their use for this purpose. Adequate oil purification represents an actual money saving to the engine operator, and the best form of oil-purification device is justified to obtain the maximum money saving and the best purification at the same time.

High-speed centrifuges are constructed to purify dirty oils containing traces of water or dirty oils containing larger quantities of water, including emulsions. In the former case the centrifuge is operated as a "clarifier," discharging a single effluent, the purified oil. The dirt originally suspended in the oil is retained in the bowl or rotor of the centrifuge as a compact cake. In the other case, where water is present in relatively large quantities, the centrifuge is operated as a "separator," discharging two effluents, one the purified oil and the other the water, which usually carries with it some of the dirt. The same apparatus may be used as either clarifier or separator by making a simple adjustment requiring but a few minutes of the operator's time.

The dirt retained in the rotor of the centrifuge should be cleaned out at regular intervals, once or twice a day in most plants. In one make of centrifuge the rotor consists of only three parts and weighs, when full of dirt, less than 50 lb. Thus it is easy to handle and easy to reassemble. With this type one may stop the centrifuge, remove the dirty rotor, replace it with a spare rotor, and have the centrifuge in operation again within 5 min.

It should be noted that although the centrifuge separates water from oil. it will not separate kerosene from oil. The water, being immiscible with the lubricating oil, is easily separated. The kerosene readily mixes with and dissolves in the lubricating oil and is not separable. For similar reasons the centrifugal purification of a lubricating oil has no effect on its viscosity or on its flash or fire points.

The centrifuge is not a filter and is not subject to the same limitations as a filter. Filters are often inefficient in removing water and fine sediment from the oil. Where they do succeed, the filter medium rapidly becomes clogged, and the capacity for filtering falls off. The filter medium absorbs oil, which must be thrown away when the filter medium is replaced. None of these things is true of the centrifuge, since it contains no filter medium and the removal of sediment and water is positive. Its capacity does not decrease as purification proceeds, as is the case with filters.

To obtain the best results with centrifuges, as with any other purifying apparatus, it is essential that they be correctly installed. There are two general systems of using centrifuges for oil purification, the batch system and the continuous by-pass system. Where the lubricating system is so arranged that the oil passes through the engine but once, as when supplied by gravity sight feed or by mechanical force feed, the centrifuge is used on the batch system. The used oil is saved. When a sufficient amount has been accumulated, the oil is centrifuged, by means of an arrangement such as Fig. 284. The dirty oil is placed in the elevated barrel and from it flows through a heater, where it is brought to a temperature of about 185°F. It then flows to the centrifuge, where it is separated into purified oil (which is discharged to the oil barrel) and water (which is discarded).

Where the engine is lubricated by a pressure circulating system, The lubricating oil is purified by the continuous by-pass method This system is so arranged that a certain percentage of the oil is continuously passed through the centrifuge for purification. The centrifuge should handle the entire quantity of oil in the system once every 2 to 6 hr , depending on the kind of engine and its

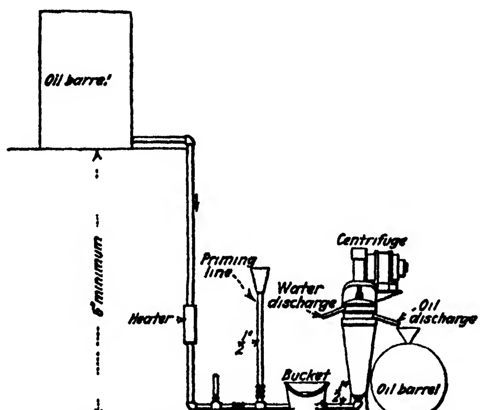


FIG. 284 —Batch system of centrifuging oil.

average load factor. Operating on the continuous by-pass system, the centrifuge is connected to handle the dirtiest oil in the system. This is shown in Fig. 285 in which the feed to the centrifuge is taken from the bottom of the sump tank. The purified oil is returned to the other side of the sump tank. In this way it is always kept clean; water and dirt are removed from it almost as rapidly as introduced.

An important point in the operation of centrifuges is the proper preparation of the oil for purification. This involves merely heating the oil and straining out coarse dirt, such as cotton waste, chips of wood, or metal. Heating is not required for light non-viscous oils. The ease with which a suspended particle may be removed from the oil depends upon the viscosity of the oil. The more fluid the oil the more easily will the suspended particles separate out. An increase of 30 to 40°F. in temperature will usually double the rate of clarification. The oil may be heated by means of the hot exhaust gases, by bayonet-type electric heat ers, or by steam heaters.

Continuous Purification.—As a result of a series of tests on an Ingersoll-Rand 300-hp. engine, A. E. Flowers discovered that continuous centrifuging is superior to the batch system. By this process the dirt is removed as fast as it forms. Furthermore, the oil remains as good as when new.

Although oil may be saved by continuous clarification, it is not thought that this constitutes its major advantage but that the

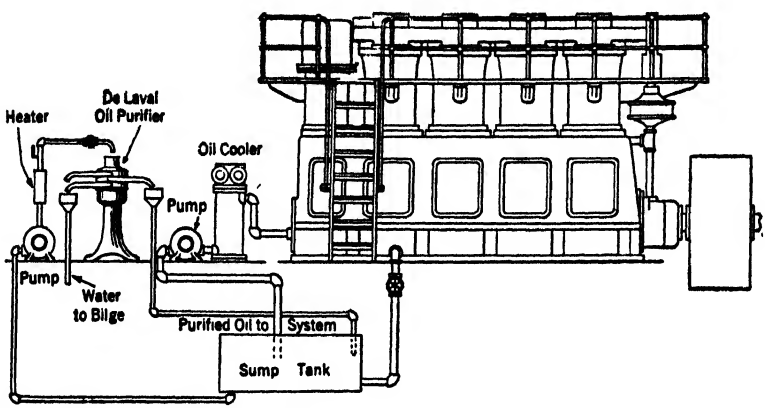


FIG. 285.—A centrifuge arranged to operate on the continuous by-pass system.

reduction of bearing wear is far more important. The bearing wear during the period that the oil was continuously clarified was not enough to measure after 10 weeks, although it had amounted to about 0.001 in. per month before continuous clarification was started. Reduced bearing wear not only increases directly the engine life but also reduces cost of labor for take-up of bearing and outages for repairs.

Batch reclamation may be useful where the engine must be completely overhauled or where some accident has contaminated or damaged the oil or where fuel or other dilution has occurred.

Special tests have shown that heating and allowing a settling period definitely assist in agglomerating the sludge to a point where it is readily removable. Other tests have shown that washing with clean boiling water and then removing the water and sludge together in a centrifugal separator tremendously improves the centrifuging process, allowing purification "through-put" rates about 100 times greater than purification without water washing.

Such simple treatments will restore all the original properties except color and to some extent the neutralization number. The reclaimed oil is clean and free of sludge but usually darker than new oil. It is not believed that color per se is important; and though it might be restored by applying excessively high centrifugal clarification treatment at very low through-put

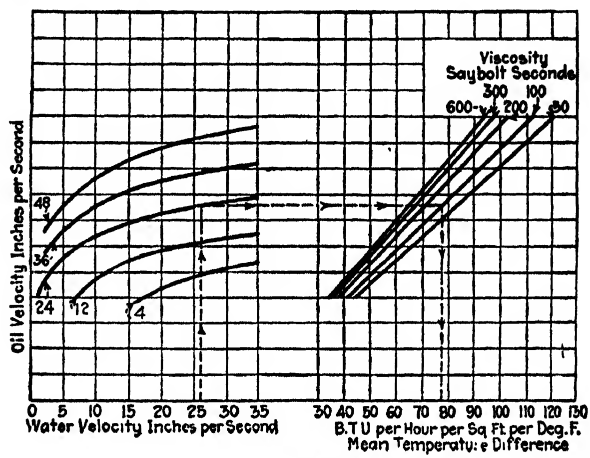


FIG. 286.—Heat transfer in oil coolers.

rates, it hardly seems economically justifiable in view of the fact that in service the oil will be darkened almost immediately.

The neutralization number is improved by the water-washing and centrifugal-separation process, because the freshly produced acidic products of oxidation, which are much more soluble in water than in oil, are removed.

The fuel-oil dilution, if any, may be removed by heating and blowing steam directly through the oil. A high temperature, 400 to 550°F., and the use of superheated steam (steam expanded down from 250 lb. per square inch) greatly reduce the time required and the steam consumption needed for fuel-dilution removal. But fuel dilution is never a serious problem save in worn engines and on engines subjected to frequent starting when cold.

Oil Coolers.—Practically all Diesels have coolers of the shell-and-tube type for the purpose of cooling the oil after use. The size, etc., are determined by the builder. In Fig. 286 is a chart

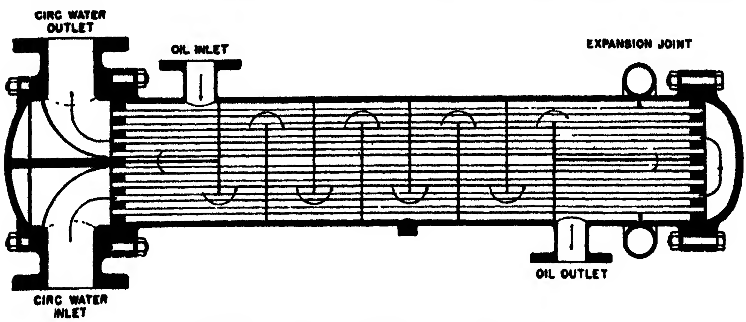


Fig. 287.—Davis shell-and-tube oil cooler.

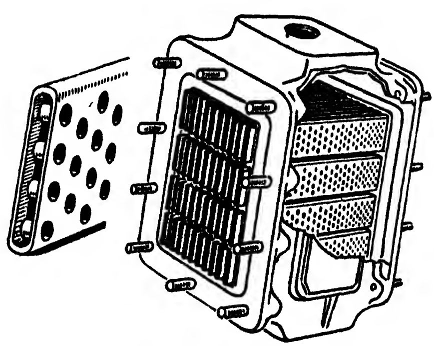


Fig. 288.—Harrison oil cooler.

by S. E. Derby as published in *Power*, showing the coefficient of heat transfer in oil coolers. The viscosity of the oil is of importance, it will be noticed.

In Figs. 287 and 288 are typical oil coolers.

CHAPTER XVIII

COOLING SYSTEMS

TYPES OF SYSTEMS; PUMPS; WATER PURIFICATION

General.—Much of the early trouble experienced with Diesel engines was due to the indifferent methods employed for the jacket cooling. No attempt was made to use a non-scale-forming water supply, and the quantity flowing through the water jackets varied over a wide range. Designers of modern Diesel power plants have given the water supply close study, and, as a result, damage traceable to poor cooling-water facilities is seldom encountered. The system must be properly designed, however, and, equally important, must be taken care of if the plant is to have a successful record.

Distribution of Heat Losses.—Various experiments on the subject of Diesel heat losses check very closely as to final results. The consensus of opinion is that the total heat evolved in the combustion of a charge of oil in the engine cylinder is absorbed in doing work and in various losses in the engine at the following percentages:

Heat	Per Cent
Generated in cylinder	100
Converted into work	30
Lost in engine friction	6
Lost in exhaust gases	28
Absorbed by cooling water	34
Lost by radiation, etc	2

Figure 289 covers the losses at various loads. These values are the result of a number of experiments on various engines. If the work done is 30 per cent of the heat generated, then the engine will consume 8470 B.t.u. per brake horsepower. The cooling water must be of a quantity sufficient to absorb 34 per cent of the amount, or 2879 B.t.u.—in round numbers 3,000 B.t.u. per hour.

Cooling Water Required.—The calculations necessary in determining the amount of cooling water required for a Diesel are quite simple. As an example of the maximum amount that could be used, the intake-water temperature at the jacket entrance can be taken as 90°. This is at least 10° higher than normal, even with a cooling pond. The engine discharge-water temperature can be taken as 115°F., although outlet temperatures as high as 175° are advisable. Then the rise in temperature will

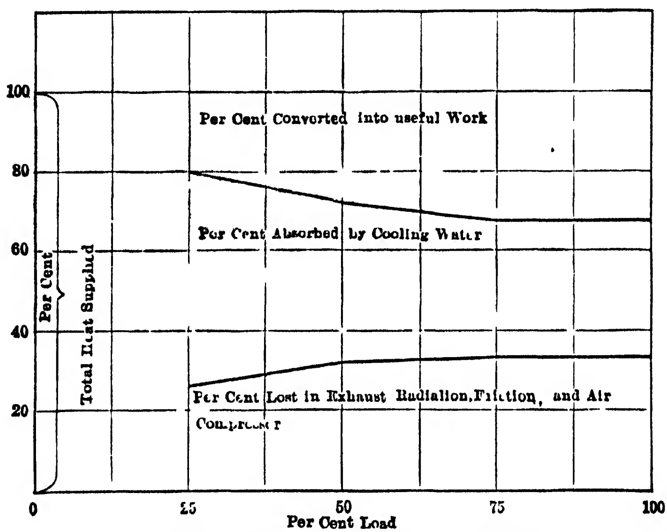


FIG. 289 Heat distribution in Diesel engines.

be 25°; consequently, each pound of water will absorb 25 B.t.u. The water per horsepower-hour will be

$$\frac{3,000}{25} = 120 \text{ lb.}$$

This expressed in the form of an equation appears as

$$W = \frac{XH}{100(t_1 - t_2)}$$

where

W = weight of water required per horsepower-hour.
X = percentage of heat absorbed by water.
H = total heat supplied to engine.
t_1 = discharge-water temperature, degrees Fahrenheit.
t_2 = intake-water temperature, degrees Fahrenheit.

TABLE XII.—TYPICAL JACKET COOLING-WATER DATA FOR DIESEL ENGINES
Three 500 hp. McIntosh & Seymour Diesels, 164 r.p.m.

Unit No. 3	Pounds per minute				Temperature (deg. Fahr.)				B.t.u. absorbed per minute			
	Full	¾	½	¼	Full	¾	½	¼	Full	¾	½	¼
Engine pump intake	......	...	...	...	83	83	83	77	......	......	......	......
Air-compressor discharge	48	33	...	44	110	110	101	104	1,796	891	......	1,188
Intercooler discharge	None in	use										
Aftercooler discharge	31	33	...	36	106	120	136	110	713	1,221	......	1,188
Exhaust-header discharge	171	139	141	176	120	118	102	100	4,327	4,865	2,689	4,048
Cylinder No. 1: Jacket	52	48	42	32	138	132	131	132	2,860	2,352	2,016	1,760
Exhaust valve	11	8	9	10	102	100	96	92	209	136	117	150
Cylinder No. 2: Jacket	56	48	40	36	140	132	130	126	3,192	2,352	1,880	1,764
Exhaust valve	11	8	9	10	106	104	98	94	253	168	135	170
Cylinder No. 3: Jacket	55	48	43	34	140	132	127	126	3,135	2,352	1,892	1,666
Exhaust valve	12	8	10	11	90	104	96	92	84	168	130	165
Cylinder No. 4: Jacket	70	76	49	49	144	122	128	122	4,270	2,964	2,205	2,205
Exhaust valve	11	9	9	10	106	104	98	96	253	189	135	190
Total: Unit No. 3	528	458	352[1]	522	...	...	...	...	21,092	17,653	11,128[1]	14,494
Total: Unit No. 2	698.5	488	482	...	...	...	...	...	23,115	16,921	14,180	
Total: Unit No. 1	647	499	485	...	...	...	...	...	22,975	18,424	13,255	

[1] Except for air compressors.

The inlet and discharge temperature given may not check with those observed in any particular installation; nevertheless the temperature range is approximately correct; this is the factor that is important. Table XII gives the temperatures and quantities of water passing through Diesel engines. This table is the result of a test on three McIntosh & Seymour 500-hp. Diesels installed by the Texas Light & Power Company at Paris, Tex. Since these engines were developing close to 500 hp. each, this value may well be assumed in computing the water rate per brake horsepower. With this assumption the water per brake horsepower per minute was $1^{25}/_{100}$ lb., or 75 lb. per brake horsepower per hour. This is considerably below the value computed, which can be explained on the grounds that the outlet temperature was maintained at a high value.

The cooling system should then be based on a pumping and cooling-tower capacity of at least 120 lb. per brake horsepower per hour of installed engine rating. The gallons of water per horsepower-hour for various temperature rises are shown in Table XIII.

TABLE XIII.—TABLE OF COOLING WATER REQUIRED IN GALLONS PER BRAKE HORSEPOWER-HOUR

Inlet temperature	Discharge temperature, deg.							
	95	100	105	110	115	120	125	120
50	7.2	6.5	5.9	5.4	5.0	4.6	4.3	4.1
55	8.1	7.2	6.5	5.9	5.4	5.0	4.6	4.3
60	9.2	8.1	7.2	6.5	5.9	5.4	5.0	4.6
65	10.8	9.2	8.1	7.2	6.5	5.9	5.4	5.0
70	13.0	10.8	9.2	8.1	7.2	6.5	5.9	5.4
75	16.2	13.0	10.8	9.2	8.1	7.2	6.5	5.9
80	21.6	16.2	13.0	10.8	9.2	8.1	7.2	6.5
85	32.4	21.6	16.2	13.0	10.8	9.2	8.1	7.2
90	....	32.4	21.6	16.2	13.0	10.9	9.2	8.1
			32.4	21.6	13.2	13.0	10.8	9.2

Types of Cooling Systems. Open System for Small Plants.—Two designs of cooling systems are in quite general use. Figure 290 outlines the open system often found in small installations. With this design the water from the engine jacket is discharged through a distributing pipe *D* on a cooling tower *C*. The water drips down through the tower and is stored in the sump A, from which it is drawn by the circulating pump B and forced through the engine jacket and out the discharge again. When this system is adapted to a horizontal engine, the discharge line should rise vertically from the engine until it is above the cooling-tower distributing pipe. With such a layout it is necessary to place a vent pipe in the discharge line immediately above the engine. This prevents the formation of steam or air pockets in the jacket with consequent overheating of the cylinder.

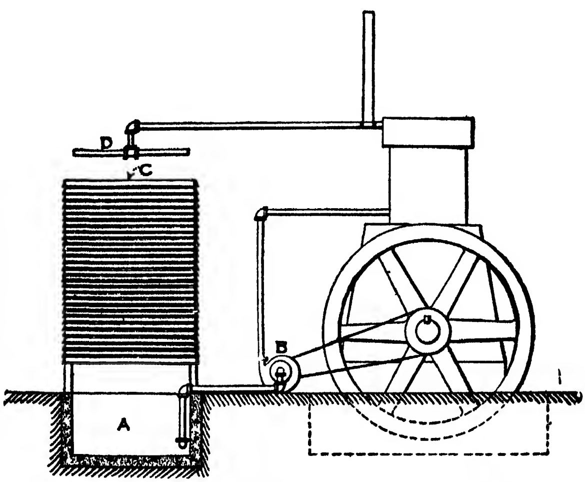


Fig. 290.—Schematical "open" cooling-water system.

Although the figure embodies a belt-driven centrifugal circulating pump, this design of pump is one that should never be employed with a system not having an overhead tank. The objection to this pump is based on the liability of losing the suction. This is of frequent occurrence when the drive belt becomes dirty or oily, and in the best kept plants a belt at times breaks.

There is a serious objection to this system which makes its use inadvisable under any condition with a Diesel engine. The discharge, being at some distance from the engine, is not under the observation of the operator. The circulation can be broken without the knowledge of the operating force; this has resulted in broken cylinder heads and jackets in a number of cases. As a safeguard, a check valve with an outside lever connected to a bell ringer is recommended.

Open Cooling System.—Figure 291 is the schematic layout of the open cooling system with a storage tank. With this system, the water is stored in the overhead tank and enters the cylinder jacket at *A*; after cooling the engine, the water discharges into the open funnel at *B*, flowing into the sump *C*. A centrifugal pump *D* lifts the water from this sump and discharges it in the top of the cooling tower *E*. In dripping down this tower the water is cooled and, collecting in the catch basin *F*, is lifted by the pump *G* and forced into the overhead tower. This system is frequently used without the overhead tank. With this plan the pump *G* forces the water through the engine jacket. It is at once

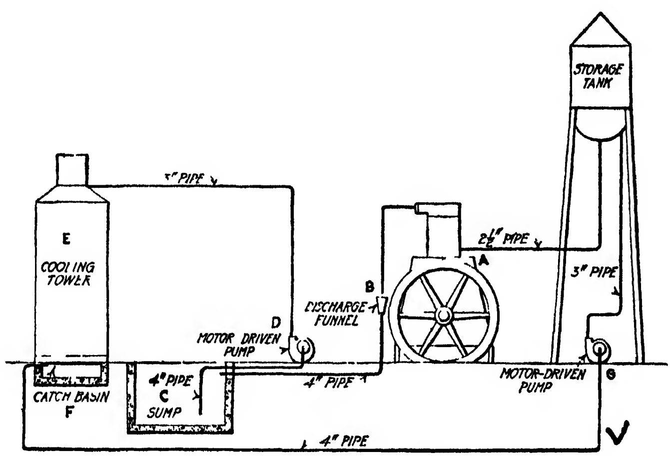


FIG. 291.—Open cooling system with overhead tank.

apparent that this latter plan is objectionable, since it is dependent on the pump *G* for the steady flow of water. If the suction is destroyed, the system at once becomes dangerous. Low-pressure engines are often installed with this system, but the cost of a Diesel plant is entirely too great to ignore the overhead tank. A 20,000-gal. tank with a 30-ft. steel tower can be installed at a cost of approximately $4,000. The interest on the investment ($240) is a low premium on the insurance of protection against engine damage due to lack of cooling water.

With a few engines the cooling water, after passing through the engine, is used to cool the exhaust header. With others the water is first passed around the air compressor and inter- and aftercooler before entering the engine jackets. Results from these cooling methods are fairly satisfactory. The proper cooling-pipe layout embodies individual lines to the air compressor and coolers, to each engine cylinder jacket, to the valve cages, and to the exhaust headers. All these lines should have brass cocks in the intake side, and the discharges should all lead to a common discharge funnel. Each discharge line may well be fitted with thermometers, whereas a single thermometer on the intake line before the lines branch to the various parts is sufficient.

Faults of Open Systems.—Not only must the supply of water be adequate to remove the heat rapidly from the cylinder walls, but it must not contain much impurities either in solution or in suspension. If it does, the water jackets will soon become clogged with scale or mud, the effects of which are just as harmful in an oil-engine water jacket as in the water space of a steam boiler. There is this difference, however: In a steam boiler, scale will form from the sulphates and carbonates of calcium and magnesium held in solution, but in an engine water jacket, where the temperature never exceeds the boiling point (212°F.), only the carbonates are precipitated as scale when the water is heated. In other words, a certain water containing a considerable quantity of sulphates but not carbonates might have a bad record as a boiler water but would be satisfactory for cooling an engine. Such cases are comparatively rare, however, as waters containing any mineral salts in solution are seldom free from carbonates. Water containing more than 6 gr. per gallon of calcium and magnesium carbonates should not be used in engine jackets.

Carbonates are held in solution by an excess of carbon dioxide gas in the water. When this gas is driven off by heat, the carbonates are precipitated in the form of a hard white scale which adheres firmly to the walls of the jacket and forms a non-conducting coating which retards the transmission of heat from the metal walls to the water. The consequence is that the metal walls attain an abnormally high temperature, causing inequalities in expansion and, when the scale is thick enough, finally resulting in excessive stresses which cause the casting to crack.

Originally, water-cooling systems were designed merely for the purpose of saving water where it was scarce or expensive. More recently, however, such systems are being used, even though hard or dirty water is abundant, in order to supply engines with soft, clean water.

The open cooling systems, just described, have unfortunately two serious faults. In the first place there is a considerable loss of water through evaporation and wind, and, in arid climates where water is scarce, sufficient make-up water is often hard to obtain at a reasonable cost. The loss is usually from 4 to 10 per cent of the water passed over the cooling tower. Many oil engines are installed in arid districts in Arizona, Kansas, Oklahoma, etc., where this quantity of make-up water can be obtained only at prohibitive cost.

The second serious fault of open cooling systems is the fact that they concentrate the impurities in the water. If the water is hard, scale will be formed in the engine water jackets, the same as in a simple cooling system where the water is discharged to waste. But in the open cooling system, even with water only slightly hard, concentration of the mineral salts in solution occurs rapidly, so that a water originally safe soon becomes loaded with scale-forming constituents and deposits a coating of scale in the engine water jackets. This concentration of impurities occurs because the evaporation at the cooling tower consists of pure water only, and the materials originally dissolved in the evaporated water remain behind, dissolved in the rest of the water. With new make-up water, mineral constituents are added to the system, which thus becomes more and more impure, resulting in scale deposits in the water jackets.

Inclosed Cooling Systems.—The modern cooling system goes under the name of "inclosed cooling system." This type overcomes the two serious faults of open cooling systems, namely, expense of make-up water and danger of scale. It requires practically no make-up water and, when once filled with soft water which cannot form scale, will recirculate it continuously for a long period.

The principal feature is that the jacket water is recooled in pipe coils over which raw water is showered. The only place where the engine water is exposed is at the visible outlet. The loss is therefore negligible, and a sufficient amount of good make-up water (rain water, for example) can almost always be easily obtained. Any quantity of water may be showered over the outside of the cooling coils.

A typical inclosed cooling system is shown in Fig. 292. It is filled at the start with soft water. The warm water from the engine outlet is discharged into a funnel on the engine-room wall, from which the water runs into a sump pit. From this pit a circulating pump delivers the water into the cooling coils (usually outside the building) and thence up into the overhead tank from which the engine is supplied by gravity.

The raw water for cooling the pipe coils, if available in large quantities, may be distributed over the coils by means of a

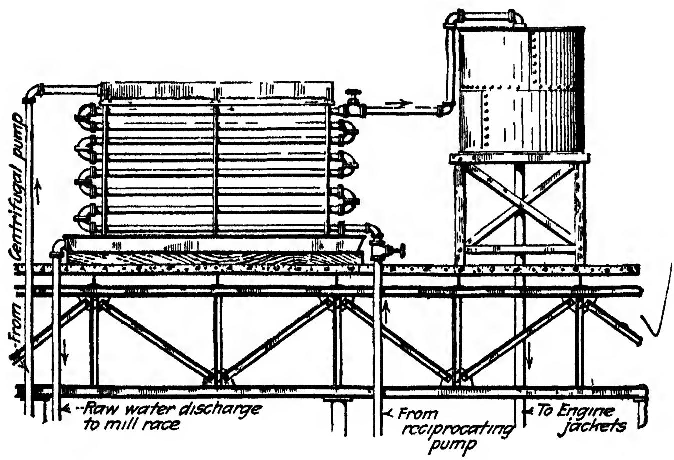


Fig. 292 —Closed cooling system.

slotted gutter pipe in the same manner as in an atmospheric-ammonia condenser of a refrigerating system. With this arrangement a cooling coil made of twelve 2-in. pipes 20 ft. long is adequate for a 100-hp., modern-type oil engine. For an engine of this size the raw-water supply should be 25 gal. per minute. The capacity of the overhead storage tank, the sump pit, and the soft-water circulating pump should be the same as for the open cooling systems previously described.

Where there is a scarcity even of inferior water for showering over the outside of the pipe coils, the inclosed cooling system in Fig. 292 may be rearranged. The raw water itself is caught in a basin below the pipes and pumped by a second pump over an open screen-type cooling tower, where it is recooled and then used again over the pipe coils. The logical location for the wire screens is over the pipe coils themselves, so that the cooled raw water as it drops off the bottom of the screens falls directly on the top pipes of the cooling coils. To secure the best results, the pipe coils are made lower than in the ordinary case previously

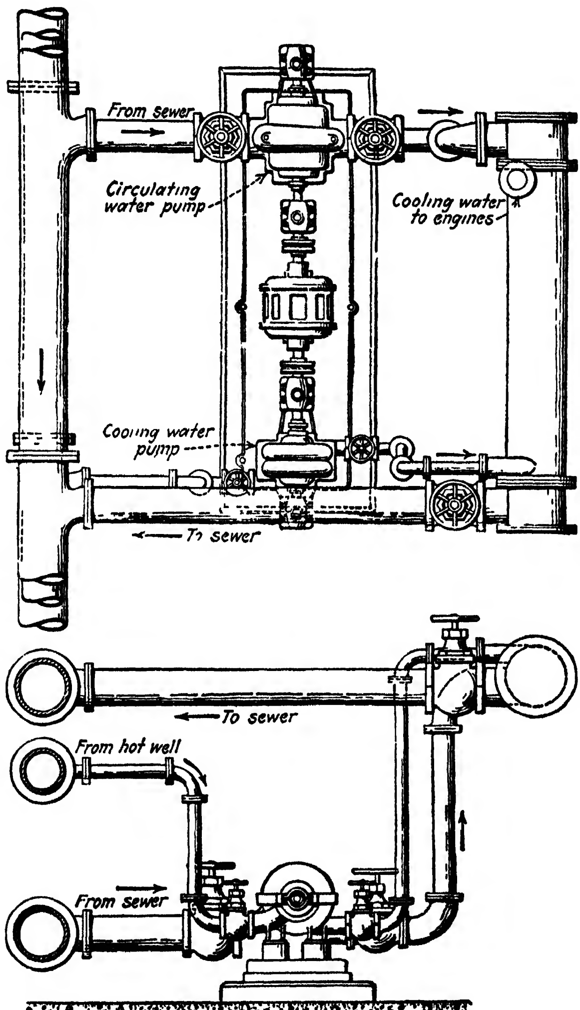


Fig. 293.—Cooling system with heat exchanger.

described, and for a 100-hp. engine two coils are recommended, each made of six 2-in. pipes 20 ft. long. Over each coil is mounted a ¼-in. mesh wire screen 6 ft. high and 20 ft. long.

A second inclosed system is illustrated in Fig. 293. This is the arrangement used on four 750-hp., Busch-Sulzer, two-stroke-cycle engines in the Calumet sewer-pumping plant at Chicago, Ill.

The cooling system consists principally of a double centrifugal-pump unit with a motor mounted between pumps as a common prime mover, a heat exchanger, and a booster centrifugal pump for handling the piston cooling water and necessary piping.

Circulating, or raw, water, which normally, will be river water, is drawn from a storm sewer by the circulating-water pump, a basket strainer being placed in the suction line. The water is

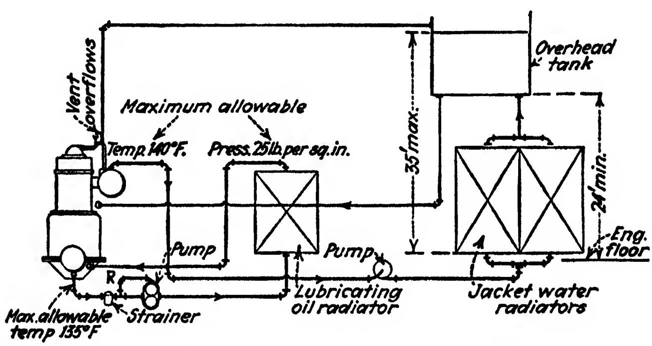


FIG. 294.—Cooling system using air-cooled radiators.

circulated through the tube side of the heat exchanger (Fig. 293) and out to the discharge sewer.

The engine-cooling water is taken from a hotwell in the basement by the cooling-water pump, which is the second element of the double pumping unit, forced through the shell side of the heat exchanger, and then forced through the cooling-water passages of the engine. The cooling water upon leaving the engine flows by gravity back to the hotwell, where it is again circulated through the cooling system. A small booster centrifugal pump takes a part of the cooling water as it leaves the heat exchanger and forces it through the cooling passages of the engine pistons, the water returning to the hotwell by gravity.

Typical cooling systems, as installed by Fairbanks, Morse & Company, are shown in Figs. 294 to 297. These cover the radiator, the cooling tower, and the tubular heat-exchanger applications.

Recirculation System.—Since it is highly desirable to have the cooling water flow through the jackets at constant velocity

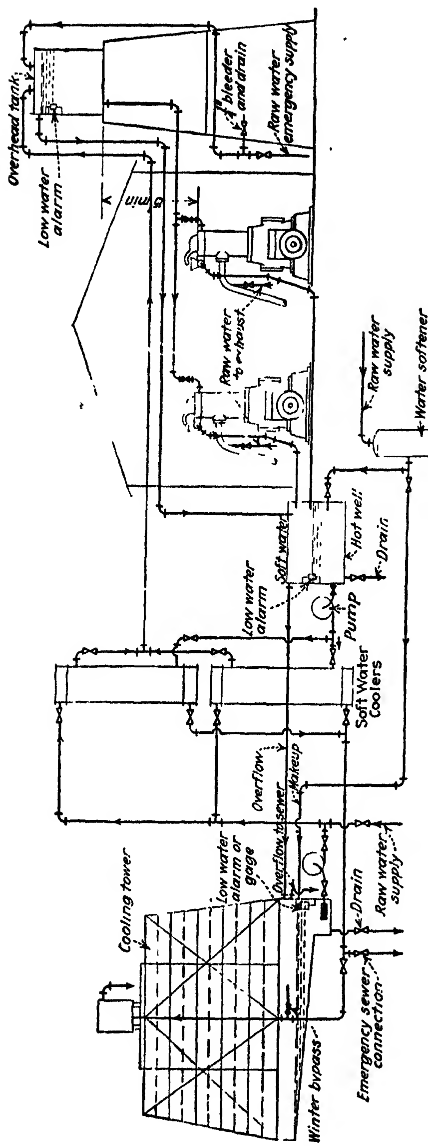


FIG. 295.—Cooling-water system using raw-water cooling tower and shell-type heat exchangers.

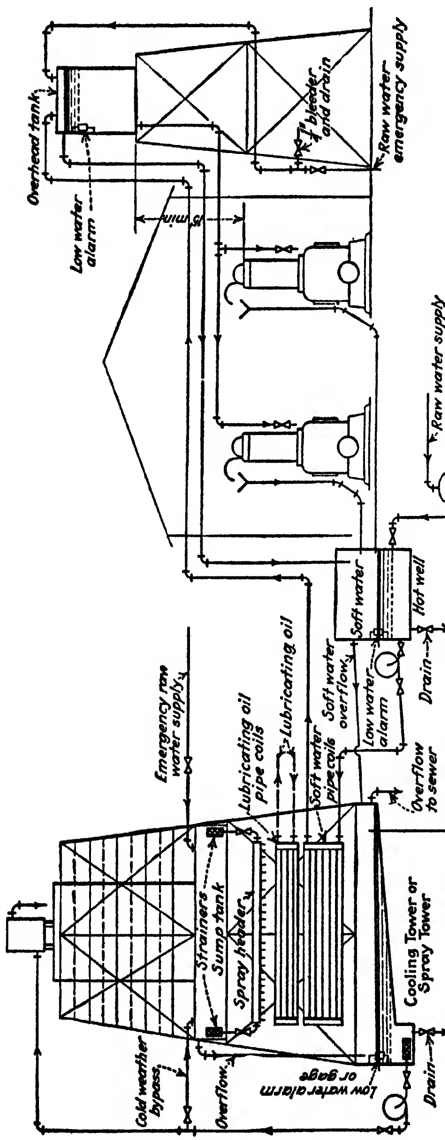


FIG. 296.—Cooling-water system with cooling tower and pipe-coil heat exchanger.

at all engine loads and at the same time have the discharge-water temperature reasonably constant, a present practice is to recirculate part of the cooling water.

A system of this type is shown in Fig. 298. Here the jacket water is discharged into a surge tank and then flows to a sump *S*. A pump picks up the hot sump water and forces it through a heat exchanger, where it is cooled by a flow of raw water. The cooled pure water enters the cylinder jackets as shown.

If the engine load drops, the water-discharge temperature is reduced. This lowering of water temperature acts on the

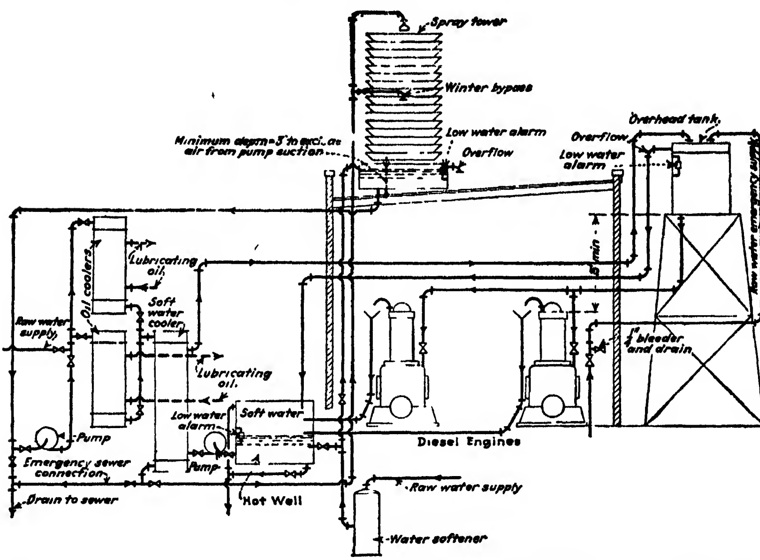


FIG. 297.—Cooling-water system with spray tower, heat exchanger, and softener for make-up water.

thermostatic device *A*, which is connected through a liquid-filled tube with by-pass valve *B*. Valve *B* opens slightly, to allow part of the water delivered by the pump to by-pass around the heat exchanger and so avoid being cooled. The temperature of the water reaching the cylinder jackets, consequently, is higher than would be the case if all the water had passed through the heat exchanger. In Fig. 299 the same result is accomplished by valve *B* allowing part of the jacket discharge water to flow directly to the pump suction.

Water Pipe.—All the water lines in the engine room are best laid in pipe chases. This places the piping out of sight and provides more room in the plant. The pipe should be extra-heavy galvanized; though the pressure is small, this thickness prolongs the life of the system. The threads in all fittings must be clean and sharp, and the pipe ends should be fully threaded. Red lead or other dope must be avoided, and the

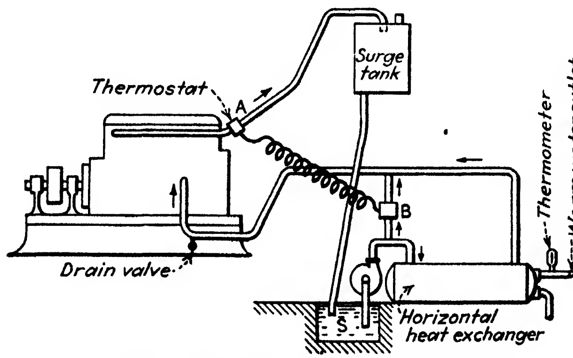


Fig. 298.—Recirculation cooling-water system.

unions should have either ground joints or copper gaskets; rubber gaskets are at best short-lived, and a pipe line should be made up in such a way that it will never give trouble. In those engines where water is admitted direct into the exhaust pipe, the drip line from the exhaust should not discharge into

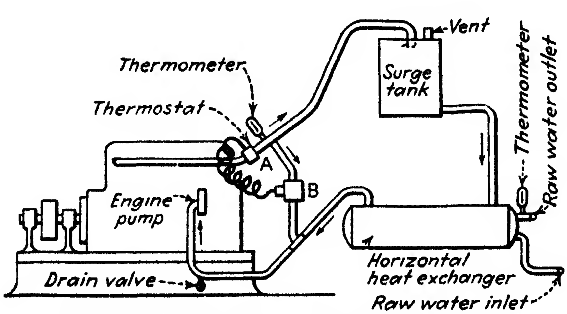


Fig. 299.—A second system with recirculation.

the cooling-water discharge; because of the carbon in suspension, it is advisable to run this drip to the sewer. Frequently, the fuel contains enough sulphur to "eat" iron pipe if employed in the drip line; consequently brass drips can be more profitably used. The piping should never be buried in concrete.

Cooling Towers and Tanks.—As has been already explained, an overhead tank is advisable in every Diesel installation. A steel tower and tank of 10,000 to 20,000-gal. capacity is sufficient for any installation under 3,000 hp , since in this maximum

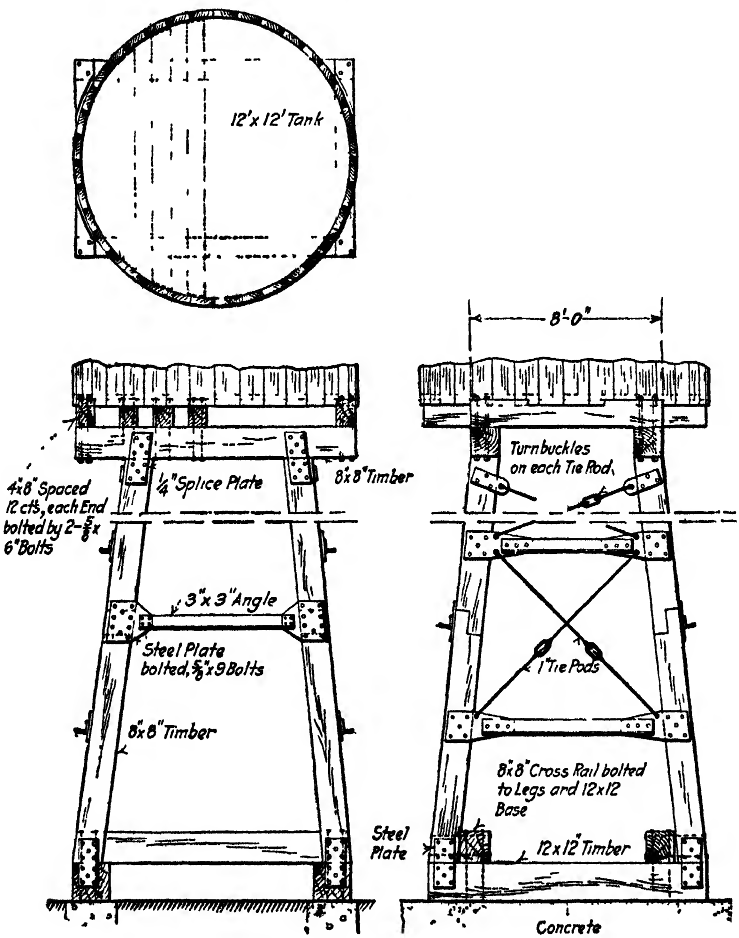


FIG. 300 —Wooden tank and tower.

case a 20,000-gal. tank would provide, in the event of pump failure, cooling water for ½ hr. Plants of 300 hp. or less will find a wooden tank on a wood tower quite satisfactory (Fig. 300). A 12 by 12-ft. tank of 2-in. cypress staves will hold close to 12,000 gal., and can be erected on a 30-ft. tower at a cost of

$1,600. A tower made of 8 by 8-in. yellow pine with the joints reinforced by steel plates and braced by diagonal steel rods is amply strong for this tank.

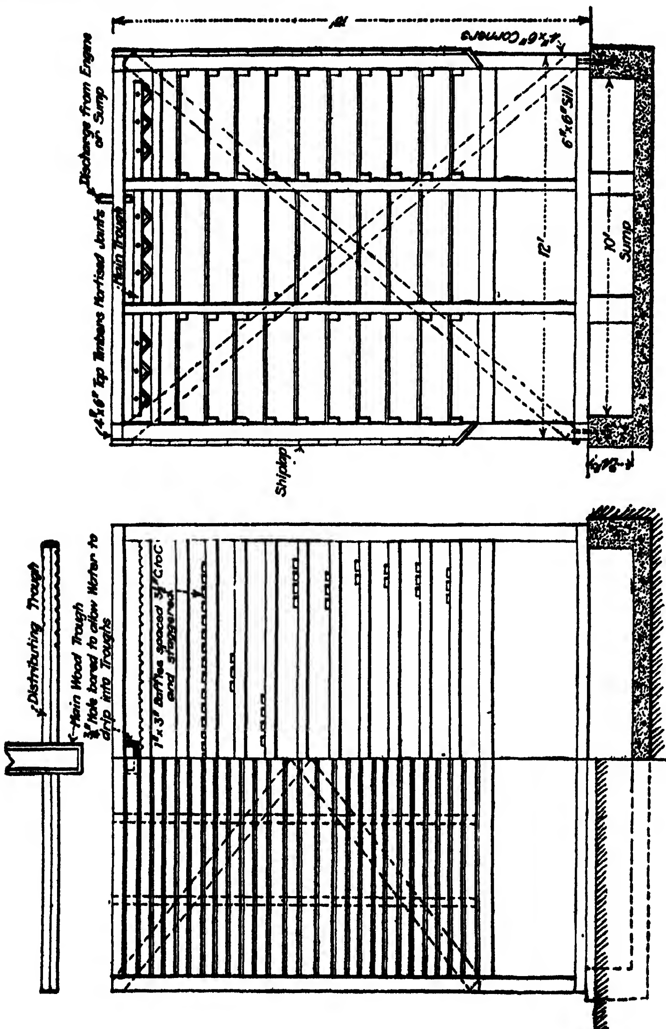


FIG. 301.—Cooling tower for 300-hp. plant.

Cooling Towers.—If a supply of raw cooling water can be secured from a shallow well, the best plan is to install a power well pump and allow the discharge to waste into the sewer. Unfortunately, such a supply is seldom available, and some form of cooling tower whereby the discharge water can be cooled for reuse becomes a necessity. This, in most installations, consists of an upright wooden tower filled with slats down which the water trickles and is cooled by the upward current of air. Like all structures, the tower may be constructed at an expense ranging from a few hundred to several thousand dollars. The former cost covers a simple tower for a small plant, but a large installation demands the more expensive construction.

Plants of 300 hp. will find the tower in Fig. 301 fairly economical in first cost and amply large for the required cooling. This can be erected at a total material and labor charge of $900. The sump under a tower is made 24 in. deep but can easily be deepened at slight expense. The 3/4 by 16-in. foundation bolts should be inserted at the time the concrete is poured. The frame of 4 by 6-in. timbers resting on 6 by 6-in. sills is large enough, especially since the shiplap sides further strengthen the structure. Shiplap is used with the idea of having the air currents enter the tower under the bottom row of baffles and pass out at the top. If the sides are open, practically no circulation is set up.

The discharge from the engine first flows into the main trough and, passing through a series of holes in the trough sides, enters the distributing troughs. The latter are notched to allow the water to overflow before completely filling the trough.

For larger plants it is economical to turn to a cooling-tower manufacturer for tower equipment. Even for small plants the manufacturer is able to sell a tower at a price lower than the cost of a homemade tower.

So far only the atmospheric tower has been mentioned, but of late the mechanical-draft tower has come into general use.

Mechanical-draft Tower.—The mechanical-draft cooling tower may be regarded as an atmospheric tower provided with some form of fan to increase the air flow up through the tower. The sides are usually sealed, and the air exits at the top. The height is usually held to 12 or 15 ft.

Spray nozzles may be used, and the interior cleaned of the slat filling. In this case the stack area must be about 25 per cent greater than with the filled tower.

The fans are usually placed in the lower side toward the bottom, as shown in Fig. 302.

Mechanical-draft towers have the following advantages:

1. Require small flow space.

2. Give uniform cooling.
3. Low water loss.
4. Less objectionable to neighbors because of less windage.

On the other hand, this type of tower costs more than does an atmospheric tower, and there is a constant consumption of power for fan drive.

Other mechanical-draft towers place the fan at the top. This fan, of propeller type, induces air flow up through the tower.

Evaporative Towers.—Of late, much has been made of "evaporative cooling towers." In fact, all cooling towers depend for their cooling action upon the evaporation of part of the water. Certainly, there is some cooling by direct conduction of heat from water to the air, but evaporation of part of the water spray occurs.

The new designs are intended to remove the heat from the jacket water flowing inside a heat exchanger, by the evaporation of part of the raw water spraying over the outside surface of the heat exchanger. In other words, the unit is a radiator over which water is sprayed. If the action were complete, the raw water dripping down from the heat exchanger would be as cool as when it issued from the spray nozzles. Unfortunately, this is not obtained, and it is necessary to cause air to flow up through the water streams, to encourage evaporation. The evaporative cooler as now built consists of a sheet-metal shell through which air travels vertically by induced- or forced-draft fans. In the path of the ascending air are located a series of tubes carrying the jacket water. These tubes have finned surfaces, and a spray-nozzle system discharges the raw water over the fins where evaporation occurs. Excess water collects in a pan, from which it is pumped back to the nozzles. Jacket water is cooled through a 20° range with off-engine water temperature between 140 and 160°F. The raw-water circulation is very low compared with other types of heat-exchanger systems, being only about one-third.

The evaporative system seems well adapted to small- and medium-sized engine installations, forming a compact unit for indoor closed systems from which the moist air is discharged to the exterior through ducts. A serious disadvantage is the accumulation of insulating scale on the fins, resulting from scale deposits left by the evaporation of the raw water. Cleaning in

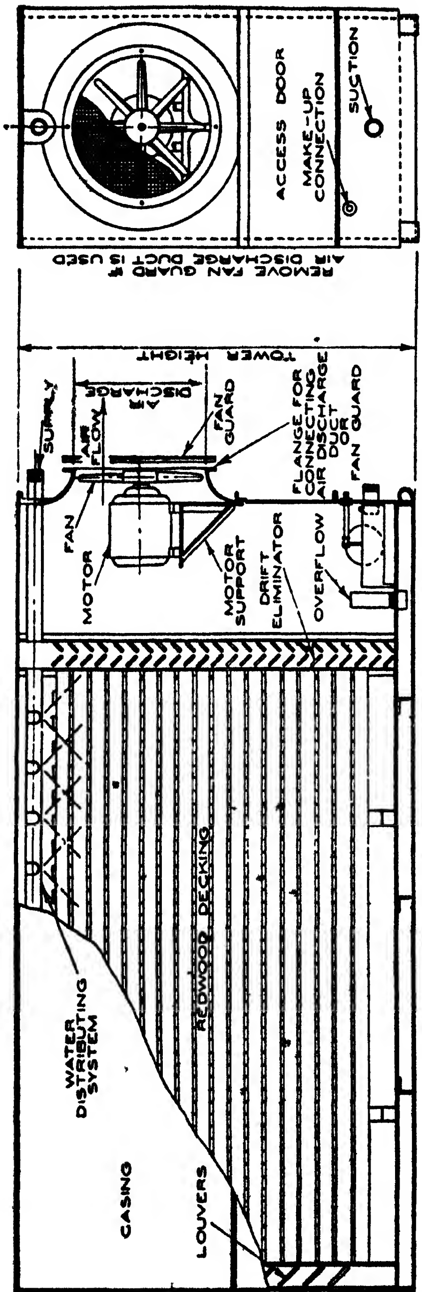


FIG 302.—Cooling tower with forced draft.

place is out of the question, and ordinary descaling methods cannot be used on fins. Tubes without fins may be used where only hard untreated water is available for make-up, but this reduces the surface to such an extent that about 50 per cent more tubes are required.

Circulating-water Pumps.—The Diesel plant requires so few auxiliaries that the engineering of this equipment is often slighted. Consequently, in many cases the pumps of the cooling-water system are entirely unsuited to the particular layout. Many engine stoppages can be traced to the faulty application of pumps that were designed for entirely different services.

The factors that should control the choice of the water pumps are reliability, suitability for the particular installation, low cost, efficiency, and accessibility.

Operation of the engine must necessarily cease upon a failure of the water supply, which directly depends upon the continual functioning of the pump. The design must be such that failure of any part is remote. This is especially vital in installations where the cooling tower is at the ground level and a pump forces the water through the jackets, while a second pump lifts it from a discharge sump and forces it over the tower. In some plants the water flows from the cooling tower to the engine, and the pump merely lifts the water from the hotwell, or sump, and delivers it over the cooling towers. If the capacity of the power-sump storage is ample to supply water for an hour or two, a pump less reliable than otherwise demanded might be used.

Variable capacity is a most desirable feature in the cooling-water pump. The amount of water needed by the engine is in fairly constant ratio to the load. The cooling system must be designed for full-load cooling and should, in fact, be capable of meeting a 25 per cent engine overload. On light loads the water flow can well be reduced; in such installations as have the pump forcing the water through the jacket, a constant flow is seldom satisfactory. If the water-discharge temperature of, say, 140° is the best for the engine, on light loads there will not be so much heat for the water to absorb, and the discharge temperatures will drop down to within a few degrees of the inlet, or cooling-tower, temperature. Not only will the engine's fuel consumption increase, but also the pump power becomes a larger percentage of the engine's output. It is possible to maintain the jacket temperature by a valve-controlled by-pass between the pump suction and discharge; this, however, does not reduce the pumping work.

In installations when a gravity head is on the jacket water, with the pump lifting the water from a sump to a cooling tower, a constant-capacity pump can be regulated by stopping it if the sump level drops by reason of a reduction in the flow through the jackets. This does reduce the pumping work but calls for constant attention from the engineer, unless the sump is of such proportions that it will hold 2 or 3 hr. discharge without the pump being operated. Of course, the cooling-tower sump must be of a corresponding capacity.

The type of drive available also influences the choice of pumps. If the engine is driving a factory through belts, it may happen that no electricity is available for the pump motor. It then becomes necessary to drive the pump from the engine by belt, chain, gear, or crank and rod. Centrifugal pumps are not recommended for these drives, for the drive pulley on the shaft must be extremely large on the driven pulley, or the pump so small as to prevent the belt obtaining a reasonable contact. Belt slapping and chain climbing and binding result when a small pulley or sprocket is used.

Centrifugal Pumps.—By far the most popular cooling-water pump is the centrifugal. This is due primarily to its many desirable features and to the almost entire absence of objectional features.

The centrifugal pump is low in first cost, reliable in service, and has a low maintenance expense. By the proper choice of the impeller diameter a pump can be direct connected to a motor of any speed. As the 1,800-r.p.m. motor is not only cheaper but also more efficient than a low-speed one, the adaptability of the pump to meet any speed requirement reduces the initial investment of the pumping equipment.

One objection to be raised against the centrifugal is that it is not self-priming. If the pumps must actually lift the water, there must be a foot valve and some means for priming the suction line and pump. If the sump can be placed above the floor line, a horizontal pump can be made self-priming, or a vertical submerged pump may be used. The latter, although more expensive, is far superior to the horizontal type for all installations where the sump is placed below the pump.

As a location for the cooling-water pump the basement is to be avoided. In fact, no equipment should be placed there, save the lubrication-oil tanks. All pumps should be under the operator's eye.

The minimum pipe length should be used, but not to the point where convenience of location is sacrificed.

Pump capacity depends upon the size of the engines to be cooled and upon the climatic conditions. If located in a northern state, a summer temperature of 70° is seldom exceeded. Consequently, the quantity of water required is less than in case of, for example, a Louisiana plant where the temperature may be around 90° even with a cooling tower. Other localities where the air temperature may be as high may show a water temperature of 10° less, due to the lower humidity.

At one time engine builders gave but little thought to the subject of pumps, but as it came to be realized that the operation of the plant was not so dependent upon the engine's reliability as upon the working of the water pumps and other accessories, pumps suitable for local conditions were given attention.

Handling of Cooling Water.—After an engine is stopped, the working parts contain a large amount of heat. To remove this heat and bring the piston, cylinder, and cylinder head to a low temperature without danger of any part cooling faster than another and so causing distortion, the cooling water should be allowed to run for 15 to 30 min. The flow need not be heavy.

Upon starting the unit the cooling water should be started through the engine before it is turned over. In engines that are worn considerably, however, the cooling water may lower the cylinder temperature below the ignition point. Under such circumstances it is permissible to start the engine before the water valve is opened. Since the jacket is filled with warm water, there will be no danger of fracture.

In extreme cases of low compression, operators have gone so far as to drain the jackets before starting in order to eliminate all cooling loss. In one motorship this was the common practice, but the danger that accompanies the action is obvious. It is far better to have a small vertical boiler and heat the jacket water before starting the engine; even live steam may be used if the jackets are first heated with the steam and then filled with hot water.

The operator should endeavor to keep a constant jacket-discharge temperature. Much damage has occurred by reason of too wide a variation in the jacket temperature.

Chemical Treatment of Cooling Water.—One common method of removing solids is to add chemicals to the water before it enters the cooling system, the chemical reaction releasing the incrusting solids, which are removed by settling or filtration.

The chemical alone, however, does not solve the operator's problem. He still must use judgment and not permit too great a concentration of solids in the water, with their consequent slow accumulation of scale. It is next to impossible to reduce the hardening properties to zero under normal operating conditions; and when scale in the water jacket reaches $\frac{1}{32}$ in., it should be removed with acid or chemical, as directed by the engine manufacturer or chemical company.

Another method, most commonly used is the zeolite, or exchange-silicate softener. This agent is used in both its natural and its synthetic state and for Diesels constitutes a very satisfactory softener. The total hardness of most natural waters is almost completely removed by merely passing it through a column of prepared zeolite. This requires intelligent operation on the part of the operator, who must not permit the concentration of soluble salts to exceed about 3,000 gr. per gallon, which can be checked with a hydrometer, and he must not let the softener continue to run after the zeolite has given up all its sodium salts. This is readily determined by a simple chemical testing apparatus furnished with the softener. The zeolites used are silicates of aluminum, iron, or other metal combined with the bases, sodium or potassium. These silicates possess the property of exchanging their associated bases for the calcium and magnesium of the water. All make-up water, unless distilled, should be passed through a zeolite system, when the natural water is scale forming.

Water coming through the zeolite bed exchanges the sodium base for the calcium and magnesium in the water, reducing the water to zero hardness, the calcium and magnesium being retained by the zeolite. The chemical reaction is as follows:

Sodium zeolite + calcium bicarbonate = calcium zeolite + sodium bicarbonate

This continues until all of the exposed sodium salts are used up If more raw water is passed through, it will not be affected and will enter the system in its virginal hard state, permitting scale

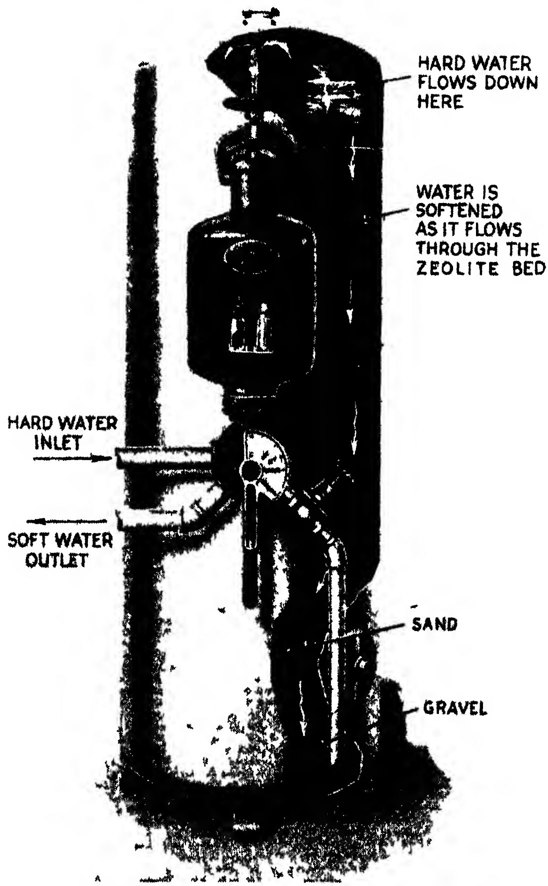


FIG 303 —A zeolite softener for Diesel plants

formation The exchange process is the reverse, the chemical reaction being calcium zeolite + sodium chloride = sodium zeolite + calcium chloride

The reactions for the magnesium are the same as for the calcium. The amount of water passing through at each regeneration depends upon the type of zeolite used and the hardness of the water. The zeolite water softener is quite simple in construction and is easily installed. In Fig. 303 is a small machine built by the Permutit Company.

Removal of Algae.—The most common method of treatment for the control and destruction of algae, which form on a tower or in a reservoir, depends on the use of copper sulphate. The most successful way to apply this is by brushing it on after the reservoir has been drained, washing the dead and disintegrated growth out with a hose afterward. For this method of treatment the reservoir can be divided into halves by a central baffling wall, and the water from one side can be used while the other side is being treated. It is also possible to treat the water while in the reservoir with this chemical by breaking it into coarse lumps and dragging it back and forth in perforated metal cans over the sides and bottom of the reservoir until it is all gone into solution. When this method of treatment is used, however, all the dead growth remains in the reservoir and must be skimmed off or removed in some way to prevent it from becoming foul.

Temperature of Cooling Lines.—Each engine possesses individual characteristics which preclude any set rules as to the temperatures that should exist in the discharge cooling lines from the various parts. Table XIV is the schedule that is followed in a plant containing three 1,000-hp. vertical Diesels. These values give the best possible operating results as applied to these particular units.

TABLE XIV.—COOLING-WATER DISCHARGE TEMPERATURES

Discharge	°F.
Air compressor	105
Inter- and aftercooler	105
Exhaust header	160
Cylinder jacket	150
Exhaust valve . .	150

Recovering Heat in Cooling Water and Exhaust.—There is, as indicated in Fig. 289, a large amount of the heat of the fuel lost to the cooling-water jacket and to the exhaust. The sum of these two losses may be taken, strictly speaking, as 60 per cent of the heat in the fuel. By suitable means a large percentage of this loss may be recovered.

Two methods have been used with success. One of these, by far the simpler, is to connect the engine's cooling jacket to the heating system of the factory. By using ample radiating surface the entire jacket-water heat may be used in heating the plant For example, assume that the engine capacity is 500 hp. With a fuel consumption of 0.4 lb per brake horsepower-hour, the hourly rate of fuel consumption would be 500 × 0.4 = 200 lb. If the oil contains 18,500 B.t.u. per pound and the jacket absorbs 30 per cent or this, the jacket water will deliver hourly to the heating system 200 × 18,500 × 0 30 = 1,110,000 B.t.u. or the equivalent of approximately 1,100 lb of steam. An evaporation of 5 lb. of water per pound of coal is representative of heating-plant practice. The jacket-water recovery is then equal to 220 lb. of coal per hour, or 5,280 lb. per day.

FIG. 304 —Sims exhaust heater.

Recovery of Exhaust Heat.—A large part of the exhaust-heat loss may be recovered by installing some form of exhaust-gas heater. For low-pressure work, say, 10 lb. gage, a cast-iron heat interchanger similar to Fig. 304 has been quite generally used in this country. The amount of steam evaporated, or water heated, is shown in Table XV, these data being issued by the Sims Heater Company.

For high-pressure work a steel-shell boiler is necessary, and the length of the boiler must be such that the gas temperature does not drop below the vapor-condensation point. If this is not taken care of, corrosion will occur in the tube ends.

The waste-heat boiler has been largely developed in England; and if part of the jacket water is used for the boiler feed, it is possible to obtain a boiler output of 1 to 1½ lb. of steam per Diesel horsepower output. Of the heat in the exhaust, not over 50 per cent can be recovered. In the first place, the exhaust is at, say, 700°F., but the heater cannot reduce this to below 250° on account of danger of moisture settling. In addition, for each

TABLE XV.—CAPACITY OF SIMS HEATERS

Engine hp	Dimensions of heater	No of section	Water connections in	Exhaust connections in	Heating capacity gal per hr using exhaust only giving water temperature of			Heating capacity, gal per hr using exhaust and jacket water giving water temperature of			Weight of boiler
					150°	180°	200°	150°	180°	200°	
10	18 by 15 in	2	1	3	45	35	30	110	80	70	360
15	18 by 20 in	3	1	3	70	50	45	160	125	120	475
20	18 by 24 in	4	1½	5	90	70	60	215	160	170	600
25	18 by 30 in	5	1½	5	110	85	75	270	210	180	700
30	18 by 35 in	6	1½	5	130	105	90	330	250	220	820
35	18 by 40 in	7	1½	5	150	120	105	380	290	250	935
40	18 by 45 in	8	1½	5	170	140	125	435	335	290	1,050
45	18 by 50 in	9	1½	5	190	160	140	490	375	330	1,160
50	18 by 55 in	10	1½	5	210	175	155	545	420	370	1,280
60	24 by 39 in	6	2	8	250	200	175	635	485	430	1,640
70	24 by 44 in	7	2	8	290	235	200	720	550	490	1,860
75	24 by 49 in	8	2	8	340	260	230	810	620	550	2,080
100	30 by 39 in	5	2	8	450	350	300	1 090	840	720	2,220
125	30 by 45 in	6	2	8	580	430	380	1,360	1,040	900	2,570
150	30 by 51 in	7	2	8	680	520	460	1,630	1,250	1,100	2,900
175	30 by 57 in	8	2½	12	780	610	535	1,840	1,420	1,270	3,260
200	30 by 64 in	9	2½	12	900	700	600	2,050	1,670	1,450	3,600
225	30 by 70 in	10	2½	12	1,000	780	690	2,390	1,880	1,640	3,950
250	30 by 77 in.	11	2½	12	1,150	870	760	2,720	2,100	1,820	4,300
275	30 by 83 in	12	2½	12	1,250	960	840	2,900	2,300	2,000	4,650
300	30 by 90 in	13	2½	12	1,350	1,050	920	3,140	2,500	2,180	5,000

Using both exhaust and jacket water, this heater will produce 5 lb of steam per horsepower per hour at 10 lb. pressure pound of oil burned, a pound of steam is formed, so at least 1200 B.t.u. escapes in this steam per pound of oil.

Water-tube Exhaust-heat Boiler.—The Foster Wheeler Corporation makes a bent-tube boiler shown in Figs. 305 and 306.

The steam drum is at the top, with fittings as usual, but water connections are unusual. Feed water first enters the upper rows of heating tubes, known as the "economizer section." The

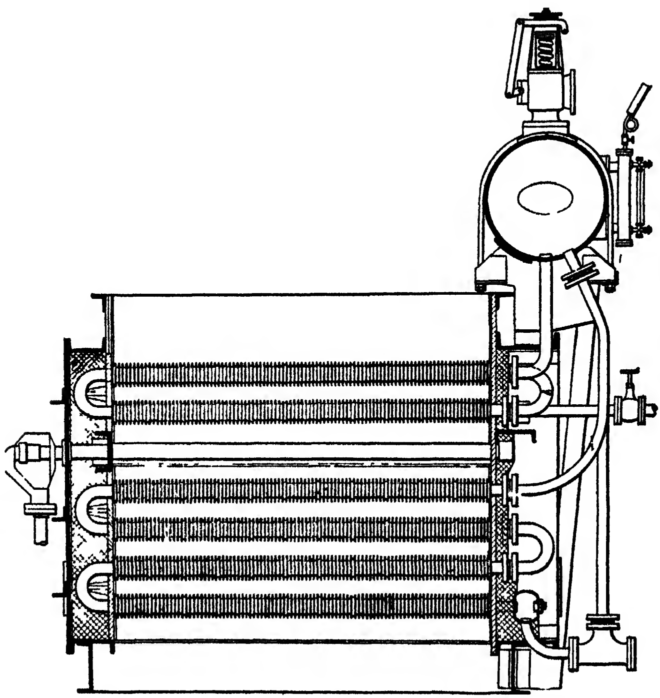


FIG. 305.—Foster Wheeler water-tube exhaust heater.

relatively cool water reduces exhaust gases to minimum temperature. Water from the economizer enters the steam drum and flows down to a header connected to the lowest row of heating tubes. In the boiler shown there are only two downcomer pipes.

The water circulation is then back and forth through the heating tubes, or elements, and upward to the steam drum, there being one drum connection for each vertical row of heating elements. This arrangement is necessary, since no back waterleg or equivalent is used. Instead, there are orifice plates in each downcomer pipe. These are proportioned to restrict the flow of water and prevent reversal of direction. The design has been entirely satisfactory; no steam or water hammer has developed; and ample circulation has produced high steam production.

The casing is made airtight. Element tubes are passed through a tube sheet at the steam drum, or front end, and are rolled directly into the sheet. Then the connecting flanges are

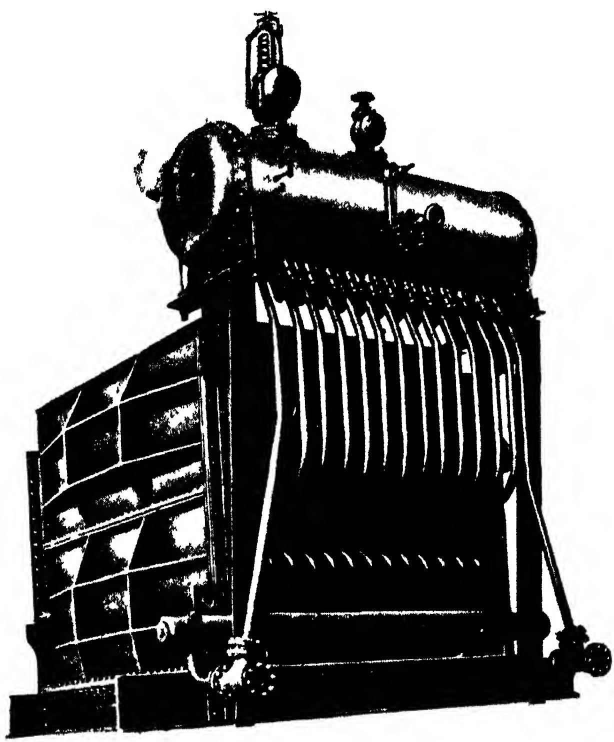


Fig 306 —Casing of boiler shown in Fig 305.

placed on the ends of the tubes, but the tubes are fast at one place only. The return-bend ends of the tubes are supported in plates and are free to move back and forth in the plates. This eliminates any stress at the flanged connections. The return bends are packed in insulation and housed in a tight, steel box.

The boiler, as shown, may be mounted over a small combustion chamber fitted with an independent oil burner for heating the boiler when the engine is shut down.

The exhausts from several moderate-sized engines may be led to one waste-heat boiler if desired.

The assembly is mounted on a structural-steel frame, and the casing is formed of heavily reinforced castings, as shown in Fig. 306.

Since the boiler is not damaged by running dry, the operation is far simpler than with other steam boilers. Instead of keeping the water level constant and regulating the steam supply by the heat in the furnace, the heat for this boiler is constant, and the steam output is controlled by the amount of water admitted. From no steam up to maximum capacity this boiler is governed by merely setting the feed valve for the amount of steam desired. Watching the water glass or steam gage is unnecessary.

The boilers are usually furnished for pressures of 50 to 75 lb. but can be built for higher pressure if desired.

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