HIGH-SPEED DIESEL ENGINES

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HIGH SPEED COMBUSTION ENGINES

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THE AUTOMOTIVE CHASSIS

HIGH-SPEED DIESEL ENGINES

For Automotive, Marine, Railroad
and Industrial Use

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WITH A CHAPTER ON GAS TURBINES

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P. M. HELDT

Member, Society of Automotive Engineers
Formerly Engineering Editor, Automotive Industries

A CHILTON BOOK

CHILTON COMPANY

Publishers · · · · · *Philadelphia*

SEVENTH EDITION

PRINTED IN THE UNITED STATES OF AMERICA

PREFACE TO THE SEVENTH EDITION

For this edition the text again has been completely revised, and new illustrations have been added. While some changes have been made in every chapter, the greatest amount of new material will be found in Chapters II, VII, XII, XIII, XIV, XVI, and XVIII. Chapter II now appears under the head "Thermodynamic and Other Characteristics," instead of "Thermodynamics of the Diesel Cycle," and therefore has a somewhat wider scope. Some new governing devices are described in the chapter on "Modern Injection Pumps and Governors," while to the chapter on "Two-Stroke Engines" have been added—among other things—charts giving expressions for the shaking forces and rocking couples in two-stroke engines with different numbers of cylinders and different crank arrangements.

Two chapters of the previous edition, those on "Aircraft Engines" and "Miscellaneous Types of Oil Engine," have been omitted in this one, the reason being that these engine types are no longer of practical importance. Diesel engines for aircraft are no longer being produced, because it has been found that their advantages over the conventional engine type do not compensate for their disadvantages. Spark-ignition oil engines also have receded into the background. They gained a certain degree of popularity during the infancy of the high-speed Diesel, when the high combustion pressures of the latter gave rise to some trouble. These difficulties have now been largely overcome, and as the spark-ignition oil engine uses a lower compression and is less economical, it has little to recommend it.

THE AUTHOR.

PREFACE TO THE FOURTH EDITION

During the seven years which have elapsed since the last complete revision of this book, many changes have occurred in Diesel practice, and another thorough revision had become necessary. At the time of the previous revision, the American

Diesel industry was still in its infancy, but since then it has expanded greatly, and its products have been refined in many ways. A large amount of experience in different fields of application has accumulated, and will serve as a basis for further development. The basic principles of the engine and of its injection equipment remain the same, and comparatively few revisions were needed in portions of the book devoted to them. On the other hand, nearly all of the actual engines and items of equipment described in the previous edition had been superseded by later designs, and where the changes were of sufficient importance to warrant it, descriptions and illustrations of the newer models were substituted for the older ones.

This book is not intended as a catalogue of the products of the Diesel industry, and no attempt has been made to mention or even describe all of the various makes of engine being built. But it has been my object to cover all of the more important principles made use of in the design of Diesel engines and their equipment, and to illustrate the application of these principles by brief descriptions of current products incorporating them. In the new edition relatively more space is given to American, and less to foreign, makes of engine. . . .

As the industry grows older and the scale of its operations increases, problems connected with the care and maintenance of engines assume greater importance, and this has been taken account of by adding a chapter on Lubrication. Another new chapter is that dealing with Supercharging, a practice that is receiving increased attention in connection with engines for railroad, marine, and stationary applications. The chapter on Aircraft Engines has been largely rewritten and considerably shortened, because of the virtual stagnation in this field.

I wish to express my thanks to Professor P. H. Schweitzer of The Pennsylvania State College for kindly reading the proofs of this edition and for making a number of valuable suggestions. I also want to express my indebtedness to the publishers of various periodicals for permission to reproduce drawings which first appeared in their respective publications. Quite a number of illustrations are from *Automotive and Aviation Industries* of Philadelphia; several are from papers presented at meetings of the Society of Automotive Engineers, New York, and others are from three British technical journals, *The Automobile Engineer, Engineering,* and *The Engineer,* all three published in London. The assistance thus rendered is gratefully acknowledged.

THE AUTHOR.

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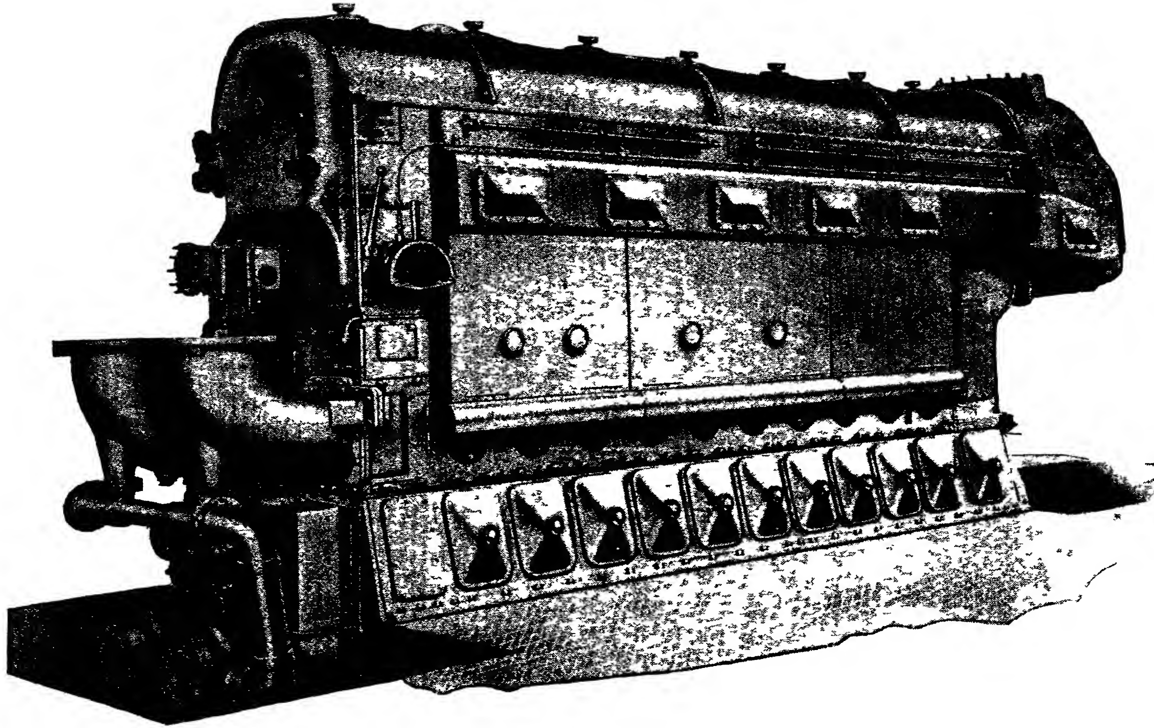


Fairbanks-Morse 10-Cylinder, 2000-HP, Two-Stroke, Marine Diesel Engine

CHAPTER I

Nature of the Engine and Its Fields of Application

A Diesel engine is a combustion engine in which the fuel is ignited by the heat of compression. To produce a temperature sufficiently high for ignition, it is necessary to carry the compression to somewhere between 350 and 600 psi, depending on the nature of the fuel, the initial temperature of the air drawn into the engine, and the temperature of the engine itself. While compression ignition eliminates the need for separate ignition equipment and therefore simplifies the engine, it is really an incidental feature of the operating cycle, the chief reason for using such high compression being that it increases the thermal efficiency of the engine; in other words, that it reduces the amount of fuel consumed per unit of work done. The use of this high compression is made possible by compressing air only, instead of a combustible mixture of air and vaporized fuel, which latter would be ignited too early during the compression stroke. Fuel is injected into the charge of highly-compressed and highly-heated air during a small fraction of the cycle beginning shortly before the end of the compression stroke. As the fuel is sprayed into the highly-heated air under great pressure, and therefore thoroughly atomized, it need not be as volatile as that required by carburetor-type engines, and Diesel fuels generally are considerably cheaper, gallon for gallon, than gasoline and similar volatile fuels.

Engine Cycles—Diesel engines, the same as other types, are usually built with a number of working cylinders, because the use of multiple cylinders increases the frequency and reduces the magnitude of individual impulses on the crankshaft, and thus makes for smoother operation. It also makes it easier to balance the inertia forces on reciprocating parts, thereby reducing engine vibration. However, the sequence of operations with single charges of air and fuel, or what is known as the engine cycle, is the same in all cylinders, and in discussing engine cycles we will therefore refer to a single cylinder.

In a Diesel engine the cylinder is first filled with air from the atmosphere. This air is compressed in the cylinder to a small fraction of its original volume. When the air in the cylinder is close to its maximum compression, fuel is injected into it by means of suitable equipment, including one or more injection nozzles in the wall of the combustion chamber. Owing to the high temperature which has been imparted to the air by its compression, the fuel ignites spontaneously and burns in the combustion chamber. Combustion of the fuel liberates a large amount of heat, most of which is imparted to the gases of combustion and increases their pressure. The gases usually attain their maximum pressure shortly after the end of the compression stroke. During the following down-stroke they expand in the cylinder, pressing against the piston and doing mechanical work on it. After the gases have been fully expanded they are expelled from the cylinder, and the cycle then starts anew.

In a four-stroke or four-cycle engine the sequence of operations outlined in the foregoing is completed during four piston strokes or two crankshaft revolutions, while in a two-stroke or two-cycle engine it is completed during two piston strokes or one crankshaft revolution. Both four-stroke and two-stroke Diesel engines have been used for a long time, but in the high-speed field the four-stroke always has predominated; that is to say, more manufacturers are building four-stroke than two-stroke engines. This may be due to the fact that many firms that took up the manufacture of Diesel engines had had previous experience in the manufacture of carburetor engines, which latter generally operate on the four-stroke cycle. Two-stroke carburetor engines are less efficient and less flexible than four-stroke, and are poorly adapted to use on road vehicles. They are being used to a certain extent in other fields, chiefly where small units are required and the service is intermittent, so that the fuel cost is not an important item.

When applied to Diesel engines the two-stroke cycle compares more favorably with the four-stroke as regards fuel economy and flexibility, and it has the advantage over the latter that power strokes are more closely spaced, hence the same degree of uniformity of power delivery can be obtained with fewer cylinders. Besides, the output per unit of piston displacement is generally considerably greater, and bearing loads are less, because in the two-stroke the bearing load due to inertia of reciprocating parts is more or less neutralized by that due to gas pressure on the piston during every stroke, instead of during every second stroke only. These advantages, however, are offset by a number of disadvantages. A blower or scavenging pump is required for the two-stroke engine, which adds to the weight and bulk of the powerplant and requires power for its operation. Heat stresses are more difficult to cope with in the two-stroke, because of the greater frequency of explosions in each cylinder, and for the same reason the wear on liners, valves, and valve-actuating mechanism per unit of time is greater.

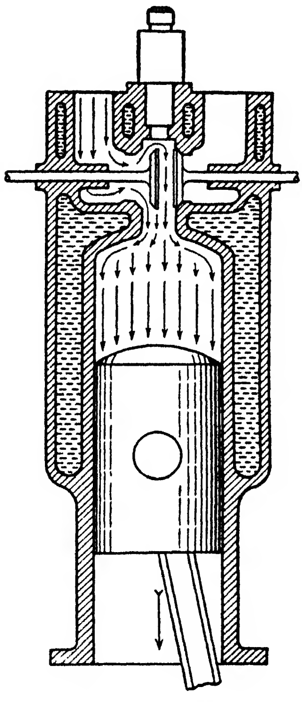


FIG. 1.—INTAKE STROKE.

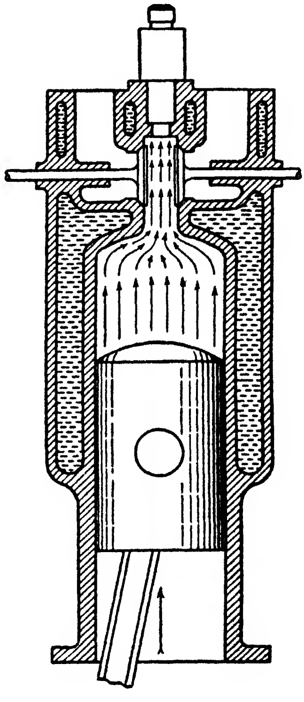


FIG. 2.—COMPRESSION STROKE.

Four-Stroke Cycle—In Figs. 1 to 5 are shown diagrammatic sectional views of a four-stroke Diesel cylinder, with the moving parts in different positions corresponding to different phases of the cycle. In this particular design the intake and exhaust valves are arranged coaxially in a pocket on the cylinder head, at right angles to the cylinder axis. It is more common to have the valves seat directly on the under side of the cylinder head and their stems extend parallel with the cylinder axis, but the arrangement illustrated makes it more convenient to show both valves and the fuel-injection nozzle in the same plane.

In Fig. 1 the intake valve is open and the piston is supposed to be moving down in the cylinder. This downward motion of the piston creates a suction effect which draws air into the cylinder through the intake valve. In a multi-cylinder engine the intake ports of individual cylinders usually are connected by an intake manifold, and atmospheric air is drawn in through this manifold and through an air cleaner connected to its outer end.

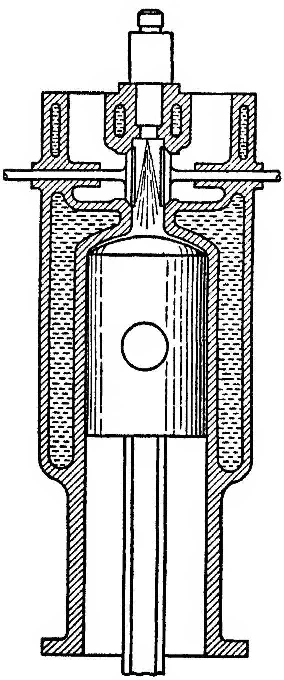


Fig. 3.—Fuel Injection.

At the end of the intake stroke the intake valve closes. During the following up-stroke of the piston (Fig. 2) both valves are closed and the air is compressed in the cylinder. As the piston approaches the end of this stroke, the air pressure in the cylinder generally is of the order of 500 psi, and the temperature of the order of 1000 F. As shown in Fig. 3, just before the end of the compression stroke fuel is injected into the compressed and heated air through an injection valve suitably located in the wall of the combustion chamber. As the ignition temperature of Diesel fuel ordinarily is materially lower than the temperature of the air at the end of compression, the fuel ignites spontaneously and burns substantially completely in the combustion chamber.

During the following down-stroke of the piston (Fig. 4), which is known as the power stroke, the burning gases expand in the cylinder and do work on the piston. From the piston the power is transmitted to the crankshaft through the connecting rod. During the greater part of the power stroke, also, both valves in the combustion-chamber wall remain closed. Toward the end of that stroke, however, the exhaust valve begins to lift, and it remains open during the following up-stroke, the exhaust stroke (Fig. 5), during which the products of combustion are expelled. They are literally pushed out of the cylinder by the rising piston. This completes the cycle.

Two-Stroke Cycle—In the four-stroke engine the cylinder acts as a pump during two of the four strokes of the cycle, drawing in fresh air during the intake stroke and expelling the gases of combustion during the exhaust stroke. In a two-stroke engine, on the other hand, there are no separate intake and exhaust strokes, and a separate pump or blower must be provided to fill the cylinder with air and to scavenge it of burnt gases. In some two-stroke engines, particularly of the carburetor type, the crankcase is used as a scavenging pump.

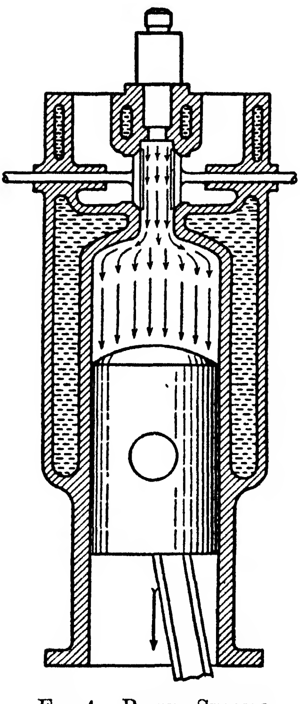


FIG. 4.—POWER STROKE.

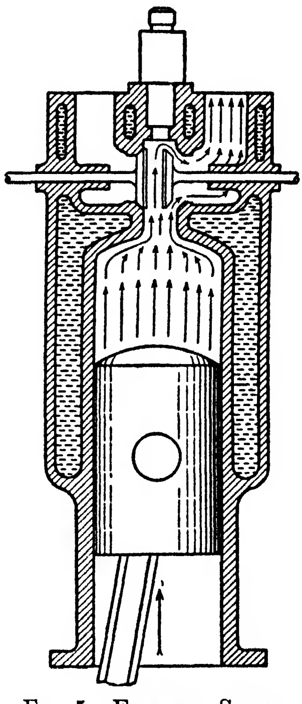


FIG. 5.—EXHAUST STROKE.

However, the maximum air pressure and air delivery obtainable from the crankcase are insufficient for best results, and on modern Diesel two-stroke engines a separate blower is employed.

Fig. 6 is a diagrammatic sectional view of a modern two-stroke Diesel engine. In engines of this type, intake and exhaust may take place either through poppet valves, as in the four-stroke, or through ports in the cylinder wall which are uncovered by the piston at the proper moments. In the engine represented in Fig. 6 the cylinder is charged through ports at the bottom, and the exhaust escapes through valves in the cylinder head. As it is desirable to have the fuel nozzle in the center of the cylinder head, two exhaust valves are located in the cylinder head in this design, on opposite sides of the fuel nozzle, both being opened and closed simultaneously. In some larger engines three or four exhaust valves are used.

At the right of the engine cylinder in Fig. 6 is shown a Roots-type blower, which consists of a housing enclosing two lobed rotors geared together. This blower operates on the same principle as a gear-type oil pump, air being transferred

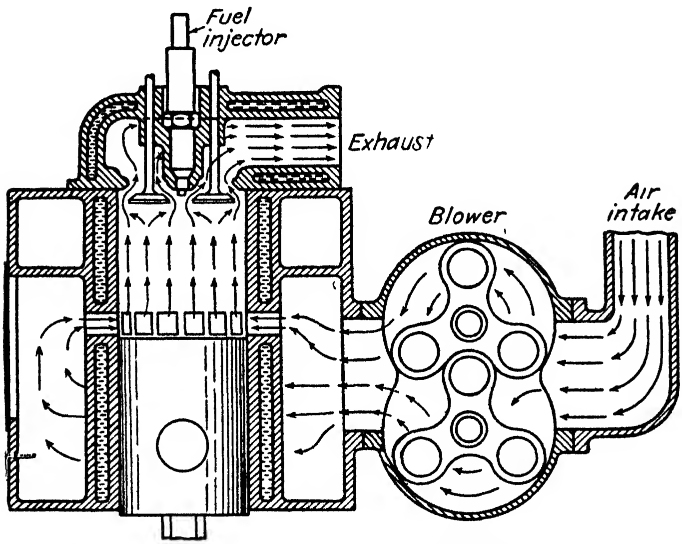


Fig. 6.—Scavenging Process in Two-Stroke Engine.

from the intake to the delivery side in the spaces between the housing and the lobes of the rotors. The air is compressed and delivered to an air chamber or air box which surrounds the lower part of the cylinder.

When the piston has completed about 70 per cent of its down-stroke (the power stroke), the exhaust valves begin to open, allowing the burnt gases to escape and the pressure in the cylinder to drop. A short time later, when the cylinder pressure has become approximately equal to the pressure in the air chamber, the ports at the bottom of the cylinder are uncovered by the piston. As the exhaust valves remain open, the pressure in the cylinder drops further, and during the remainder of the down-stroke and the first part of the up-stroke the remaining burnt gases are swept from the cylinder by air entering through the scavenging ports. In Fig. 6 the piston is at the bottom of the stroke, and both the ports and the valves are fully open. During the scavenging period some of the air entering the cylinder at the bottom escapes through the valves at the top. As a result, all or practically all of the dead gases are swept from the cylinder, and the latter is completely filled with fresh air. It may even be supercharged, depending on the relative timing of exhaust-valve closing and scavenging-port closing. Moreover, by

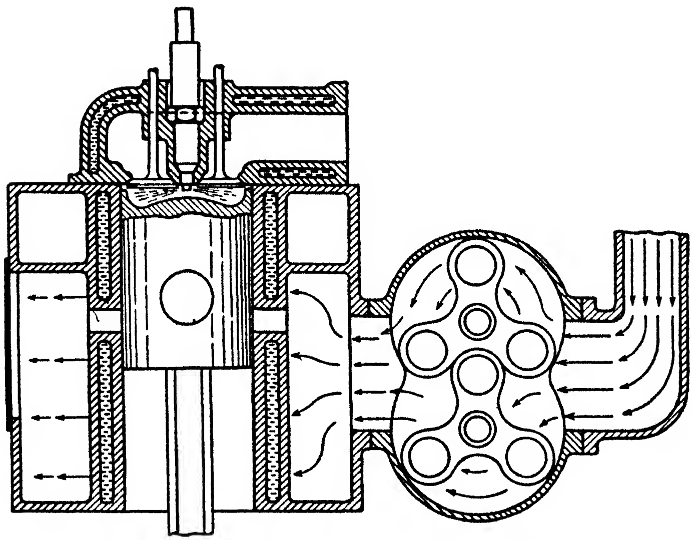


Fig. 7.—Fuel Injection in Two-Stroke Engine.

blowing fresh, cool air through the cylinder, the internal parts are cooled, which permits of operating the engine at a higher speed than would otherwise be permissible.

When the piston has completed about 15 per cent of the up-stroke, the ports and valves close in quick succession, and during the remainder of the up-stroke the air is compressed in the cylinder. Toward the end of the stroke (Fig. 7) fuel is injected into the combustion chamber, where it is ignited and burned. Then follows another power stroke.

It is obvious from the foregoing that a Diesel engine requires neither a carburetor nor an ignition unit, which makes for simplicity. On the other hand, a fuel pump and a number of injection nozzles are needed. The injection pump generally is not unlike an ignition magneto in form, and, like the latter, it requires a drive from the crankshaft. Injection nozzles resemble spark plugs to some extent, and, the same as in the case of the plugs, either a single nozzle may be used per cylinder (which is the usual arrangement), or there may be two or more nozzles in each cylinder.

Early Development—The engine received its name from its inventor, Dr. Rudolf Diesel, who in 1893 published a book under the title, "Theory and Construction of a Rational Heat Engine." Diesel applied for patents and interested two German firms in his design. Great difficulties were encountered during the early developmental period, and the first practical results were achieved only in 1897. Many things were tried during the four-year experimental period, including the use of powdered coal as fuel, and so-called "solid" injection of liquid fuel into the combustion chamber by means of a pump. Coal always contains abrasive mineral matter, and it is quite conceivable that it caused rapid wear of cylinder walls and possibly also of metering and feeding apparatus. As regards the solid injection of liquid fuel, it appears that machine-shop techniques of the period were not equal to the task of producing the high-precision pumps required to accurately meter and inject under high pressure the minute quantities required per cycle. Success was finally achieved by metering each charge of liquid fuel into a chamber in the cylinder head at low pressure, and then, at the proper moment, blowing it into the combustion chamber by means of a jet of air from a source of compressed air maintained at a pressure about twice that of the compression pressure in the engine. After a workable engine had been produced, licenses for its manufacture were let in different countries, and from that time on its development proceeded at a rather rapid rate. That development, however, was entirely in the direction of large, low-speed engines, first for stationary and then for marine work.

Owing to the high compression employed, there was always a possibility of accidental, very high explosive pressures, due either to the absorption of oil vapor by air entering the cylinders (fuel-injection air, in compressors) or to excessively early injection of the fuel charge; and to guard against disastrous results from such accidents, Diesel engineers became accustomed to designing almost without regard to what in the automotive industry is known as weight efficiency. In stationary powerplant engineering weight does not count, except in so far as it may affect the first cost of the engine;

and in marine work the reduction in the weight and bulk of the fuel that must be carried for a long trip, made possible by the adoption of the Diesel engine, was so important that it far overshadowed any possible saving in weight in the engine itself.

Carnot Cycle—Diesel's original plan was to make use of the Carnot cycle, which for a given temperature range gives the highest thermal efficiency of all known heat-engine cycles. With this cycle, the gaseous charge in the cylinder is first compressed isothermally, or at constant temperature. This requires that the heat generated by compression be removed as fast as it is produced, and Carnot, who thought in terms of the hot-air or "caloric" engine, proposed to do this by placing a "refrigerator" under the cylinder. In an internal-combustion engine, about the only way the heat could be eliminated as sensible heat would be by injecting water or some other volatile, non-combustible liquid into the charge during this part of the cycle. When the compression stroke is partly completed, the refrigerator is removed, and during the remainder of the stroke the charge is compressed adiabatically, that is, without loss of heat.

From the beginning of the expansion stroke heat is supplied to the charge in the cylinder, from a source of supply underneath the cylinder according to Carnot's proposal. In an internal-combustion engine, fuel would be injected to supply the heat, and the rate of injection would be so regulated as to keep the temperature of the charge constant. After a certain part of the expansion stroke had been completed, fuel injection would be stopped, and from that point on the charge would be allowed to expand adiabatically, that is, without the addition of further heat and without loss of heat to the outside. Thus the diagram of the cycle (Fig. 8) consists of two isothermal and two adiabatic curves. The dotted lines in Fig. 8 represent continuations of the isothermal curves.

The chief practical disadvantage of this cycle is that, in order to obtain an appreciable amount of power from a cylinder of moderate size, it is necessary to use an enormous maximum pressure. This is reflected by the "leanness" of the diagram, or by its relatively small area, which is a measure of the work done by the engine per cycle. Not only do such high working pressures necessitate very heavy construction, but they make it difficult to seal the combustion chamber effectively, and they also decrease the mechanical efficiency, so that any gain in thermal efficiency is likely to be more than offset by increased mechanical losses.

Constant-Pressure Cycle—Recognition of these defects of the Carnot cycle led Diesel to adopt the constant-pressure cycle. An ideal constant-pressure diagram is shown in Fig. 9. Here the air is compressed in the cylinder adiabatically throughout the compression stroke, from *1* to *2*. Then, as the piston starts on the expansion stroke, as far as point *3*, fuel is injected at such a rate that the heat of its combustion keeps the pressure in the cylinder constant. This involves a more rapid rate of injection than with the Carnot cycle. From the point at which fuel injection ceases, which may be at about

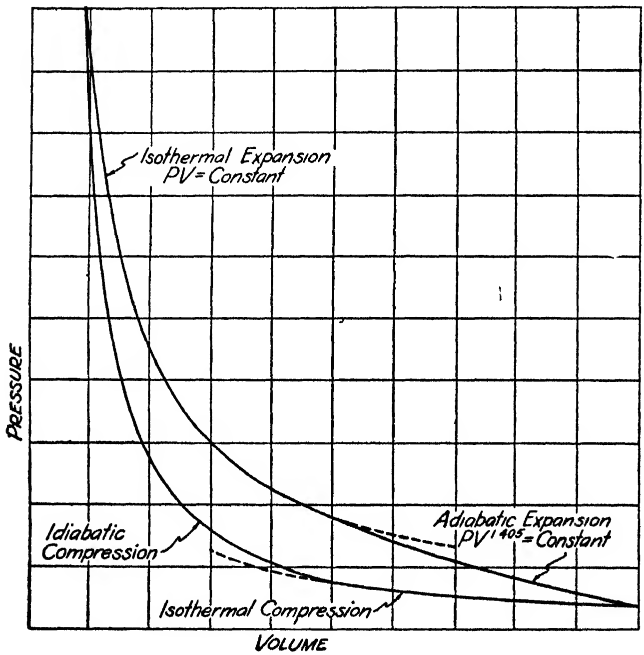


Fig. 8.—Diagram of Carnot Cycle.

one-tenth of the stroke from the dead center position, the expansion continues abiabatically until the end of the stroke (point *4*). Then the heat remaining in the gases is discharged at constant volume with the burnt gases. Thus the constant-pressure cycle diagram consists of an adiabatic-compression line, a straight horizontal constant-pressure line, an adiabatic-expansion line, and a straight vertical constant-volume line.

Ignition Lag—While it is possible to approach this theoretical cycle in practice with large, low-speed engines, with high-speed engines it is impossible of realization, owing to the fact that there is always a certain ignition lag. Combustion does not start at the moment fuel injection into the cylinder begins, and does not end when injection ceases. In some engines the delay is practically independent of the speed and corresponds to a much larger crank angle at high than at low speed. This makes it necessary to start injection in a high-speed engine before the end of the compression stroke. If ignition delay were disregarded and the fuel injected only from the beginning of the expansion stroke, some of the heat of combustion would be imparted to the charge only toward the end of the expansion stroke, which would make efficient conversion of this heat into mechanical energy impossible.

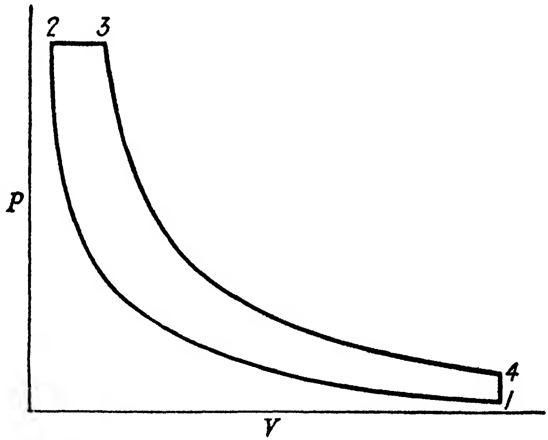


FIG. 9.—DIAGRAM OF CONSTANT-PRESSURE CYCLE.

It can easily be seen that if the fuel were injected into the cylinder so early during the compression stroke that its combustion were practically completed at the beginning of the expansion stroke, then we would have virtually a constant-volume cycle. The objection to this would be the same as that to the Carnot cycle. The constant-volume cycle also has the advantage over the constant-pressure cycle of a higher thermal efficiency, but the maximum pressure in the cylinder would be exceedingly high, with the same consequences as pointed out in connection with the Carnot cycle.

Mixed Cycle—For this reason actual high-speed Diesel engines are operated on a mixed cycle, sometimes called the Sabathé cycle. All or practically all of the fuel is injected before the completion of the compression stroke, but owing to ignition lag, combustion and consequent heat addition begin only at or very slightly before the beginning of the expansion stroke, and while the heat is not added instantly or at constant volume, it is added sufficiently rapidly to cause an increase in pressure in the combustion chamber beyond the maximum pressure of compression. The use of this mixed cycle in high-speed Diesel engines is not a deliberate choice but an absolute necessity. According to whether injection takes place earlier or later in the cycle, the actual operating cycle will approach more closely to the constant-volume and the constant-pressure cycle, respectively.

Fuel Economy—Since their introduction, Diesel engines with aggregate ratings of many million horse power have been produced, and they have become firmly established in the railroad, marine, stationary, and automotive fields. In the great majority of cases the choice of the Diesel is based on its higher fuel economy, or, more specifically, on its lower fuel cost per net horse power-hour. Only in the aircraft field did the Diesel prove unsuccessful. Its specific consumption is only slightly less than that of modern carburetor-type aircraft engines, and the slight saving on fuel costs that is possible is not sufficient to overcome the disadvantages of greater weight, higher cost, greater roughness in operation, and greater starting difficulty under adverse conditions. In military aviation, moreover, the tendency is away from piston-type engines and toward jet propulsion, which further lessens the incentive to develop Diesel engines for aircraft.

Interesting figures with respect to the comparative efficiencies of different types of combustion engines were furnished by a farm-tractor competition held in England in the Fall of 1930. In the bench trials or belt-horse power trials, a total of 31 different engines were tested, of four different types, as follows: Twelve kerosene engines, ten gasoline engines, five Diesel and four hot-bulb engines. The average fuel consumptions of the four classes at full load and at one-quarter load are given in the following table:

SPECIFIC FUEL CONSUMPTION IN LB PER HP-HR

Type of Engine	*Full Load*	*One-Quarter Load*
Gasoline	0.77	1.97
Kerosene........	0.79	2.80
Hot-bulb........	0.83	1.69
Diesel......... ...	0.58	0.88

It will be seen that the Diesel engine consumes only about 75 per cent as much fuel as the gasoline engine under full load, and considerably less than half as much at one-quarter load. At low loads the kerosene engine is the least efficient, which, of course, is due to difficulties of vaporization and distribution. The hot-bulb type, according to these tests, is slightly less efficient than the gasoline engine at full load and more efficient at one-quarter load. The tests, of course, were made under strictly comparable conditions.

Fuel Consumption-Load Characteristics—In the foregoing the specific consumptions of various types of engines are compared on a weight basis, the reason being that in dynamometer or laboratory tests the fuel consumption is always determined by weighing. However, fuels are bought on a

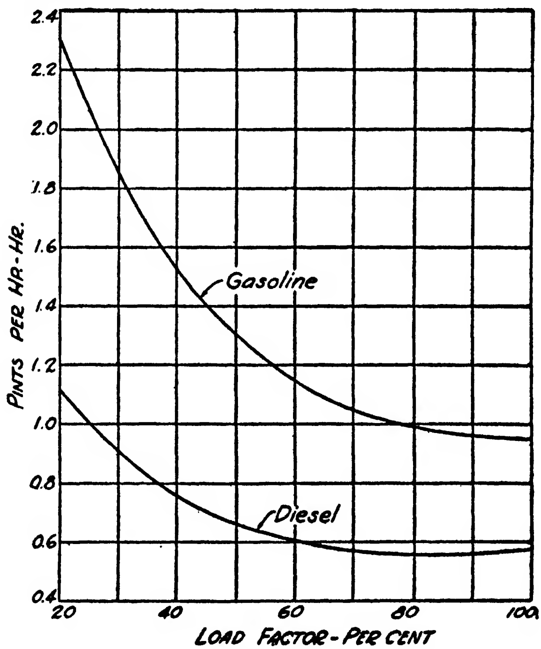


Fig. 10.—Fuel Consumption of Diesel and Gasoline Engines.

liquid-measure basis, and since Diesel fuel is materially denser, or of higher specific gravity, the comparison is even more favorable to the Diesel when made on this basis. Fig. 10 shows consumption curves of a Diesel and a gasoline tractor engine based on official reports of the University of Nebraska. Both engines were of the same make (Caterpillar) and of practically the same output.

Two remarks are in order in connection with Fig. 10. In the first place, the consumptions shown for both types of engine are rather higher than most catalogue figures, which is due to the fact that in the Nebraska tests fuel-consumption figures are obtained under normal operating conditions, whereas in factory tests the engines sometimes are cooled by running water, so that it is not necessary to drive the fan, and the muffler is left off. Both of these expedients increase the brake horse power, and therefore decrease the specific fuel consumption. Secondly, the Diesel engine from which the results plotted in Fig. 10 were obtained is a precombustion-chamber engine, which is admittedly the least efficient type of Diesel.

In Germany the Daimler-Benz Company manufactures a light passenger car which it equips with either a gasoline or a Diesel engine, at the option of the purchaser. Both engines have the same piston displacement of 108 cu in., but the shipping weight of the Diesel car is slightly greater, 1826 lb instead of 1780. Advertised fuel mileages of the two cars at different speeds (miles per U. S. gallon) are substantially as follows:

Speed, mph....	15	20	30	40	50	60
Gasoline.......	32	33	33	31	28	20.5
Diesel.........	60	58	53	47	38	27.5

Non-Poisonous Exhaust—An advantage of the Diesel over the spark-ignition engine from the standpoint of public health is that its exhaust contains very little of the highly poisonous gas carbon monoxide, of which there is usually a considerable proportion in the exhaust from the spark-ignition engine. Tests of exhaust gases from the engines of city buses were made in Manchester, England. Samples of the exhaust were taken under six different conditions of operation, varying from idling to full load.

Since the Diesel engine always operates with an excess of air, while the opposite usually holds in the case of the spark-ignition engine, the constituents of atmospheric air (oxygen and nitrogen) naturally were found in considerably greater proportion in the exhaust of the Diesel engine. Under three of the six conditions of operation no carbon monoxide could be detected in the exhaust from the Diesel, and under the other three conditions it ranged from 0.1 to 0.2 per cent, while in the exhaust from the spark-ignition engine there was carbon monoxide in the exhaust under all conditions, the average percentage being 6.1.

The U. S. Bureau of Mines also made a study of the constitution of exhaust gases from four-stroke Diesel engines, because of the increasing use of such engines in mines and tunnels, and the curves of Fig. 11 are based on its results. The tests were made with a fuel of 38.8 A.P.I. gravity (0.83 spec. gr.), 78 cetane number, 14 per cent hydrogen content and

19,910 Btu per pound heat value. The chemically-correct mixture ratio of 14.75 is indicated by a vertical line on the chart. A Diesel engine, however, never operates with so rich a mixture. Within the range of mixtures used in regular operation the carbon-monoxide content never exceeded 0.12 per cent.

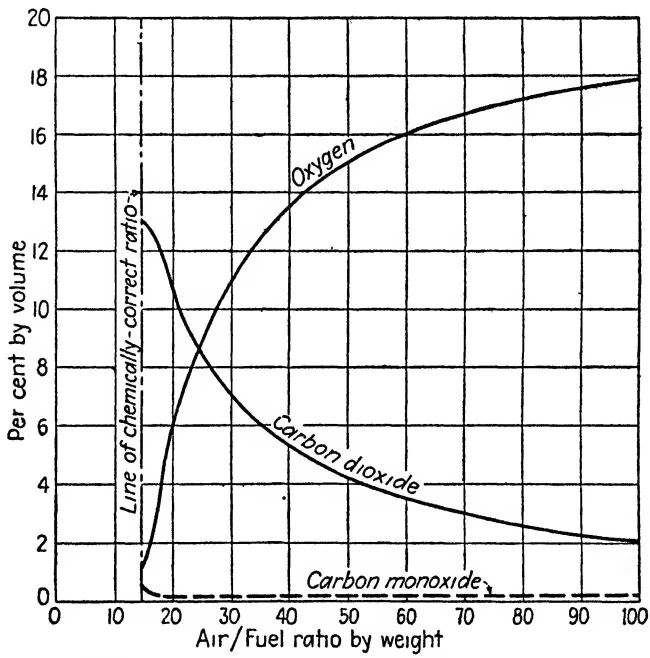


FIG 11.—VARIATION OF EXHAUST COMPONENTS WITH ENGINE LOAD.

Railway Applications—In no other field has the Diesel engine achieved so striking a success as in that of railway rolling stock. There are essentially four applications of these engines in the railroad field—to switching locomotives, to rail cars, to streamlined trains, and to main-line locomotives. What makes the Diesel attractive as a powerplant for switching locomotives is the fact that fuel consumption ceases the moment the engine is shut down. From the beginning of the railway era, steam locomotives had been used for switching cars in the yards. Since these locomotives were in actual service only a small fraction of the time, but had to be kept under steam continuously, the fuel cost was an important item in their operating expense.

According to F. G. Gurley of the Burlington Lines, which have had Diesel switching locomotives in service since 1934, the hourly cost of a steam switching locomotive during the middle forties was $4.215, while that of one of the Diesel switching engines was only $2.004. The greatest saving, of course, was on fuel, which cost $1.438 per hour for the steam locomotive and only $0.229 for the Diesel. Repairs also were much less on the Diesel, $0.36 per engine-hour as compared with $0.876, the latter item including both running and general repairs. There was also a considerable saving on wages, as the Diesel requires no fireman. Depreciation was figured at $0.046 for the steam locomotive and $0.363 for the Diesel, which certainly does not favor the latter.

In mid-summer 1945 there were 2560 Diesel-electric switching locomotives in service on the railroads of the United States, and by 1952 steam switching locomotives had been nearly completely replaced.

Closely related to the railroad switching locomotive are the mine locomotive and the locomotive for narrow-gauge railroads, and in these fields also the Diesel engine has secured a firm footing, especially abroad.

Rail Cars—The high cost of running steam trains and the competition of motor buses and private automobiles rendered the operation of many branch lines unprofitable during the period following World War I. The result was that in many cases steam trains on such lines were replaced by rail cars. Most of the earlier rail cars were equipped with gasoline engines, for the simple reason that suitable Diesel engines were not available at the time.

The thermal efficiency of a steam locomotive of the type usually employed on branch lines is very low, not exceeding 6 per cent, figured on the power developed at the rails. On the same basis it is possible to get an efficiency of 20 per cent with gasoline engines, and 25 per cent with Diesel engines. Aside from its higher fuel economy the Diesel engine has the advantage of greater safety over the gasoline engine for this work.

The National Railways of Canada began using Diesel-engined rail cars about 1925, and in 1930 they had about a score of such cars in service, equipped with Beardmore engines. There are similar services all over the world, and in many cases the more frequent and faster schedules which the individual rail cars, or light trains composed of a motor car and a trailer, made possible, resulted in a considerable increase in traffic. In the United States this field is rather limited, because the competition of private automobiles and motor buses has cut deeply into the available volume of traffic, and many branch lines were abandoned during the depression of the thirties.

Streamlined Trains—A new note was struck in rail transportation by the placing in service in 1932 of the Flying Hamburger, a high-speed "articulated" train composed of two cars on three trucks, with Diesel-electric drive. A generating set comprising a Maybach 12-cylinder V engine of 410 hp was mounted on each end truck, while the central truck carried the electric motors. This train on its trial run covered the distance between Hamburg and Berlin at an average speed of 77 mph, and attained a maximum speed of 100 mph.

Among the first streamlined articulated trains placed in service in this country were the Zephyr, which was built for the Chicago, Burlington and Quincy Railroad by the Edw. G. Budd Manufacturing Co., of Philadelphia, and a three-car train built for the Union Pacific Railroad by the Pullman Company. The Zephyr, which was placed in service in November, 1934, was built principally of stainless steel and carried a Winton eight-cylinder in-line two-stroke Diesel engine of 600 hp, while the Union Pacific train was built mainly of aluminum and carried a 12-cylinder V engine of 600 hp. The original Zephyr engine weighed 12,000 lb, or 20 lb per hp. That was quite an advance over the engine of the first American Diesel locomotive of 1925, which weighed 63 lb per hp.

Diesel-Electric Locomotives—Practically all of the Diesel-electric streamlined high-speed trains proved very popular with the traveling public, and the three- and four-car articulated trains soon proved inadequate for the available traffic. In 1936 the Electro-Motive Corporation of La Grange, Ill., a subsidiary of General Motors Corporation, came out with a double-unit 3600-hp Diesel-electric locomotive capable of hauling regular passenger trains, and several such locomotives were ordered by the railroads in 1937. From that time on rapid strides were made in the application of Diesel-electric locomotives on American railroads. A triple-unit type of 5400 hp was added, and Electro-Motive Corporation locomotives are now being supplied as single-, double-, and triple-unit designs. Each unit carries two 12-cylinder V-type engines of 8-in. bore by 10-in. stroke. Originally these engines were rated 900 hp each, but later the output was boosted to 1000 hp, hence the latest double-unit locomotives are of 4000 and the triple-unit of 6000 hp. The triple-unit locomotives will handle 13 standard Pullman cars.

In mid-summer of 1945 American railroads had 642 Diesel-electric passenger and freight locomotives in service. Some of the smaller roads had entirely replaced steam locomotives, while some of the larger ones were preparing to operate their more important divisions entirely with Diesel-electric equipment. One of the advantages of such equipment is its high degree of availability. For instance, the Baltimore & Ohio, which in 1945 had 22 Diesel passenger and six freight locomotives in service between Washington, St. Louis, Chicago and Detroit, reported that the availability of the former had been 95.5 and that of the latter 93.7 per cent. Diesel switching locomotives had an availability of 95 per cent. The Diesel locomotives of this road averaged 18,000 miles per month, and a maximum of 25,000 miles had been attained.

Taking advantage of the high availability and the low maintenance cost of its two-cab, 4000 hp locomotives, the New Haven line uses them in passenger service during the day and in freight service at night. This line in 1944 had 30 such locomotives which are geared for a maximum speed of 80 mph.

In Freight Service—When the Diesel engine achieved its first major success in the railroad field, on the streamliners, many railroad men thought it was in large part due to the "build up" the engine had received in the press and the hold it had secured on the mind of the riding public; and that in freight service, where this does not count, it would not be found practical. However, the sponsors of the Diesel locomotive did not let themselves be influenced by these pessimistic views, but went right ahead and built large freight locomotives. The latter have proved especially advantageous on lines with severe grades, because of the greater flexibility of the electric drive. A number of four-cab, 5400 hp freight locomotives were placed in service in 1944. A feature of these locomotives is that they operate with electric braking on long down grades, the traction motors being converted into generators and the energy developed by them dissipated in resistors mounted under the cab roofs.

How the use of Diesel-electric locomotives increased after 1936 may be seen from Fig. 12, which shows the number of such locomotives in use by railroads, switching and terminal companies in the United States each year, up to 1947. As the chart shows, between 1946 and 1947 the number increased by 1500, and production continued to increase the following years. At the beginning of 1952 it was estimated that 19,000 Diesel locomotives were in service and that more than one-half of the country's railroad traffic was carried on with them.

Motor Trucks—Motor trucks employed in the transportation of merchandise over long distances consume large quantities of fuel, and the fuel bill can be materially cut if Diesel engines are substituted for gasoline engines. In Europe Diesel engines have almost completely replaced the gasoline engine in trucks of more than 2 tons capacity. Germany led in the application of Diesel engines to motor trucks, partly because the first successful development work on high-speed Diesels was done in that country, and partly because motor fuels were much more expensive there, gasoline selling at three to four times the prices prevailing here. As early as 1934, 89 per cent of all trucks of between 3 and 4 tons payload capacity registered in Germany were equipped with Diesel engines; 87.1 per cent of all trucks between 4 and 5 tons, and 92.3 per cent of all trucks over 5 tons.

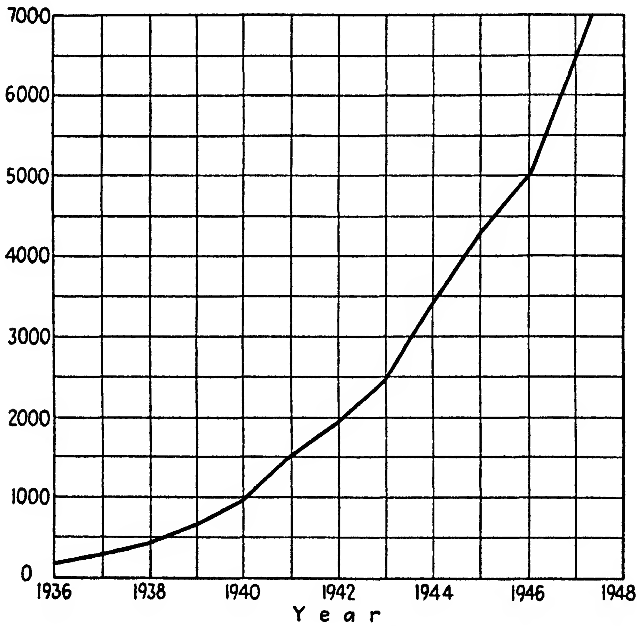


FIG. 12.—NUMBERS OF DIESEL-ELECTRIC LOCOMOTIVES IN USE BY RAILROADS, SWITCHING AND TERMINAL COMPANIES.

In Great Britain the number of registrations of new trucks with heavy oil engines (which term is practically synonymous with Diesel engines) was 1494 in 1934, 1430 in 1936, and 1891 in 1938. During the war period no registration figures were published. In Great Britain there is a sharp distinction between trucks of more and less unladen weight than 2½ tons, respectively, the former being subject to a lower speed limit and a higher tax. All trucks of the heavier class are now Diesel-equipped.

In this country the advent of the Diesel truck came some years later. In 1935 the number of such trucks in use was estimated as only about 1400. Up to the end of 1938 about 5000 additional Diesel trucks had been manufactured, but a considerable proportion of these was exported, and the number in operation in this country in that year was estimated at about 4000. The year 1938, however, marked a definite turning point in Diesel-truck history, for in that year four large American truck manufacturers (Dodge Brothers Corporation, International Harvester Co., Mack Manufacturing Corp., and Yellow Truck & Coach Manufacturing Co.) announced lines of Diesel trucks for the first time. Since then the application of Diesel engines to trucks has progressed steadily; in 1940 about 1600 Diesel trucks were produced in this country and in 1941 about 3000.

While the United States participated in World War II all manufacturers of Diesel engines produced to capacity, but most of the engines, of course, were used for war equipment. No actual figures of engine production are available for this period, but a general idea of its growth may be obtained from the following figures of metal consumption in the production of fuel-injection equipment by the American Bosch Corporation, which supplies a large proportion of all American Diesel engine manufacturers with such equipment:

Year....	1940	1941	1942	1943	1944
Tons ...	1230	2500	3660	4435	4860

The installation of Diesel engines in trucks was retarded somewhat by their considerably higher first cost, which must be amortized before there can be any real savings as compared with gasoline-engined trucks. The higher cost is due in large part to the injection equipment, which costs much more than the carburetor and ignition equipment which it replaces. Another item contributing to the increased cost is the larger starter and battery required for Diesels, which must have nearly four times the capacity of similar equipment for gasoline engines of equal output. Finally, the relatively small scale on which Diesel truck engines were built during the early years was a factor in raising their cost.

During its early development period the high-speed Diesel was greatly helped by the fact that its fuel was either tax-free or subject to a lower tax than gasoline. That, however, did not last long, and in most of the States of the Union Diesel fuel now is subject to the same tax per gallon as gasoline. In Great Britain previous to 1935 there was a tax of 8 pence per Imperial gallon on gasoline and only one pence per gallon on Diesel fuel. Under these conditions it was figured that the additional cost of a Diesel 5-ton truck was saved on fuel cost in 17,000 miles of operation. In 1935 the tax on Diesel fuel was raised to the same figure as that on gasoline, and it then took 22,000 miles of operation to compensate for the increased cost of the Diesel truck. These mileages, however, are only small fractions of the useful life of a good truck.

Economy Dependent on Intensity of Use—Whether a Diesel truck is more economical than a gasoline truck, and if so how much more, depends upon the intensity of use of the truck. Certain of the items of cost are fixed, such as interest on investment, insurance, garaging and general expenses; some vary slightly with the intensity of use, these items including driver's wages, depreciation and maintenance, while others vary substantially in direct proportion to the annual mileage, these including fuel, tires, lubricating oil and other supplies. It is obvious that if the truck is used only occasionally, the saving due to the lower fuel cost is not very great, and may not equal the additional fixed and semi-fixed costs, such as the increased interest on investment and the greater depreciation and maintenance cost, which operators seem to think must be figured with because of the relative newness of the Diesel engine, its higher working stresses, and its greater liability to obsolescence. It is interesting in this connection that in the United States the Diesel truck met its most favorable reception on the Pacific Coast, where a great deal of long-distance haulage is carried on over the highways.

At the half-century mark more than a dozen truck manufacturers in the United States were offering Diesel-powered trucks and tractors, mainly large units of from 27,000 lb gvw up, which are used predominantly in long-distance hauls. It was to be expected that the use of Diesel engines would gradually be extended in the direction of lower ratings, and in 1952 GMC Truck and Coach Division announced a line of medium-weight trucks and tractors, of from 19,500 to 35,000 lb gvw. These vehicles are equipped with a three-cylinder two-stroke GMC engine developing a maximum of 110 hp at 2100 rpm. Diesel trucks within this weight range are being offered also by one or two of the smaller producers. That competition with gasoline-engine trucks in this field is severe is indicated by the fact that one of the larger producers, who formerly offered a number of Diesel trucks and tractors of 20,000 gvw rating, in addition to an extensive line of gasoline-engined trucks, abandoned the former.

No statistics of the number of Diesel trucks in use in the United States are available, but there is no doubt that it is increasing.

Buses—The first Diesel-engined bus in the United States was placed in service by Public Service Coordinated Transport, a subsidiary of the Public Service Corporation of New Jersey, in 1929. It was a conversion, a Diesel truck engine of foreign make being installed in an old bus chassis with electric drive, and although a favorable report on it was issued some time later, no more installations were made during the next five years. The first bus carrying an American-built Diesel engine as original equipment was placed in service in Boston in 1935. This delay in the application of Diesels in bus transportation in this country was probably due in considerable measure to the depressed state of business, which discouraged bus operators from buying new equipment, and especially equipment materially higher in first cost than they had been accustomed to paying.

Between the middle of 1937 and the end of 1938 some 250 Diesel-engined buses with electric drive were placed in service in the United States. During the next three years the number of Diesel buses in operation in this country multiplied practically ten-fold. At the beginning of this period it had become the established practice in city buses to place the powerplant under the rear seat, where it does not take up any floor space that can be utilized for carrying passengers. Electric drive lends itself particularly to such installation, which calls for remote control of the powerplant. Both Westinghouse and General Electric developed automotive electric drive equipments of larger capacity (125 hp). These equipments, including the generator, motor, reversing switch and control items, weigh approximately 14 lb per hp, the generator alone weighing slightly more than 6 lb per hp.

With electric drive, since the weight of the bus is considerably increased and all of the engine power must first be converted into electric power and then reconverted, the fuel consumption naturally is increased, and some operators figure with an increase of 20 per cent. But the electric drive has many advantages from the operating standpoint, which at that time seemed to warrant both the increased first cost and the higher fuel consumption.

An interesting development was the placing in service by the Burlington Transportation Company (a subsidiary of the C. B. & Q. Railroad), in 1939, of 21 buses equipped with 165 hp two-stroke Diesel engines between Chicago, Los Angeles, and San Francisco. At first numerous operating difficulties were encountered, most of them due to overheating of the engines on long mountain grades. Close cooperation between the manufacturers of the engine and the bus, on the one hand, and the bus operator on the other, finally overcame all of the difficulties, and more buses of the same make were added to the fleet. In this service a saving of 30 per cent was effected on fuel costs, as compared with the use of gasoline buses.

Hydraulic Drive—While electric drive is very satisfactory from the standpoints of operating convenience and smoothness of acceleration, it is expensive, heavy, and not very efficient. During the 1940's automatic transmissions for passenger cars were developed, which include either hydraulic couplings or hydraulic torque converters. These offer practically the same operating advantages as the electric drive, and are lighter, cheaper to manufacture, and more efficient. Their greater efficiency is due to the fact that the greater part of the time the power is transmitted directly through either a friction clutch or a hydraulic coupling, whereas with electric drive there is a double conversion (from mechanical to electric and back to mechanical) under all conditions. Transmissions of this type were developed for use on buses, and nearly all of the Diesel-engined buses produced in 1951 were equipped with them. In 1952 five of a total of 16 U. S. manufacturers of buses offered Diesel-powered models, four different makes of engine being employed. One manufacturer (Twin Coach) had acquired manufacturing rights to a British Diesel engine (Leyland) just previously.

According to statistics gathered by *Bus Transportation*, of 9073 motor buses manufactured in the United States in 1951, 603 were exported. Of the remaining 8380, which went into domestic service, 4142 carried gasoline engines; 3607, Diesel engines, and 631 were designed for operation on liquefied petroleum gas. Of the total number, 3543 were equipped with hydraulic drives or torque converters, the remainder having conventional geared drives.

Industrial and Farm Tractors—In the United States the high-speed Diesel engine at first found its most rapid acceptance in the tractor field. The Caterpillar Tractor Company, which manufactures tracklayer-type tractors for industrial and agricultural purposes, brought out a Diesel-engined tractor in the Fall of 1931, which met with instant success. The company offered both gasoline-engined and Diesel-engined tractors of the same capacities, and in spite of the fact that the prices of the Diesel-engined tractors were about 30 per cent higher, more than 90 per cent of the demand in these sizes was for Diesel tractors, and 10,000 of these were built up to November 13, 1935. Other tractor manufacturers, encouraged by the success of Caterpillar, also took up the Diesel engine, notably the International Harvester Company, which also developed a Diesel tractor engine of its own.

As tractors are being used in the field and on construction work often far away from service stations, it is particularly necessary that need for frequent checks and adjustments on the fuel-injection apparatus be eliminated. To meet this requirement, the Caterpillar Tractor Company designed its injection pumps and nozzles so that they are interchangeable as complete units. These units are carried by dealers in their regular parts stock, and because of the complete interchangeability feature, only a small investment is required to protect a large number of units in the field. Worn-out equipment is exchanged for new on a flat-rate basis.

Diesel-engined tractors appeal particularly to industrial operators and owners of large farms producing diversified crops, who can keep their equipment in service a large part of the year; for, the same as with other Diesel-engined equipment, the economy features are effective only while the machine is at work, whereas the additional fixed charges go on whether it is working or not. Caterpillar Tractor Company, whose machines are used mainly in construction work, in 1946 stated that among its customers the demand for Diesels greatly outstripped that for spark-ignition engines and that it was all but out of the spark-ignition business.

In 1951 four out of a total of five American manufacturers of crawler-type tractors produced Diesel-engined models. In wheeled tractors, which carry smaller engines and are used in farm work for the most part, the use of Diesel engines is less common, only 15 out of a total of 121 models being Diesel-equipped in 1951. These were the products of four manufacturers out of a total of 19. The Diesel engine was making progress in this field too, because in 1949 only a single manufacturer offered Diesel-powered wheeled tractors.

Marine Applications—In the marine field the Diesel engine is firmly established so far as large vessels are concerned. According to Lloyd's Register of Shipping, on September 30, 1938, the total horse power of marine heavy oil engines, either under construction at the works or being installed aboard vessels, was nearly twice as great as the total horse power of marine reciprocating steam engines and marine steam turbines combined. According to the same source, in 1950 motor ships comprised 61 per cent of the output of merchant ships outside the "Iron-Curtain" countries (for which no figures were available). Most large marine oil engines, of course, operate at very low speeds, 115 to 150 rpm. That notwithstanding the fact that what is said to have been the first real high-speed Diesel engine was designed for marine work. It was an eight-cylinder V-type engine of about 8-in. bore by 9-in. stroke and developed about 200 hp at 600 rpm. The weight was 22½ lb per hp, and fuel injection was by compressed air. This engine was built in 1909 in St. Petersburg (now Leningrad), Russia, for installation in a yacht built for the Russian industrial magnate Nobel.

For motor boats the Diesel engine not only has the advantages of using a cheaper fuel and of consuming smaller quantities for a given performance, thus effecting a saving and making it possible to store fuel for longer journeys, but the greater safety of the less volatile fuel is an important factor. Gasoline vapor is nearly four times as heavy as atmospheric air, and if a leak occurs anywhere on the fuel system of a motor boat, the fuel vapors collect in the hold and mix with the air there, forming an explosive mixture which can be set off by the slightest flame. Every summer a number of disastrous motor boat explosions are reported from points along the sea coast and on inland waters, which can be explained only on the supposition that gasoline had leaked from the fuel system and formed an explosive mixture which became ignited accidentally. With Diesel fuels such accidents are impossible.

While the Diesel marine engine made rapid progress during the thirties and early forties, its application remained largely confined to utility and commercial boats on the one hand, and to pleasure craft of the yacht type on the other. Its application to small pleasure boats meets with some of the same obstacles as that to private passenger cars. Besides, the small motor boat is in active service for only a few months of the year, and then quite often only during week ends, so that the capital invested in it is idle most of the time and the annual fuel consumption is relatively small. Under such conditions the Diesel does not show up to best advantage. Nevertheless, the outlook for the Diesel in the marine field was encouraging and the number of Diesel engines at the national motor boat show in New York increased from year to year. In 1939 such engines were shown by 12 exhibitors.

That the Diesel marine engine is continuing its way downward into vessels of smaller displacement is indicated by the following tabulation of Diesel-powered craft produced in the United States during each of four recent years:

Year	*Vessels over 100 Gross Tons*	*Vessels up to 100 Gross Tons*
1948.....	102	1194
1949.....	89	1467
1950.....	77	1869
1951.....	81	2162

While installations in the larger vessels decreased during the four-year period, those in smaller vessels almost doubled.

In the Navy—In 1933 the U. S. Navy authorized Diesel power boats for all new cruisers and destroyers of that year's construction program. It secured a license for the manufacture of an established design, and production of these engines was begun at the Norfolk Navy Yard. During the following eleven years more than 5000 such engines for standard BuShips were produced at Norfolk, and several thousand more were purchased from an independent manufacturer.

Production of Diesel engines was greatly expanded following the outbreak of war, and in 1943 one manufacturer alone supplied more than 20,000 engines, all of the same size. In October of that year the aggregate horse power of Diesel engines under contract for installation in Navy craft for the first time exceeded the aggregate horse power of steam turbines being installed, in spite of the fact that all capital ships and aircraft carriers were still being powered with turbines. In the course of the war the U. S. Navy became the largest operator of Diesel engines in the world, and it has been estimated that from 1942 to 1946 it placed orders for Diesel engines and Diesel-powered equipment to the extent of 60 million hp.

There is little doubt that in the course of time the Diesel engine will be used also in capital ships. One of the great advantages of these engines is their instant readiness to operate under full load, which means a quick getaway for a ship lying at anchor and the ability to instantly increase speed while under way. It is the belief of some Naval officers that with Diesel propulsion such disasters as those of Oran, Taranto and Pearl Harbor could have been greatly lessened. However, the outputs required are so large and the different types of Diesel or combination Diesel and gas-turbine powerplants that might be used are so numerous that it will probably take a good many years before the changeover gets under way. The so-called pocket battle ships built by the Germans under the displacement and other restrictions of the Versailles Treaty were powered with Diesel engines.

Industrial Uses—Previous to 1935, manufacturers of high- and moderate-speed Diesel engines in the United States catered particularly to the industrial field. The engine is well suited to the operation of electric generators of moderate size in stationary and portable powerplants, because its high thermal efficiency and the cheapness of the fuel on which it operates make the operating cost quite low. Moreover, its high speed of rotation permits of direct connection to a high-speed generator, which is smaller, lighter and cheaper than a low-speed generator of equal output. Such a direct-connected generating set is compact and requires little floor space; it is particularly suited to marine applications.

Another representative industrial application is to power shovels, the equivalent of the familiar steam shovel, and most manufacturers in the United States now equip their shovels with Diesel engines. Scrapers and other earth-moving machines also are Diesel-powered, as a rule. Other industrial applications are to blowers, bridges, clay-working machines, compressors, cotton gins, cranes, crushers, ditchers, drilling rigs, excavators, flour mills, hoists, ice machines, loggers, oil pumps, paving machines, water pumps, and saw mills.

Aircraft—The only field in which the Diesel engine has not proved permanently successful is that of aviation. Development of Diesel engines for aircraft was taken up with considerable enthusiasm toward the end of the 1920's, but great difficulties were encountered and some serious set-backs suffered. In this country the Packard Motor Car Company developed a nine-cylinder radial, air-cooled engine for which it obtained an approved-type certificate from the Department of Commerce, and with which a non-refueling world's record was established; but the engineer in charge of the work, Capt. L. M. Woolson, lost his life in an airplane crash, and this, possibly in conjunction with the prolonged business depression, led to the abandonment of this enterprise. Another Diesel of similar type was developed by Guiberson Engine Company of Dallas, Tex., which is now inactive. Both the Packard and the Guiberson were of the radial air-cooled type, which dominated the aircraft-engine field during that period. As an example of a design which makes it possible to obtain a high specific output—one hp for 2 lb of dry weight—from Diesel engines, a cross section of the Guiberson is reproduced in Fig. 13. The drawing shows the counter-weighted, single-throw crankshaft, the finned steel cylinder, and the screwed- and shrunk-on aluminum cylinder head, which contained a single poppet valve that served for both intake and exhaust, remaining open for more than one complete revolution of the crankshaft. Also shown are the valve-actuating mechanism and one of the injection pumps with its drive. The Guiberson also did not reach the production stage as an aircraft engine, but in a slightly modified form was produced in considerable numbers for installation in "tanks" during World War II.

Because of its higher compression and combustion pressures, the Diesel cannot be built quite as light as a carburetor-type engine, which was probably its chief handicap. Constant improvement in aviation gasoline made it possible to use higher compression ratios and to reduce the specific consumption of carburetor-type aircraft engines, for which a specific consumption as low as 0.375 lb per bhp-hr has been claimed in at least one case. That reduced the possible gain from the adoption of the Diesel and discouraged development work in this particular field, which is of necessity very expensive. Following World War II jet propulsion came to the front in military aviation, which put another damper on development work on piston engines, and the Diesel aircraft engine is now practically extinct.

Small Diesel Engines—Small gasoline engines are being manufactured in large numbers for farm and industrial uses, for scooters, lawn mowers, etc. Diesel engines also have been built in relatively small sizes, but it is difficult for the small Diesel to compete with the gasoline engine, especially in applications where operation is highly intermittent. The Diesel costs more to buy, is more difficult to start, and does not run quite as smoothly as the gasoline engine. Its principal advantage is the saving on fuel costs which it makes possible, and if the annual consumption is low, this may not balance the disadvantages. In certain applications the greatly reduced fire hazard due to the use of a non-volatile fuel and the absence of poisonous carbon monoxide in the exhaust may make the Diesel preferable to the gasoline engine even if its overall operating cost should be somewhat higher. Another thing in favor of the small Diesel would seem to be the fact that it is easier to air-cool than a gasoline engine.

According to the U. S. Census, almost 15,000 Diesel engines with a rating of less than 31 hp were built in this country in 1947. Most of these were marine engines, or stationary engines intended for services where their reduced fire hazard counts.

The situation with respect to small engines is rather different in Europe, and especially on the Continent, where gasoline at retail sells at from three to four times as much as in the United States. Owing to lower labor costs, the difference in the prices of equivalent gasoline and Diesel engines probably is less there than here, and the saving in fuel costs which the Diesel makes possible is relatively much more important. There are numerous small Diesel industrial engines on the market in all of the industrial countries of Europe, and wheeled farm tractors, even though usually of rather small size, are Diesel-powered, as a rule.

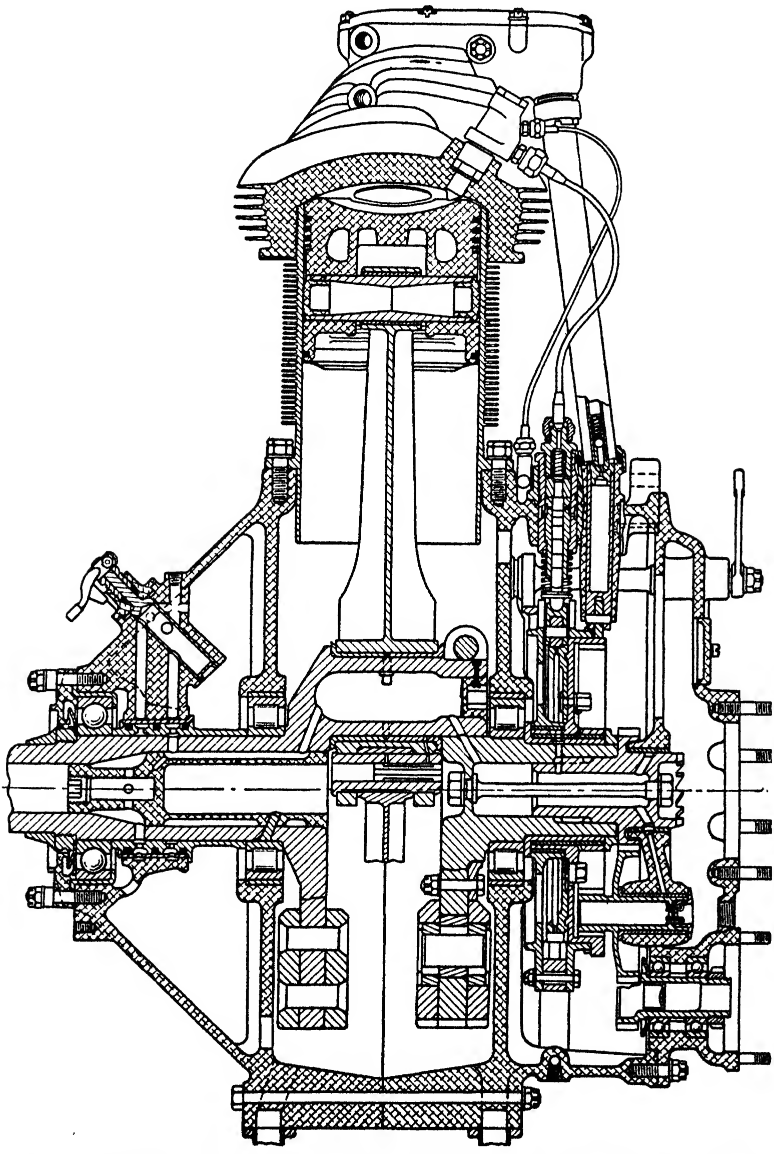


FIG. 13.—SECTION THROUGH CRANKSHAFT AXIS OF GUIBERSON AIRCRAFT ENGINE.

CHAPTER II

Thermodynamic and Other Characteristics

When air or gas is compressed in a container or pressure vessel, and is then allowed to cool to its original temperature, the compression is said to be isothermal (constant-temperature). In that case the relations between the initial and final volumes and pressures are represented by Boyle's law, according to which pressure and volume are inversely proportional. The equation of isothermal compression is

$$PV = \text{constant},$$

where P is the absolute pressure and V the volume of the air. The reciprocal process, known as isothermal expansion, is subject to the same law.

Adiabatic Compression—When air is compressed in the cylinder of a pump or compressor, both its pressure and temperature are increased, for in compressing the air, work is done, and this work is converted into heat. The rise in pressure then results partly from the decrease in volume and partly from the addition of heat to the air. It follows from this that for a given reduction in volume, the pressure of the air (or gas) will increase more if the compression is adiabatic than when it is isothermal. Similarly, for a given increase in volume there is a greater drop in pressure if the expansion is adiabatic than when it is isothermal (in which latter case heat would have to be supplied from outside). The law of adiabatic compression and expansion is

$$PV^n = \text{constant},$$

where the exponent n is equal to the ratio of the specific heat at constant pressure to the specific heat at constant volume of the particular gas under consideration. For atmospheric air n is equal to 1.405.*

* In the illustrations, which are carried over from previous editions, the polytropic exponent is represented by γ, the former standard symbol.

True adiabatic compression is practically impossible of accomplishment, for when gases are being compressed their temperature always increases and there is always at least some interchange of heat between them and the cylinder walls. In an internal combustion engine in normal operation the greater part of the combustion chamber wall is at not much over 200 F, and since in a Diesel engine during compression the air is raised to a temperature of more than 1000 F, the charge of air loses heat to the cylinder walls during a considerable portion of that stroke. Throughout the expansion stroke the temperature of the gases in the cylinder is higher than that of the cylinder wall, and there is then also flow of heat from the former to the latter. The result is that during compression in the cylinder the pressure of the air does not rise quite as much as in adiabatic compression in the same proportion. During the expansion stroke, however, the pressure does not drop as rapidly as in adiabatic expansion, because combustion continues during a considerable part of that stroke, and while the increase in volume and loss of heat to the walls result in a decrease in pressure, the continued addition of heat tends to keep the pressure up. These two effects, that is, loss of heat to the cylinder walls during the compression stroke and addition of heat during the expansion stroke, can be taken account of by giving the exponent in the equation for adiabatic changes of state a value lower than 1.405.

Pressure of Compression—The pressure in the engine cylinder at the end of the compression stroke is directly proportional to the pressure therein at the beginning of that stroke. When cranking the engine over for starting, the initial pressure (at the beginning of the compression stroke) will be substantially equal to atmospheric pressure, but at higher speeds it will be somewhat lower, depending on the speed, the ratio of valve-opening area to cylinder volume, and the valve timing. For an engine with liberal-sized valves and running at moderate speeds, the pressure at the beginning of the compression stroke may be assumed to be about 14 psi.

The pressure at the end of the compression stroke is given by the equation

$$P_c = P_i r^n,$$

where P_c is the absolute pressure in the cylinder at the end of the compression stroke; P_i, the initial pressure, at the beginning of the compression stroke; r, the volumetric compression ratio, and n, an exponent. It has been found from engine indicator diagrams that in normal operation the value of n is about 1.33. Therefore, using a value of 14 psi for the pressure at the beginning of the compression stroke and a value of 1.33 for n, we get for the compression pressure under normal operating conditions,

$$P_c = 14r^{1.33} \text{ psi abs.}$$

Values obtained by means of this equation for values of the compression ratio up to 16 are plotted in Fig. 1.

Temperature of Compression—The absolute temperature of the air in the combustion chamber at the end of the compression stroke is directly proportional to the absolute temperature of the air at the beginning of that stroke. The absolute temperature is equal to the regular Fahrenheit temperature plus 460 deg.

The temperature of the air in the cylinder at the beginning of the compression stroke varies considerably with conditions of operation. When the engine is being started from cold it is evidently equal to the atmospheric temperature, whereas when the engine is running steadily under load, the air entering the cylinder will abstract heat from the parts with which it comes in contact, namely, the inlet valve and the cylinder and piston walls. Under conditions of full load, the mean temperature of the air at the end of the inlet stroke will be about 200 F. The absolute temperature of the air at the end of the compression stroke is given by the equation

$$t_c = t_i r^{n-1},$$

where t_i is the absolute temperature at the beginning of the compression stroke and r and n have the same meanings as previously. Remembering that the absolute temperature is equal to the regular temperature plus 460°, we have for the F temperature at the end of the compression stroke

$$T_c = (200 + 460)r^{0.33} - 460°.$$

Values obtained from this equation for compression ratios up to 16 are plotted in Fig. 1. It will be seen that for a compression ratio of 16 we obtain a compression temperature under full load conditions of 1170 F. This is a much higher temperature than is actually required for the ignition of fuel oil, which has an ignition point of about 600 F, but it must be remembered that the conditions here assumed are the most favorable to the production of high temperatures.

Starting Conditions—When starting the engine the exponent will have a lower value however—not because the cylinder walls are then cold, since the temperature of the air at the beginning of the compression stroke also will be proportionately lower—but because at the low cranking speed the time occupied by the compression stroke will be so much longer, and more of the heat will pass into the cylinder walls, besides which there will be proportionately more gas leakage. For a cranking speed of 200 rpm the value of the exponent probably will not be greater than 1.20.

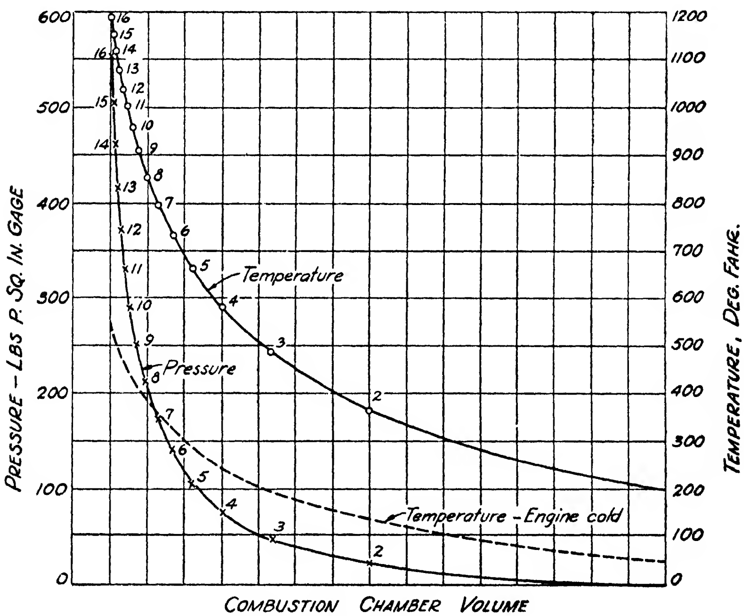


FIG. 1.—CURVES OF PRESSURE AND TEMPERATURE OF AIR DURING COMPRESSION IN A CYLINDER.

Since the initial temperature is also much lower, it seems that even with a compression pressure of 16 to 1 the temperature of compression is hardly sufficient to effect ignition of the fuel. It is rather fortunate in this connection that the temperature of the air charge toward the end of compression is not uniform. The temperature will be lowest near the cylinder walls, and there will be a considerably hotter core of air at the center of the compression space, on which dependence can be placed for ignition. But if the compression ratio is less than 15 to 1 in engines with moderate-sized cylinders, auxiliary ignition devices for starting from cold must be provided, as a rule. The dependence of compression pressure and temperature on engine speed will be discussed in more detail in the chapter on starting.

Modified Compression Ratio—The term "compression ratio" as used in the foregoing signifies the ratio of the volume of the combustion chamber at the beginning of the up-stroke of the piston to the volume at the end of that stroke. It is this ratio which is generally referred to when the compression ratio of an engine is given. However, since there can be no compression of any consequence in the cylinder until the inlet valve closes, and closing of this valve takes place only after a fraction of the compression stroke has been completed, some recent writers have used the term as meaning the ratio of the volume of the combustion chamber at the moment the inlet valve (or inlet port) closes to the volume at the end of the compression stroke.

This latter ratio is really the more rational one. In the development of high-speed engines, the moment of closing of the inlet valve is generally so set that the engine develops maximum power at high speed, which means that it takes in the greatest possible amount of air. As long as the pressure within the chamber is less than atmospheric, air continues to flow into it if the inlet valve is still open, hence, to get the maximum amount into the cylinder the inlet valve must be closed at the very moment the pressure within the chamber becomes equal to atmospheric. At low engine speeds atmospheric pressure is reached in the combustion chamber slightly earlier, and during the remainder of the period the inlet valve remains open, some air will be forced out again. This, of course, means that there is then a slight over-pressure in the combustion chamber, but since the engine speed is slow, the necessary outflow is produced with very little over-pressure, and for all practical purposes the cylinder pressure at the moment of inlet-valve closing may be considered equal to atmospheric pressure in this case too.

Variation of Compression Pressure and Temperature with Load—At a given speed, the greater the amount of fuel injected per cycle, in other words, the greater the torque load, the greater will be the amount of heat absorbed by the walls of the combustion chamber and the higher the temperature of these walls. Naturally, with an increase in wall temperature there is less loss of heat from the charge to the combustion-chamber walls during the compression stroke, which tends to raise the value of the polytropic exponent n. In Fig. 2 are given some test results and values derived therefrom by C. B. Dicksee. The engine on which the tests were made had a compression ratio of 14.5, based on the combustion-chamber volume at the moment of inlet-valve closing, and was operated at 1000 rpm. The valves were sufficiently large so that at this low speed the combustion-chamber pressure at the moment of valve closing could be taken as atmospheric.

It will be seen that the compression pressure increased from 540 psi when no fuel was injected to 610 psi at full

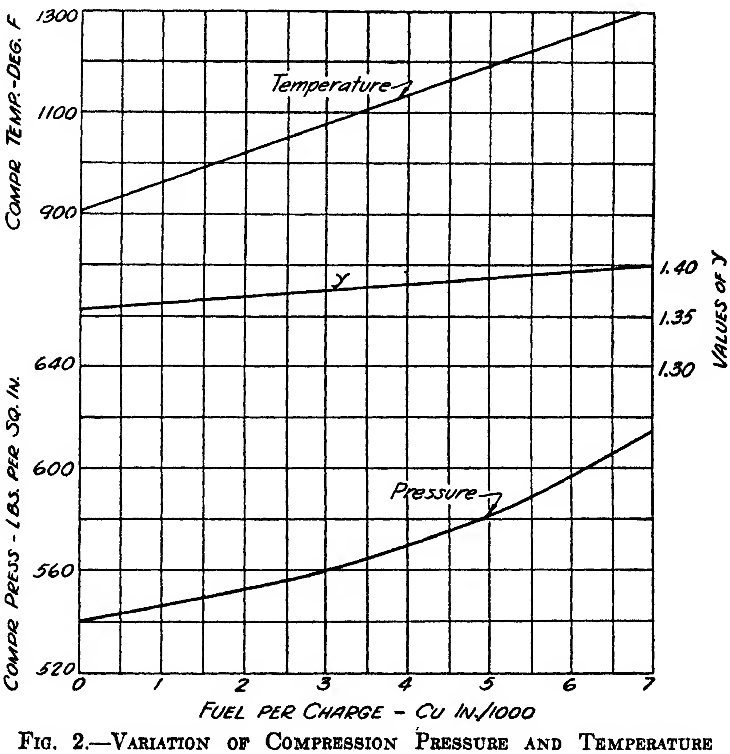


Fig. 2.—Variation of Compression Pressure and Temperature with Load (Dicksee).

load. A compression ratio of 14.5 based on combustion-chamber volume at inlet closing is equal to a ratio of around 16 based on the maximum combustion-chamber volume. The values of $n(\gamma)$ plotted are considerably higher than those given earlier in this chapter, which is due to the fact that they are based on the modified (reduced) compression ratio. The compression temperatures in the diagram were calculated from the ratio of pressure multiplication and the initial temperatures, which latter were based in part on exhaust-gas-temperature measurements. It will be seen that the final temperature of compression increases from about 900 F at no load to more than 1300 F at full load.

Thermal Efficiency of Constant-Pressure Cycle—Although the constant-pressure cycle is not used in high-speed Diesel engines, it should be of interest to discuss some of its characteristics, the more so because it is impossible to represent the actual operating conditions in a high-speed Diesel engine by a definite theoretical cycle. Referring to Fig. 3, the air is taken into the cylinder at a pressure P_0 and a temperature T_0, both in absolute units. The air is then compressed in the cylinder adiabatically (by assumption) until its pressure is P_1 and its temperature T_1. Then heat is added while the air expands, the pressure remaining the same, but the

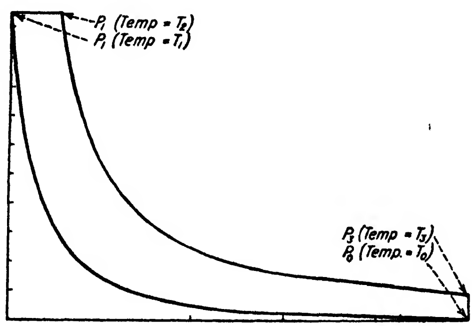


FIG. 3.—NOTATION FOR PRESSURE AND TEMPERATURE AT DIFFERENT POINTS OF THE CYCLE.

temperature rising to a higher value T_2. Next the burnt gases expand adiabatically until at the end of the stroke the pressure becomes P_3 and the temperature T_3. The gases are then discharged from the cylinder and the temperature becomes the same as at the beginning of the cycle, viz., T_0.

Now let us assume that r is the ratio of compression of the air and R the ratio in which the cylinder volume increases while heat is being added (by combustion of the injected fuel). Heat is supplied to the air at constant pressure, but is rejected at the end of expansion at constant volume. The efficiency of the cycle is

$$\eta = \frac{\text{Heat supplied} - \text{Heat rejected}}{\text{Heat supplied}}.$$

By inserting in this equation expressions for the heat supplied and the heat rejected in terms of the initial and final temperatures during heat supply and heat rejection, and of the specific heats at constant pressure and constant volume, respectively, and reducing to the simplest form, we get

$$\eta = 1 - \frac{R^n - 1}{nr^{n-1}(R - 1)}.$$

By using in the foregoing equation a value of 1.405 for n we obtain what is known as the air-cycle efficiency (constant-pressure). In Fig. 4 are given curves showing the variation

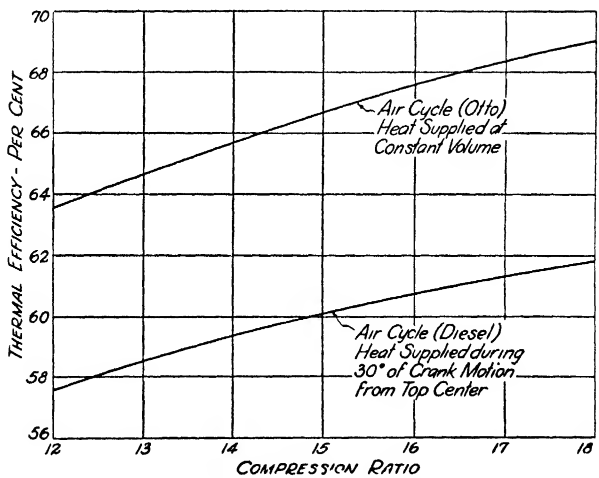


Fig. 4.—Theoretical Efficiencies of Constant-Volume and Constant-Pressure Cycles.

of the air-cycle efficiency with the compression ratio. The upper curve applies to the constant-volume cycle, where all of the heat is added while the piston is at the top of the stroke and the air therefore at its minimum volume. The efficiency of the constant-volume air cycle is calculated from the equation

$$\eta = 1 - \left(\frac{1}{r}\right)^{0.405}.$$

These efficiencies, which vary from 63.5 per cent for a compression ratio of 12 to 1, to 69 per cent for a compression ratio of 18 to 1, would be obtained under absolutely ideal conditions. In practice the efficiencies are much lower, first because usually not all of the fuel is burned; next, whatever fuel is burned is not burned instantly at the top of the stroke, and after-burning reduces the efficiency; and finally, because a considerable portion of the heat is lost through the cylinder walls. What is called the brake thermal efficiency is still smaller, being reduced by the mechanical losses (due to friction) within the engine.

The lower curve in Fig. 4 applies to the constant-pressure air cycle, and the data for it were calculated by means of the equation

$$\eta = 1 - \frac{R^n - 1}{nr^{n-1}(R - 1)},$$

heat addition being assumed to take place during 30° of crank motion from the top dead center position.

Effect of Injection Timing—In a high-speed Diesel engine the cycle of operation is intermediate between the constant-volume and constant-pressure cycles whose theoretical efficiencies are shown in Fig. 4. Owing to the so-called ignition lag, at least a part of the fuel must be injected before the piston reaches the top of the stroke, and owing to this advance in injection there is, as a rule, a considerable increase in the cylinder pressure above the maximum compression pressure. For any given engine speed, advancing the beginning of injection (within limits) raises the combustion pressure and brings the cycle of the engine closer to the constant-volume cycle, while retarding the beginning of injection brings it closer to the constant-pressure cycle. The actual effects of advancing and retarding the beginning of injection are clearly shown by Fig. 5, which is taken from an A.S.M.E. paper by John A. Spanogle on The Quiescent Combustion Chamber. In the tests, the settings of all of the engine controls remained the same throughout, except the beginning of injection. The engine was supercharged, the inlet pressure being maintained at 4.3 psi gauge, and the inlet temperature at 125 F. The beginning of injection was varied between the limits of −39° and −9° of crank angle. Points of cut-off are indicated in the chart by small circles on the respective curves, and it will be seen that in each case fuel injection extended over substantially 30° of crank motion.

An injection lead of 39° is evidently too great for the conditions of engine operation, because the pressure rises beyond the limit of the indicator even before the dead center is reached. From the standpoint of maximum power, a lead of 30° for the beginning of injection seems to be the best, but with it the combustion-chamber pressure still rises to about 950 psi. With an injection lead of 16°, substantially constant-pressure combustion is obtained, the combustion pres-

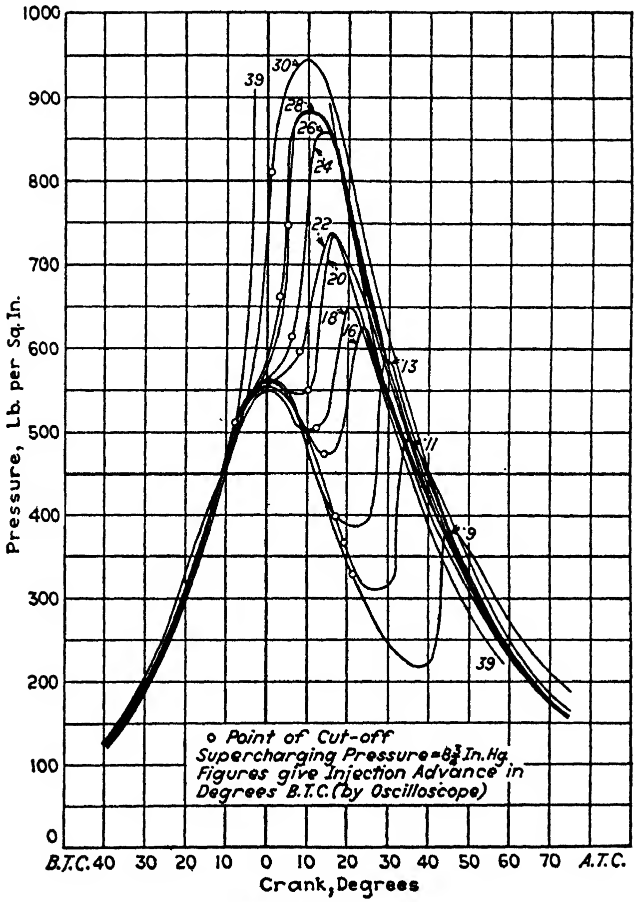


FIG. 5.—EFFECT OF INJECTION TIMING ON CYLINDER PRESSURE.

sure rising from the compression pressure of 550 psi to a maximum value of about 625 psi, after first having dropped to 475 psi. If the earliest and latest injection starts are neglected, the pressure in the combustion chamber at 50° past dead center is practically independent of the injection lead.

The diagrams of Fig. 5 are portions of so-called pressure-time diagrams, in which distances along the horizontal axis represent crank angles or elapsed times, whereas the theoretical diagrams previously given were pressure-volume diagrams, in which distances along the horizontal axis represent combustion-chamber volumes.

Effect of Cut-Off on Efficiency—To see how the efficiency varies with the cut-off, assume that the compression ratio r is 14; then for values of the cut-off ratio R of 1.25, 1.50, 2.00 and 2.50, corresponding to fuel injection during about 2.0, 3.9, 7.7 and 11.5 per cent of the stroke, respectively, the theoretical efficiencies are 63.6 per cent, 62.0 per cent, 59.2 per cent and 55.8 per cent respectively. Thus the theoretical efficiency of the engine increases as the load on it decreases. The reason for this is that the fuel as a whole is injected earlier, hence its combustion takes place earlier during the expansion stroke, and the heat generated has a better chance to perform mechanical work on the piston.

Tests of fuel consumption on Diesel engines are sometimes made under conditions of constant speed and variable load, and the consumption in lb per horse power-hour is then plotted against the torque load. Such a curve usually shows the specific consumption to be a minimum at about 80 per cent of full load and to increase as full load is approached. That is due mainly to the fact that at and near full load the fuel quantity injected is so large that not all of the fuel can find the oxygen needed for its combustion in the short time available, and the combustion efficiency is low in consequence. At less than half load the specific fuel consumption increases markedly, because the friction loss, which is practically independent of the torque load, plays a more and more important part.

In a general way the power and efficiency of a Diesel engine can be increased by advancing the injection, but unfortunately this improvement is usually accompanied by increased roughness of operation. However, when the engine is under light load it is possible to advance the injection without creating marked roughness, and a slight gain in economy can be thus obtained.

Quantity of Fuel Required—In a carburetor-type engine there is a rather definite relation between the quantities (weights) of air and gasoline that will give the highest fuel economy, because, since the air and fuel are intimately mixed previous to ignition, substantially all of the fuel can be burned in a quantity of air that contains just enough oxygen to chemically combine with the fuel. But in a Diesel engine, owing to the fact that mixing of the fuel and air and combustion go on simultaneously, it is impossible for all of the fuel molecules to find their requisite oxygen molecules if only just enough oxygen is present. For the highest fuel economy in a Diesel engine a very considerable excess of air is therefore required. In fact, if enough fuel were injected into the cylinder so that its complete combustion would consume all of the oxygen present, there would be much black smoke in the exhaust, which is indicative of unburnt carbon particles. The quantity of fuel that must be injected per cycle into a cylinder of given displacement for full-load operation can best be determined from performance data of actual engines.

Let ηp be the brake mean effective pressure in psi.
V, the displacement of one cylinder in cu in.
n, the number of power strokes per minute.
f, the specific fuel consumption in lb per hp-hr.
ρ, the specific gravity of the fuel.

The brake horse power developed by the cylinder is

$$\frac{V}{12} \times \eta p \times n \times \frac{1}{33{,}000} = \frac{V\eta pn}{396{,}000}.$$

The fuel consumption per minute is

$$\frac{V\eta pn}{396{,}000} \times \frac{f}{60} = \frac{V\eta pnf}{23{,}760{,}000} \text{ lb.}$$

Since one cu ft (1728 cu in.) of water weighs 62.5 lb, one pound of water is equal to 27.67 cu in., and one pound of fuel is equal to $27.67/\rho$ cu in. Therefore, the consumption of fuel per minute is

$$\frac{V\eta pnf}{23{,}760{,}000} \times \frac{27.67}{\rho} = \frac{V\eta pnf}{859{,}000\rho} \text{ cu in.}$$

Since there are n power strokes per minute, the volume of fuel required per power stroke at full load is

$$Q = \left(\frac{V\eta pnf}{859{,}000\rho}\right)\Big/ n = \frac{V\eta pf}{859{,}000\rho} \text{ cu in.}$$

Good average values for full-load operation in automotive-type Diesel engines are $\eta p = 80$ psi and $f = 0.44$ lb per hp-hr, while the specific gravity of the fuel oil usually employed in these engines is about 0.82. Inserting these values we get for the volume of fuel required per cycle

$$\frac{V \times 80 \times 0.44}{859{,}000 \times 0.82} = \frac{V}{20{,}000} \text{ cu in.}$$

Excess Air Factor—The proportion of excess air in the charge in any particular case for which the performance data are known can be found from the equation

$$\text{Excess air} = 100\left(\frac{859{,}000ep}{r\eta pf} - 1\right) \text{ per cent,}$$

where r is the ratio of the volume of air (at atmospheric pressure and temperature) required to completely burn one volume of liquid fuel, and e, the volumetric efficiency. For an average Diesel fuel r can be set down with sufficient accuracy as equal to 10,000, and the equation then becomes

$$\text{Excess air} = 100\left(\frac{85.9ep}{\eta pf} - 1\right) \text{ per cent.}$$

For the case for which data are given in the foregoing, if we assume the volumetric efficiency to be 80 per cent, the proportion becomes

$$\text{Excess air} = 100\left(\frac{85.9 \times 0.8 \times 0.82}{80 \times 0.44} - 1\right) = 60 \text{ per cent.}$$

Brake Mean Effective Pressure—Dr. P. H. Schweitzer in a paper read before the American Society of Mechanical Engineers has given the following equation for the brake mean effective pressure (bmep) of any type of combustion engine, two- or four-stroke, carburetor or fuel-injection:

$$\text{bmep} = 180\,\frac{0.4}{f} \cdot \frac{14.5}{r_{\text{th}}} \cdot \frac{\eta_{\text{sc}}}{1 + (\lambda - 1)\eta_{\text{sc}}}\,\eta_{\text{vol}} \text{ psi,}$$

where f is the specific fuel consumption in lb per bhp-hr; r_{th}, the theoretical air/fuel ratio or the mass of air which is just sufficient to completely burn unit mass of fuel; η_{sc}, the scavenging efficiency (defined in Fig. 6); λ, the excess-air factor $(=r/r_{\text{th}})$, and η_{vol}, the volumetric efficiency.

In the derivation of this equation use is made of the abbreviations defined in the charging diagram, Fig. 6. The horse power of the engine can be expressed in terms of the piston displacement, the bmep, and the number of cycles per minute, and it is, of course, also equal to the quotient of the total fuel consumption per hour by the specific consumption. This quotient may be expressed in terms of V_{pure}, the volume of pure air in the cylinder before combustion; ρ, the weight in lb of a cu ft of pure air at normal pressure and tempera-

DEFINITIONS:

CHARGING EFFICIENCY. $\eta_{ch} = \frac{V_{ret}}{V_{disp}}$ SCAVENGING EFFICENCY: $\eta_{sc} = \frac{V_{ret}}{V_{ret} + V_{res}}$

DELIVERY RATIO. $L = \frac{V_{ret} + V_{short}}{V_{disp}}$ VOLUMETRIC EFFICIENCY: $\eta_{vol} = \frac{V_{ret} + V_{res}}{V_{disp}}$

EXCESS AIR FACTOR: $\lambda = \frac{V_{pure}}{V_{theo}}$ UTILIZATION FACTOR: $\eta_{ut} = \frac{V_{ret}}{V_{ret} + V_{short}}$

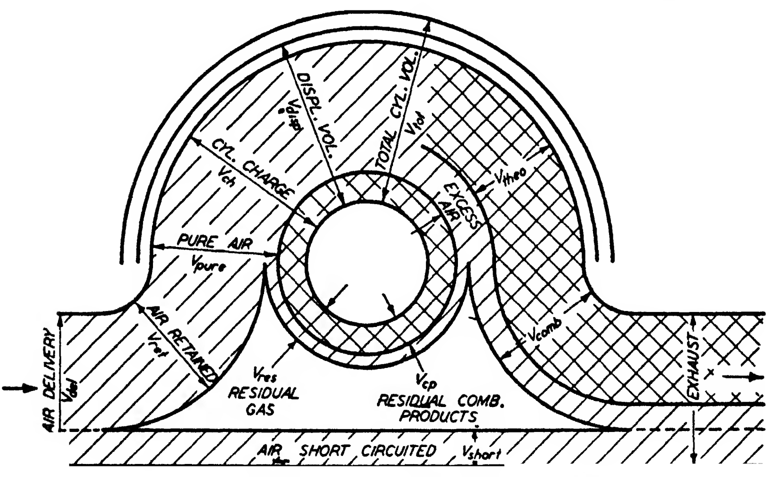


FIG. 6.—DIAGRAM OF ENGINE CHARGING PROCESS.

ture; $2n/c$, the number of cycles per minute; r, the actual air/fuel ratio; r_{th}, the theoretical air/fuel ratio; λ, the excess-air factor, and f, the specific fuel consumption. By equating the two expressions for hp, solving for bmep, inserting known values and simplifying, we arrive at the equation on page 42.

In the charging diagram V_{short}, the short-circuited air, represents the air which, after entering the cylinder, escapes through the exhaust valves or exhaust ports before compression begins, during the valve-overlap or port-overlap period. In a four-stroke engine in which the valve periods do not overlap none of the air is short-circuited, and this part of the diagram therefore disappears.

If in the equation for the bmep f is made equal to 0.44 lb per bhp-hr, η_{sc} to 1.00, r_{th} to 14.7, λ to 1.40 and η_{vol} to 0.80—figures which are representative of operating conditions in a four-stroke, automotive-type Diesel engine—the bmep comes out to about 92 psi. The so-called "continuous horse-power" rating of most automotive-type Diesel engines is based on a bmep which is within this limit, though some engines are rated at bmeps up to 100 psi, and supercharged engines even higher. If, on the other hand, the engine should require 60 per cent excess ($\lambda = 1.60$) in order to operate without excessive smoke in the exhaust, the bmep would figure out to 81 psi. The example then corresponds to that for which the excess-air factor was calculated in the foregoing.

Variation of Thermal Efficiency with Speed and Load—In Fig. 10 of Chapter I were shown curves of specific fuel consumption vs. load factor at constant speed for gasoline and Diesel engines. The specific fuel consumption f, of course, is inversely proportional to the thermal efficiency, the two being related by the equation

$$f = \frac{256{,}500}{\eta H} \text{ lb per hp-hr,}$$

where η is the thermal efficiency in per cent and H the heat value of the fuel in Btu per lb. As shown in the chart referred to, one difference between the gasoline and the Diesel engine is that whereas in the former the specific consumption decreases with an increase in the load factor throughout the range, in the latter it reaches a minimum at about 80 per cent load factor, and thereafter increases moderately. That the specific consumption is greater at low than at high load factors is, of course, due to the fact that at constant speed fuel must be burned at a substantially constant rate to supply the power necessary to overcome internal resistances, and this constitutes a greater proportion of the total fuel burned at low than at high load factors.

Fig. 7 shows the variation of the specific fuel consumption of a typical Diesel engine over a large part of its operating range with both speed and load (bmep). In this particular engine the specific consumption is a minimum (0.36 lb per bhp-hr) at a bmep of 85 psi and a speed of 1325 rpm. That the consumption increases as the bmep drops is due to the lower mechanical efficiency at low load factors. The increase in the fuel consumption at bmeps higher than 85 psi is due to less efficient combustion at high load factors, and the increase in specific consumption at speeds above 1325 rpm is due to an increase in internal losses and a decrease in the volumetric efficiency with increase in speed.

The maximum fuel quantity per cycle is limited by the stop on the control rack of the injection pump. This quantity varies with the engine speed, and the maximum bmep varies accordingly, as shown by the fuel cut-off line. The specific consumptions shown in Fig. 7 are somewhat lower than those previously given, which is due to the fact that they relate to an engine of the most efficient, direct-injection type and to the bare engine rather than to an engine driving all necessary

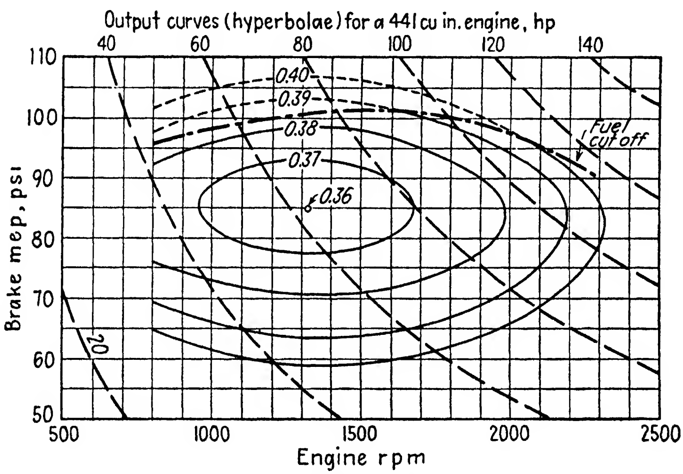


FIG. 7.—VARIATION OF SPECIFIC FUEL CONSUMPTION WITH SPEED AND LOAD.

accessories. The hyperbolae drawn in on the chart are lines of constant horse power. They show that this six-cylinder, 441-cu in. engine will develop a maximum of 110 hp at a specific fuel consumption of 0.40 lb per bhp-hr.

Effect of Atmospheric Variables on Performance—In making horse-power tests of gasoline engines it has long been customary to make corrections for deviations from standard atmospheric conditions (29.92 in. of mercury column and 60 F). However, different correction factors are used in Great Britain and the United States. The density of the atmosphere is directly proportional to the absolute pressure and inversely proportional to the absolute temperature, and the British, assuming that the engine during each cycle draws in a mass of air which is directly proportional to the density of the atmosphere, use the following equation in applying the correction factor:

$$\text{Hp} = \text{Hp}_1 \frac{29.92}{p_1} \cdot \frac{T_1}{T},$$

where Hp is the corrected horse power; Hp_1, the horse power determined from dynamometer readings; p_1, the absolute pressure in in. of mercury; T_1, the absolute temperature of the air at the carburetor inlet, and T, the absolute temperature corresponding to 60 F.

Experiments conducted in the Altitude Laboratory of the National Bureau of Standards in Washington, D. C., led to the conclusion that the engine horse power is more nearly inversely proportional to the *square root* of the absolute temperature, and in this country the following correction formula for gasoline-engine horse power has been standardized:

$$\text{Hp} = \text{Hp}_1 \frac{29.92}{p_1} \left(\frac{T_1}{T}\right)^{0.5}.$$

Circumstantial evidence in support of the latter formula is afforded by the equation for the flow of gas through an orifice, according to which the mass rate of flow is directly proportional to the area of the orifice and the pressure drop across it, and inversely proportional to the *square root* of the absolute temperature of the gas ahead of the orifice.

It seems logical to assume that the reasoning behind either one of these two correction formulae is not entirely flawless, for both have been in use for many years, and if one were absolutely rational, the sponsors of the other undoubtedly would have recognized its error before now. Against the U. S. formula, which assumes the air taken in by the engine during the suction stroke to vary inversely as the square root of the absolute temperature, it may be urged that conditions in the engine are not exactly the same as those to which the flow formula applies, as the rate of flow into an engine cylinder, in addition to depending on the pressure drop and the temperature of the air ahead of the inlet valve, is affected also by conditions beyond the valve. When the air enters the cylinder at a higher temperature, it absorbs less heat from the cylinder walls and the dead gases from the previous explosion. Thus the air in the cylinder is expanded less by heat addition during the inlet stroke, the pressure differential producing the inflow is maintained better, and the volumetric efficiency will be higher. The term "volumetric efficiency" as here used signifies the ratio of the mass of air drawn in and retained in the cylinder at the end of the intake period, to the mass of a volume of air equal to the piston displacement, at the temperature and pressure of the ambient atmosphere.

It also has been established by the National Bureau of Standards that moisture in the atmosphere reduces the output of a combustion engine by the proportion of the partial pressure of the vapor content to the total atmospheric pressure.

Conditions Different in Diesel Engines—While there has been some discussion of the effect of atmospheric conditions on the output of Diesel engines, up to the time of this writing no definite correction formula for Diesel-engine horse power seems to have been adopted anywhere. There is, of course, an important difference between carburetor and Diesel engines in this respect, for whereas in the former any increase in the mass of air inducted into the cylinder results automatically in the induction of an equally increased quantity of fuel, in the latter the quantity of fuel injected is not affected by any changes in the mass of air inducted. In the Diesel, if the amount of air taken in is increased, it is, of course, possible to burn a larger amount of fuel in it, and the maximum useful output, as a rule, is limited by smoke in the exhaust. In discussing the effects of atmospheric variables on engine output it is therefore rational to postulate a constant smoke limit.

Effects of atmospheric variables on the horse power and specific fuel consumption of Diesel engines have been investigated by Dr. Ing. K. Zinner of Augsburg, Germany,* who has proposed a correction formula for the *indicated* horse power which may be written as follows:

$$\mathrm{Hp} = \mathrm{Hp}_1 \frac{29.92}{p_1}\left(\frac{T_1}{T}\right)^{0.8}.$$

This is similar to the present U. S. correction formula for gasoline-engine horse power, except that an exponent 0.8 is used for the temperature ratio, instead of 0.5. The reasoning behind this formula is as follows: Since the atmospheric density varies inversely as the absolute temperature, if the volumetric efficiency were constant, the mass of air taken in by

* Motortechnische Zeitschrift, Sept.-Oct., 1950.

the cylinder would vary as $1/T$ or as T^{-1}. Now, as already pointed out, the volumetric efficiency increases with the temperature of the entering air, and measurements on a particular Diesel engine showed it to vary substantially as $T^{0.3}$. The mass of air taken in by the cylinder is proportional to the product of the density of the atmosphere by the volumetric efficiency; that is, to $T^{-1} \times T^{0.3} = T^{-0.7}$. Another thing that must be taken into account is that when the air enters the cylinder at a higher temperature, the temperature level of the entire cycle is higher, and as a result there will be greater heat loss to the cylinder walls. This can be accounted for by raising the value of the exponent from 0.7 to 0.8.

The foregoing correction formula applies directly only to the indicated horse power. In practice, what is usually wanted is the brake horse power, which is equal to the product of the indicated horse power by the mechanical efficiency of the engine. If the mechanical efficiency were independent of the load, the brake horse power would vary in the same proportion as the indicated horse power, and the same correction factor could be applied to both, but such is not the case. A part of the mechanical loss or friction loss varies with the load, and this results in a change in the mechanical efficiency with atmospheric conditions. Denoting the mechanical efficiency of the engine under standard atmospheric conditions by η and the ratio of the variable part of the friction loss to the indicated horse power by a, Dr. Zinner has deduced expressions for the corrected brake horse power and the corrected brake specific fuel consumption which may be written as follows:

$$\mathrm{Hp} = \mathrm{Hp}_1 \left[\left(\frac{1-a}{\eta} \right) \left(\frac{p}{p_1} \right) \left(\frac{T_1}{T} \right)^{0.8} - \left(\frac{1-a}{\eta} - 1 \right) \right]$$

$$f = f_1 \left[\frac{ \left(\frac{p}{p_1} \right) \left(\frac{T_1}{T} \right)^{0.8} \eta(1-a) }{ \left(\frac{p}{p_1} \right) \left(\frac{T_1}{T} \right)^{0.8} - \left(1 - \frac{\eta}{1-a} \right) } \right]$$

Correction-Factor Chart—Fig. 8 is a chart which shows the dependence of Diesel-engine horse power and specific fuel consumption on all three of the atmospheric variables—temperature, pressure, and humidity. Data for the chart were calculated with the aid of the two foregoing equations, the mechanical efficiency under standard atmospheric conditions being assumed to be 0.80 (80 per cent), and the ratio a of the

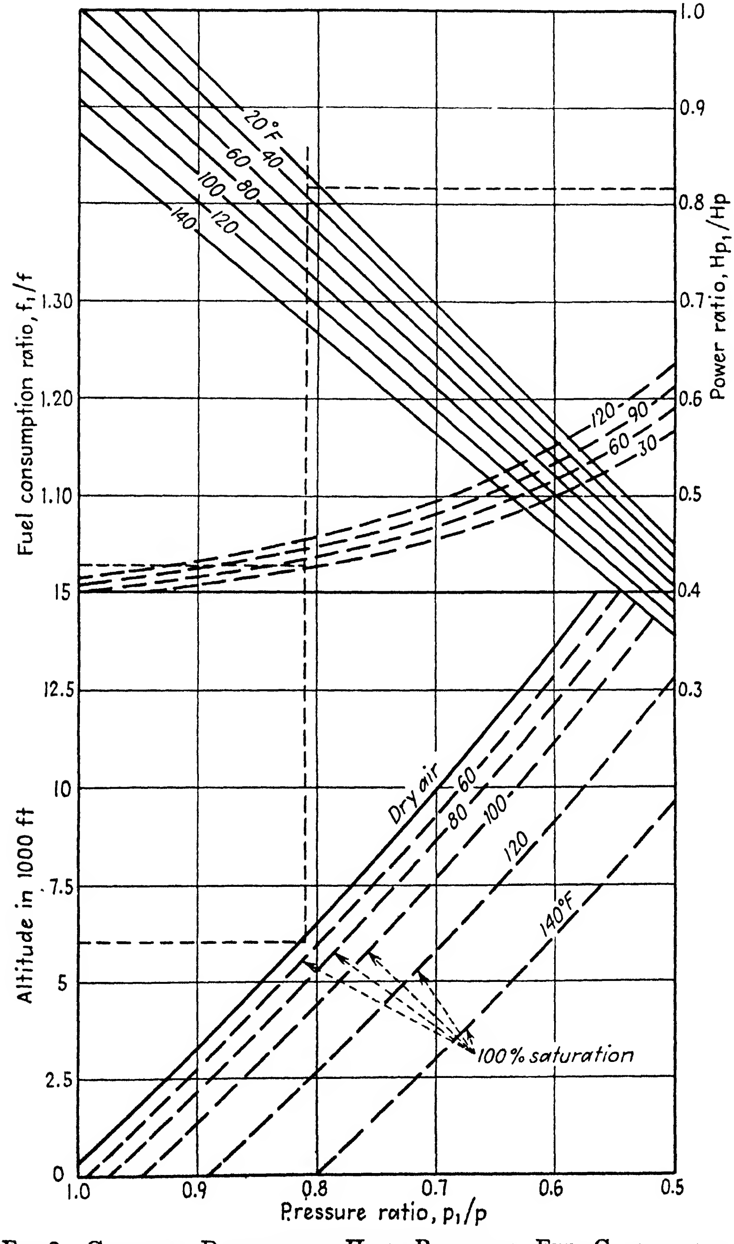


FIG. 8.—CHART FOR DETERMINING HORSE-POWER AND FUEL-CONSUMPTION CORRECTION FACTORS.

variable part of the friction loss to the indicated horse power, 0.07. Other data were obtained from tables of the variation of the mean barometric pressure with altitude and of the water-vapor content of a saturated atmosphere at different temperatures.

With the chart it is possible to determine correction factors that must be applied to the horse power and the brake specific consumption under normal atmospheric conditions (29.92 in. mercury column, 60 F, 80 per cent humidity) to obtain horse powers and specific consumptions for altitudes up to 15,000 ft, for different inlet temperatures and different degrees of humidity. Use of the chart is illustrated by an example: An engine is to be operated at 6000 ft, and it is desired to determine the available power and specific consumption in terms of the same items under normal atmospheric conditions.

In making use of the chart we have to take account of the fact that the temperature of the atmosphere decreases with increase in altitude. From a table of mean atmospheric temperatures we find that a temperature of 60 F at sea level corresponds to 36 F at 6000 ft. We may assume the relative humidity to be the same in both cases. At such low temperatures the air holds very little moisture, and humidity is a minor factor.

From the point for 6000 ft on the altitude scale on the left we proceed with a dashed line horizontally to the right to a point about one-third from the dry-air curve to the 60 F saturated curve (0.40 for the 36 F saturated curve and 0.8 times this for the 80 per cent humidity). From that point we proceed vertically upward with a dashed line cutting the two sets of curves in the upper half of the chart. From the point corresponding to 36 F between the consumption curves we proceed horizontally to the left, and from the scale there see that the specific consumption will be about 3 per cent higher than at sea level. From the point corresponding to 36 F between the horse-power curves we proceed horizontally to the scale on the right and see that the horse power will be about 82 per cent that at sea level.

The two equations given in the foregoing are applicable also when it is desired to correct dynamometer readings for deviations from standard atmospheric conditions.

Correction Factors for Two-Stroke Engines—In two-stroke engines—as well as in four-stroke with very large cylinders—the maximum practical injection quantity is usually limited by considerations of heat stresses rather than by smoke in the exhaust. The maximum injection quantity then is well below that which would result in a smoky exhaust under normal operating conditions near sea level. For that reason, if the altitude is increased above sea level, it is not necessary to immediately decrease the injection quantity. The air supply decreases, and if the fuel supply remains the same, combustion becomes less efficient and there is a decrease in both the power developed and the waste heat. The power loss, however, is not nearly as great as if the injection quantity were reduced in proportion to the air supply. At a certain altitude the density of the atmosphere is so reduced that the cylinder

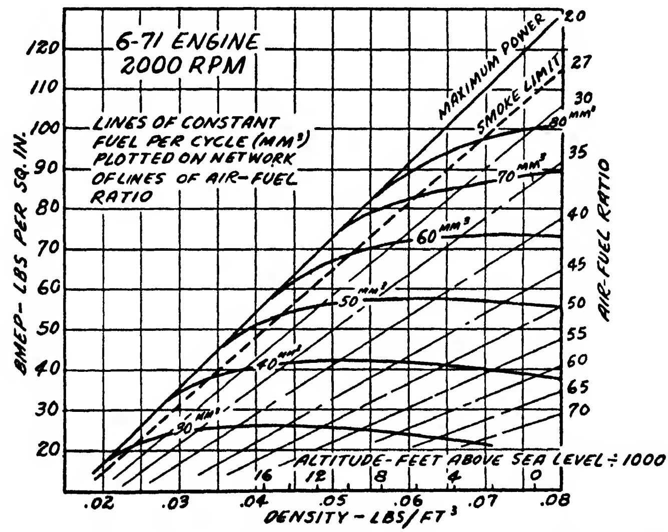


FIG. 9.—ALTITUDE CHARACTERISTICS OF A TWO-STROKE ENGINE.

receives too little air to properly burn the fuel, and beyond that point the fuel supply must be reduced. Within this range the power decreases substantially in accordance with the principles discussed in the preceding paragraphs.

Extensive tests on the effect of altitude on the power of a GM two-stroke, six-cylinder, 425-cu in. engine were carried out in 1949 and reported on in an S.A.E. paper by R. W. Guernsey. The maximum injection quantity for this engine as equipped for use on highway trucks is 70 cu mm. At the standard sea-level atmosphere and an engine speed of 2000 rpm this results in an air/fuel ratio of about 34/1, based on the total air drawn in by the blower (of which a considerable proportion passes right through and is not present in the cylinder when ignition occurs). In this engine the smoke limit corresponds to an air/fuel ratio of 27/1, which is reached at an altitude of about 6000 ft. Up to that altitude, therefore, there is no need for a change in the injection quantity, and within this range the bmep drops about 1 psi per 1000 ft. If the altitude is increased further, the injection quantity must be reduced so as to maintain the air/fuel ratio at 27/1, and between 6000 at 16,000 ft the bmep decreases approximately 3.3 psi per 1000 ft. Fig. 9, reproduced from the Guernsey paper, shows how the bmep of the engine varies with altitude when the fuel control is set to inject different quantities of fuel per cycle.

CHAPTER III

Combustion Phenomena

In the cylinders of a compression-ignition engine combustion proceeds in a manner entirely different from that in a spark-ignition engine. In the latter the fuel and air are intimately mixed before ignition occurs. Combustion starts at the spark points, and the flame travels from there through the entire charge at a velocity depending on the rate at which charge can be brought up to the ignition temperature. If the charge is quiescent, heat can be transferred from the part already burned to that which still remains to be burned only by radiation and conduction, and flame propagation is comparatively slow, which manifests itself in a slow pressure rise. If, on the other hand, the air circulates rapidly in the combustion chamber, or if there is appreciable turbulence, heat will be carried from the burning portion of the charge to the unburnt portion also by convection. This involves a much more rapid transfer of heat throughout the combustion chamber, and consequently a more rapid rate of combustion.

Ignition Lag—In a spark-ignition engine the beginning of combustion coincides with the passing of the spark. In describing the cycle of the Diesel engine we say that as the fuel charge is injected into the cylinder, it flashes into flame spontaneously. This is true in the sense that ignition is effected without the aid of outside agencies, and occurs a very short time after the beginning of injection, but there is, nevertheless, a definite, measurable time interval between the beginning of injection and the beginning of combustion as indicated by the beginning of pressure rise above the compression line. This is known as the ignition lag.

Ignition lag is a factor of great importance in the operation of high-speed Diesel engines. If it were not for this factor, the rate at which combustion proceeds in the engine cylinder could be accurately controlled by means of the injection apparatus. As it is, one has no control over the combustion of that portion of the fuel charge which is injected during the period of ignition lag. Why this is so will be plain from the following analysis of the combustion process.

Analysis of Combustion Process—Fig. 1 is a diagram of the pressure cycle of a Diesel engine, somewhat simplified or idealized to bring out more clearly the different phases of the combustion process. The curved line *ABCF* represents the rise and fall of pressure in the cylinder while the engine is being motored over without fuel injection. As the piston approaches the top dead center, the pressure rises rapidly; it attains its maximum value in dead center, when the space in which the air is confined is a minimum, and then, as the piston moves down again, the pressure decreases in the inverse manner in which it increased during the up-stroke.

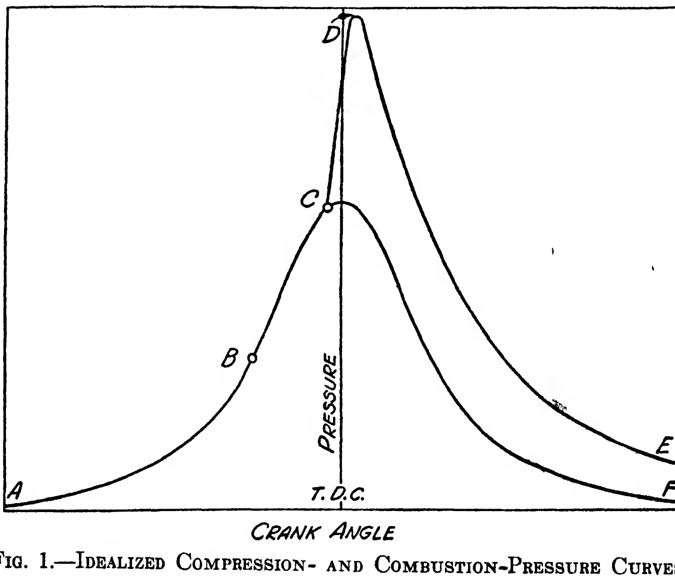


FIG. 1.—IDEALIZED COMPRESSION- AND COMBUSTION-PRESSURE CURVES.

In regular operation under power, fuel injection begins at a point of the compression stroke which is here denoted by *B*. However, for the reason already mentioned, ignition does not take place until a point *C* is reached, and during the interval *BC*, which represents the ignition lag, the pressure in the combustion chamber does not rise above the compression line. In fact, with certain engines and certain types of indicator it has been possible to discern a slight falling off of the pressure in the cylinder during this period if fuel is injected, as compared with the pressures therein when air is being compressed without fuel injection. This is due to the heat absorbed by vaporization of the fuel, which detracts from the sensible heat of the charge, and therefore from its pressure.

When ignition finally takes place, at point C, a considerable portion (if not all) of the fuel charge is already in the cylinder, and this portion burns with an explosion-like effect, resulting in a very rapid rise in pressure from C to D. It is this very rapid pressure rise which is responsible for rough operation—the so-called Diesel knock. The greater the ignition lag, the more fuel there will be in the cylinder when ignition occurs, and the greater will be the rate of pressure rise, and also, as a rule, the maximum pressure attained. Evidently, the maximum rate of combustion during this period and the maximum rate of pressure rise depend on both the length of the ignition lag in time measure and on the rate at which fuel is being injected.

The remainder of the fuel, that is, that portion which is injected after ignition has taken place, burns substantially at the rate at which it is injected, and since the volume of the combustion chamber now increases, due to the downward motion of the piston, from point D on the pressure in the cylinder drops, and the stresses induced by the rapid pressure rise during the early part of the combustion period are relieved. If the engine is to operate smoothly, the rate of pressure rise in the cylinder must be kept below a certain critical value. According to Dr. P. H. Schweitzer of Pennsylvania State College, while the critical rate varies somewhat with cylinder size and other factors, an engine ordinarily operates smoothly if the pressure rise is less than 30 psi per degree of crank motion, and knocks if it exceeds 50 psi per degree.

Effects of Air Density and Temperature on Ignition Lag —In an actual engine the ignition lag depends upon what may be called engine factors, comprising both design factors and factors of operation, fuel factors and, to a lesser degree, injection-system factors. The subject of ignition lag has been studied by many investigators. Some of the earlier experiments were made with pressure bombs or pressure vessels, while most of the later investigations were carried out on actual engines.

Ignition lag has been found to decrease as the density of the air charge and its temperature increase. The density of the air, of course, increases with its pressure, and decreases as the temperature increases, and it is proportional to the ratio of its absolute pressure to its absolute temperature. Professor Bird, at Cambridge University, investigated the relation between the ignition lag on the one hand and the density and temperature of the air on the other, and he embodied his results in a three-dimensional diagram. An equivalent of that diagram, based on Fahrenheit instead of Centigrade temperatures, is shown in Fig. 2. The ignition lag, in fractions of a second, is plotted along the vertical axis, while the temperature of the air in degrees F and the ratio of the absolute pressure to the absolute temperature (which is proportional to the density) are plotted along the other axes.

Effect of Engine Speed on Ignition Lag—The ignition lags found by Bird were quite high, which was due to the fact that his experiments were carried out with pressure bombs, and fuel was injected into air at rest. In an actual

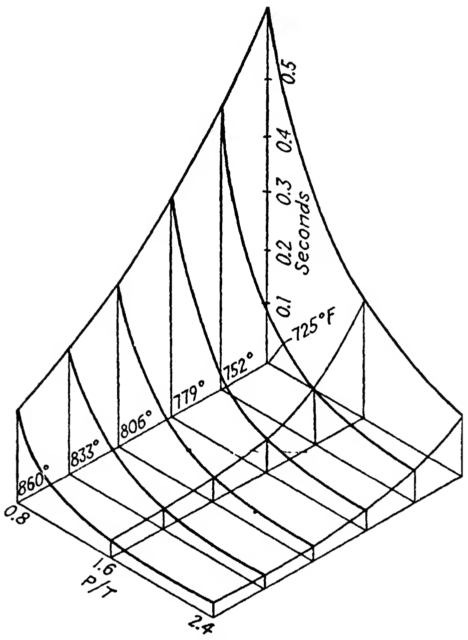


FIG. 2.—EFFECTS OF CHARGE DENSITY AND TEMPERATURE ON IGNITION LAG.

engine there is always at least some air flow, and this reduces the lag. Davies and Giffen, who examined indicator cards obtained with a Farnboro indicator on a single-cylinder Acro unit of the Associated Equipment Company, found the lag to vary with the engine speed. In the experiments carried out the fuel quantity was kept constant over the speed range 500-2000 rpm, and bmeps of the order of 80 psi were obtained. Ignition lags measured varied as follows with the engine speed, in terms of crank angle and time units:

Speed, rpm...	500	750	1000	1250	1500	1750	2000
Ignition lag, crank-angle deg	5	14	14	11	13	11	11
Ignition lag, time sec....	0.0017	0.0032	0.0024	0.0015	0.0015	0.0011	0.0009

Compared with Bird's results, these ignition lags are very short, yet they have a pronounced effect on the combustion process. The significance of the figures in the table can be

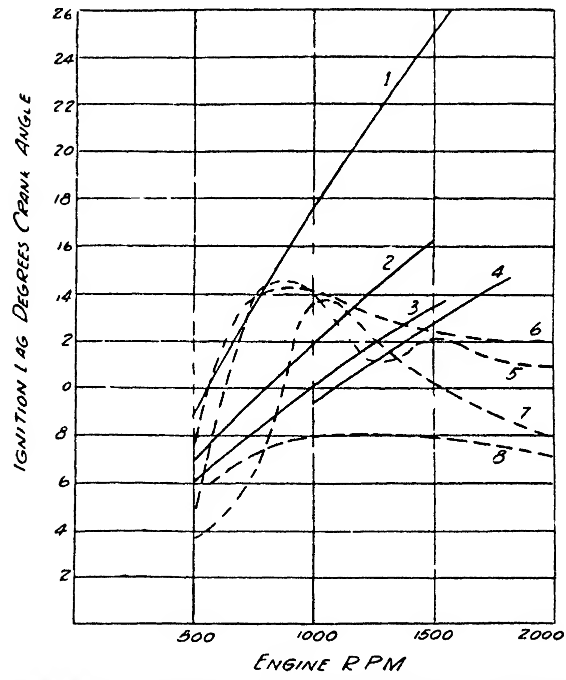


FIG. 3.—VARIATION OF IGNITION LAG, IN DEGREES OF CRANK ANGLE, WITH ENGINE SPEED.

1, 2, and 3, Le Mesurier direct-injection engine; 4, N.A.C.A. quiescent engine; 5, Davies A.E.C. engine; 6, 7, and 8, Dicksee high-turbulence engine.

better appreciated when it is considered that the injection period for bmeps of the order of 80 psi extends over about 25° of crankshaft rotation, so that over the greater part of the speed range injection is about half completed when ignition takes place.

Numerous other experiments on the relation between engine speed and ignition lag have been made in recent years, and the results of some of these are plotted in Fig. 3, which is taken from the 1934 report of the S.A.E.-A.S.M.E. Joint Committee on Diesel Fuel Research. If the ignition lag in seconds were independent of engine speed, the lag in crankshaft degrees would go up in dirct proportion to the speed. That it does not come near doing so in most engines is due to the fact that engine speed has a considerable influence on compression temperature, compression pressure, and air flow. Ignition lag in crankshaft degrees never increases as fast as the

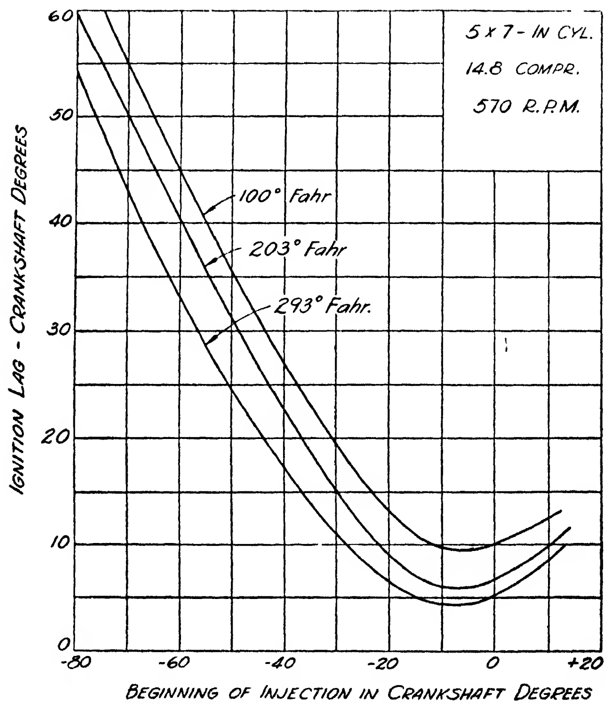


FIG. 4.—EFFECT OF ENGINE TEMPERATURE AND INJECTION ADVANCE ON IGNITION LAG.

engine speed; frequently it is practically independent of engine speed, and in some cases it actually decreases while the engine speed increases.

From the various curves of Fig. 3 the inference may be drawn that ignition lag in degrees increases with speed more rapidly in non-turbulent than in turbulent engines, and some high-turbulence engines (the A.E.C. for instance) can be so adjusted that they are rather insensitive to variations in engine speed as regards ignition delay.

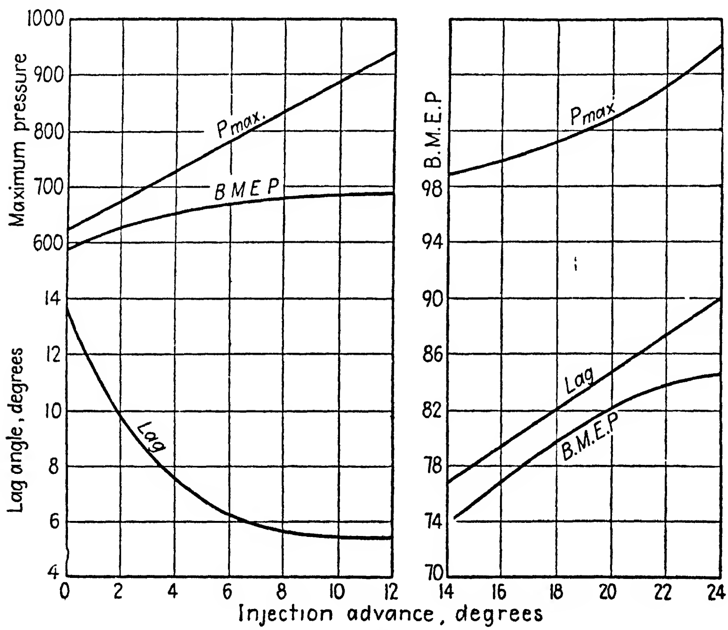
Effect of Injection Timing—In an actual engine, injection of the fuel always begins some time before the end of the compression stroke, and the air during the ignition-lag period, instead of being at substantially constant temperature, as in bomb experiments, increases in temperature during this period, due to its further compression. For this reason an abnormally early injection results in a very long ignition lag. At this early point in the compression stroke the air in the combustion chamber is still at a relatively low temperature, and heat transfer from the air to the fuel therefore can proceed only slowly. This is plainly brought out by Fig. 4, which shows the effects of both the injection advance and the jacket temperature on the ignition lag. In this particular case the ignition lag is a minimum for an injection-advance angle of about 8°. The graph also shows that the higher the jacket temperature, the lower the ignition lag.

Effect of Turbulence—The relation between injection advance and ignition lag outlined in the preceding paragraph and illustrated by Fig. 4 does not hold for all engine types, however. It seems that a vigorous air swirl in the combustion chamber changes the relationship between these two factors completely. This is brought out by characteristic curves of two engine types given in Figs. 5 and 6, which are taken from a paper by C. B. Dicksee presented to the Institution of Automobile Engineers. The curves of Fig. 5 are from an engine in which the air in the combustion chamber had a vigorous swirling motion imparted to it. Nothing was said in the paper regarding the state of the air in the other engine, but by implication it was in a state of comparative quiescence. The tests were made at fixed speeds, and constant amounts of fuel were injected, the only variable factor being the injection advance. Aside from the difference in the forms of the combustion chambers, there was a difference also with respect to the compression ratio, which was slightly higher in the case of the turbulence-type engine.

It will be seen that with the turbulence-type engine the bmep and the ignition lag in degrees are practically independent of the injection advance over a considerable range, whereas in the quiescent engine both increase rapidly with the injection advance. In the former engine the ignition lag actually decreases rapidly with increase in injection advance for small values of the latter. In both engines the injection advance was increased until the maximum pressure reached a value of 950 psi, and this required a much greater advance in the quiescent than in the turbulent engine; in fact, the maximum injection advance at which the turbulent engine was operated is less than the minimum for the quiescent engine.

From the foregoing it may be concluded that the effect of injection advance on ignition lag is closely bound up with the factors of combustion-chamber form and air flow, and that a definite relationship can be established only for a particular design.

Effects of Inlet Pressure and Compression Ratio—From what has been said it is apparent that anything which increases the density and the pressure of the air in the combustion chamber during the period immediately following the



FIGS. 5 AND 6.—SHOWING THE VARIATION OF IGNITION LAG, BRAKE MEAN EFFECTIVE PRESSURE, AND MAXIMUM PRESSURE WITH INJECTION ADVANCE IN A TURBULENT (*Left*) AND A NON-TURBULENT (*Right*) ENGINE.

beginning of injection will shorten the ignition delay. The effect of the inlet pressure on the ignition lag was first clearly pointed out by Boerlage and Broeze of the Bataafsche Petroleum Co. Laboratory, Delft, Holland, who, in an article in *Engineering* of December 4, 1931, published the diagram reproduced in Fig. 7. These investigators took a series of "90-deg-offset" indicator diagrams with an ordinary piston-type oil-engine indicator, the air supply to the engine being throttled stepwise, so that successive diagrams showed less compression and less maximum pressure. The vertical line near the middle of the composite diagram marks the beginning of fuel injection. On each pressure line the ignition point was marked off, and a line referred to as the "delay line" was drawn through these points. The ignition lag, of course, is proportional to the horizontal distance between the ignition point and the start-of-injection line. It is clear from these diagrams that the more the engine is throttled, the greater becomes the ignition lag. In further confirmation of this relationship it may be mentioned that in a test on the single-cylinder, 5 x 7-in. experimental engine of the N.A.C.A., when the air supply was throttled so that the inlet pressure was only two-thirds of an atmosphere, the ignition lag was 18

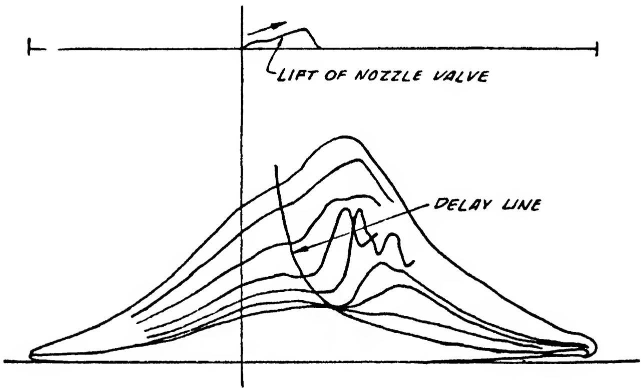


FIG. 7.—A SERIES OF OFFSET INDICATOR DIAGRAMS, TAKEN WITH ENGINE THROTTLED TO DIFFERENT DEGREES, SHOWING THE EFFECT OF INDUCTION PRESSURE ON IGNITION LAG.

crank-angle deg, whereas if the engine was supercharged to 1.3 atmospheres, the lag was only a trifle more than 10°.

The effect of the compression ratio on the ignition lag with two different grades of fuel is shown in Fig. 8, the tests having been made in the C.F.R. (Cooperative Fuel Research) single-cylinder test engine of 30 cu in. displacement at 600 rpm. It will be seen that with fuel oil the ignition lag decreased from 0.005 sec for a compression ratio of 11.5 to 0.004 sec for a compression ratio of 14.5. This difference of 0.001 sec between the ignition lags at 600 rpm corresponds to 3.6 deg of crank angle.

Physical and Chemical Lags—It has been found that the time which elapses between the beginning of injection and the appearance of a noticeable rise in pressure above the compression line is made up of two distinct items. The first, known as the physical lag, is made up of the time required to bring a minute portion of the fuel charge up to the vaporization temperature and vaporizing it, while the second, the chemical lag, is the time occupied by certain intermediate reactions and in sufficient combustion to produce a measurable pressure rise. Boerlage and Broeze have estimated that for fuels of normal

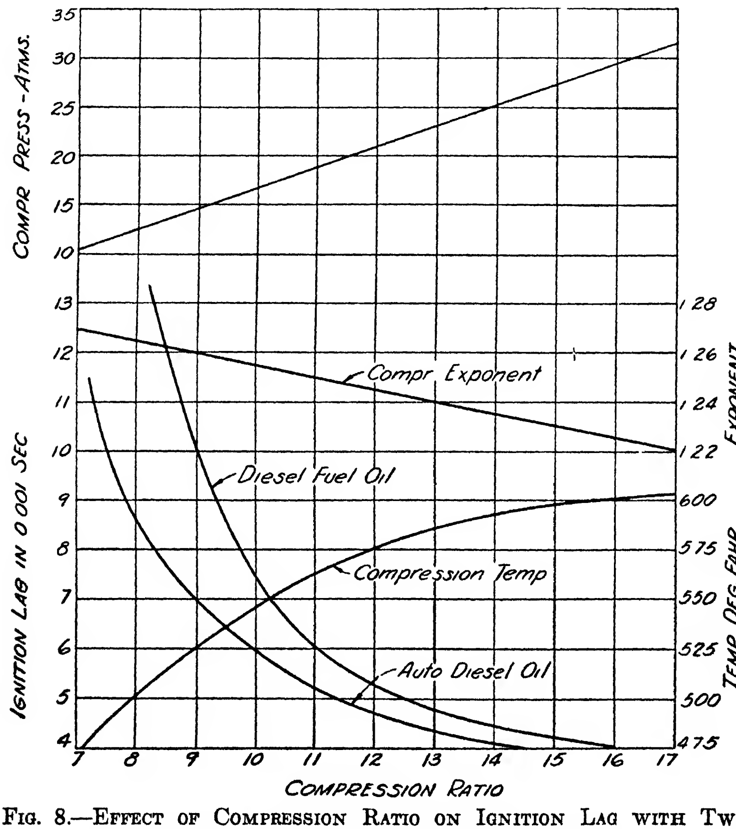


FIG. 8.—EFFECT OF COMPRESSION RATIO ON IGNITION LAG WITH TWO DIFFERENT FUELS IN THE C.F.R. TEST ENGINE.

volatility, the physical lag represents only 5 to 10 per cent of the total. It appears that the various methods of reducing the lag affect mainly the physical lag, and it has been shown by Schweitzer (*Automotive Industries*, June 25, 1938) that the ignition lag cannot be reduced below a certain minimum even by injecting into flame. The absolute minimum to which the total lag can be reduced is about one millisecond, which corresponds to 5.4 crankshaft degrees at the standard test speed of 900 rpm for the C.F.R. test engine.

Effect of Fuel Characteristics—Ignition lag is due primarily to the fact that before the fuel can ignite some of it must be raised to its ignition temperature by the air into which it is injected. The fuel, of course, is sprayed into the combustion chamber at substantially atmospheric temperature. Now, the ignition temperature varies greatly with the character of the fuel. Some of the early experimenters employed shale oil and tar oil, which have much higher ignition temperatures than the petroleum fuels ordinarily used in high-speed Diesel engines. The heat required to raise the fuel to the ignition temperature is derived from the air, and the rate of heat transfer from air to fuel is proportional to the difference between their temperatures. At the beginning of the process of heat transfer the temperature difference, of course, is the same, regardless of the character of the fuel used; but tar oils have to be carried to a temperature which is not very much below that of the air charge at the end of the compression period, and as the temperature of the fuel approaches that of the air, the rate of heat transfer decreases. Hence with these oils the ignition lag is naturally longer.

Preheating of Fuel—It has been proposed to cut down the ignition lag by preheating the fuel. But difficulties would be expected in connection with the handling of heated fuel in injection pumps and injectors. Besides, an experimental investigation by Gerrish and Ayer (N.A.C.A. Report No. 565) indicated that moderate heating will not produce the desired effect, and will even give negative results. The experiments were carried out on a pre-chamber type of Diesel engine with a common-rail injection system. In normal operation the temperature of the fuel at injection was 124 F. When it was raised to 300 F, the ignition lag increased from 9.5° to 10.5°, but when the temperature was raised further, to 750 F, the lag was reduced to 6°. The rate of pressure rise and the maximum pressure also were reduced, while the mep and the thermal efficiency were slightly increased. The engine is said to have operated more smoothly, the exhaust to have been clearer, and the carbon deposits to have been considerably less than with injection at normal temperature.

Pilot Injection—One method of reducing roughness or detonation in Diesel engines consists in injecting the fuel charge in two instalments as it were—a small pilot charge to initiate combustion, and the main portion of the charge while the pilot charge is burning. Dr. Jafar in an investigation at Birmingham University found the best proportion for the pilot charge to be about 22 per cent of the main charge, and the best timing for the pilot charge (at 1200 rpm of the engine), from 40 to 50 deg ahead of top center.

Dr. Schweitzer also made some experiments with pilot injection and obtained best results with a pilot charge not exceeding 15 per cent of the main charge, injection of the pilot charge starting 10° to 15° ahead of top center. If the same grade of fuel is used for both charges, the main charge must follow the pilot charge after 8° to 10°, but if fuel of high ignition quality is used for the pilot charge, the spacing should be less. Dr. Schweitzer also observed that when the lift diagram of the injector valve shows a small hump at the beginning of the lift (indicating that the valve is lifted first by the arrival of a pressure wave and closes again when the wave recedes), the combustion is always smooth. This hump can be produced by a proper combination of nozzle-opening pressure, injection-line length, and engine speed.

A system of pilot injection developed by the Atlas Diesel Company of Stockholm, Sweden, in 1949 was adopted by the London General Transport Board for use on its 5000 Diesel-powered buses in operation in the British capital. This system will be described in the chapter on Fuel Injection and Sprays.

Measurement of Ignition Lag—The original method of measuring the ignition lag consisted in taking a pressure-time diagram from the test engine with and without fuel injection, and noting the point on the pressure line where it begins to deviate from the compression line. The point of injection begin is automatically recorded on the diagram, and by projecting these two points on the axis of abscissas, on which there is a crank-angle scale, the lag can be read off directly.

At present the conventional method of measuring ignition lag consists in mounting two neon bulbs on the rim of the flywheel and closing the circuits through these bulbs by means of contactors which are operated, respectively, by the injector valve and a diaphragm under the influence of the cylinder pressure. The latter, known as the combustion indicator or the combustion pick-up, is so designed that the contact will not be closed by any pressure due to compression, but only by the higher pressure, or the greater rate of pressure rise, due to combustion. The two neon bulbs are located some distance apart on the rim of the flywheel, behind narrow slits in a protractor which passes underneath a sight tube, and if the angular spacing of the two slits equals the ignition lag in crankshaft degrees, both lamps flash while passing the sight tube, and appear to the observer to give simultaneous flashes. Repeated adjustment of the protractor until simultaneous flashes are obtained is a rather tedious process, for which reason fuels are now tested by the constant-lag method. The two bulbs are set 13° apart on the flywheel, and with the engine in operation, the compression ratio is varied until both flashes appear in the sight tube.

Holfelder, a German research man, holds that ignition lag can be measured much more accurately by photographically recording the appearance of flame in the combustion chamber, than by recording the rise in either the pressure or temperature. This method, however, calls for the use of a test engine with a quartz window in the combustion-chamber wall and of very expensive high-speed photographic apparatus, and is therefore limited to a few specially-equipped laboratories.

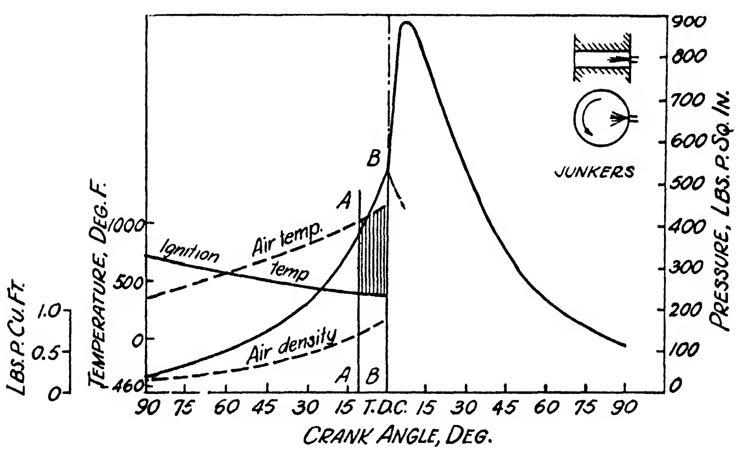


FIG. 9.—CYLINDER DIAGRAM OF JUNKERS ENGINE.

Analysis of Indicator Diagram—A study of Diesel combustion characteristics as revealed by indicator cards was made by Professor Neumann of Hannover Technical College during the early years of the high-speed engine. Pressure-time cards were taken under full-load conditions from engines of three different types, one with direct injection into a "quiescent" combustion chamber, another with direct injection into a "turbulent" combustion chamber, and the third with injection into a precombustion chamber (in which latter the combustion is initiated and from which it spreads to the main combustion chamber). The method of analysis employed is illustrated by Fig. 9, which shows a diagram taken from a Junkers two-stroke, double-piston engine, with direct injection into the flat cylindrical combustion chamber in which air turbulence is induced by tangential inlet ports. Lines are drawn in on the diagram to show the variation of the pressure, temperature and density of the air during the compression stroke, these values having been calculated on the assumptions that at the beginning of compression the air in the cylinder was at atmospheric pressure and a temperature of 176 F, and that the exponent of polytropic compression was 1.35. Another line on the diagram shows the decrease in the ignition temperature of the fuel with increase in air density, based on experimental results of Tauss and Schulte (discussed in the chapter on Fuels for Diesel Engines).

The vertical line *A-A* marks the point of the cycle at which delivery by the pump began, which Professor Neumann erroneously took to be identical with the beginning of injection into the combustion chamber. It has since been found that, owing to the compressibility of the fuel and the elasticity of the tube containing it, there is a certain injection lag, and the ignition lag therefore is shorter than indicated. Actually the distance *A-B* represents the sum of the injection lag and the ignition lag.

The engine with direct injection into a "turbulent" combustion chamber showed the highest combustion pressure, about 880 psi, as compared with 670 psi for the engine with direct injection into the "quiescent" combustion chamber, but the difference is largely accounted for by the compression pressures, which were 565 and 400 psi respectively. In the engine with precombustion chamber, the pressure in the main combustion chamber rose only slightly above the compression pressure with the onset of combustion, from 595 to 625 psi. This moderate combustion pressure is a characteristic of the precombustion-chamber type of engine.

Combustion-Efficiency Curves—C. B. Dicksee has pointed out that in the experimental development of an engine it is an excellent plan to plot the indicated mean effective pressures as ordinates against the fraction of the air charge consumed as abscissas. Starting with no load, the imep at first increases directly with the amount of air used, or nearly so, but sooner or later the characteristic deviates from the straight line, and in some cases it will turn rapidly and assume a more or less horizontal direction. The point at which a serious departure from a straight line first occurs depends largely on the effectiveness of the means provided to enable the fuel particles to quickly find the oxygen necessary for their combustion. The load at which the exhaust becomes smoky is also of interest in this connection. Fig. 10 shows the imep which would be developed for any given quantity of available air consumed with an absolute volumetric efficiency of 85 per cent and various values of the thermal efficiency.

A pronounced early deviation from the straight line without smoke in the exhaust indicates that the air is not being used until late in the cycle, either on account of lack of tur-

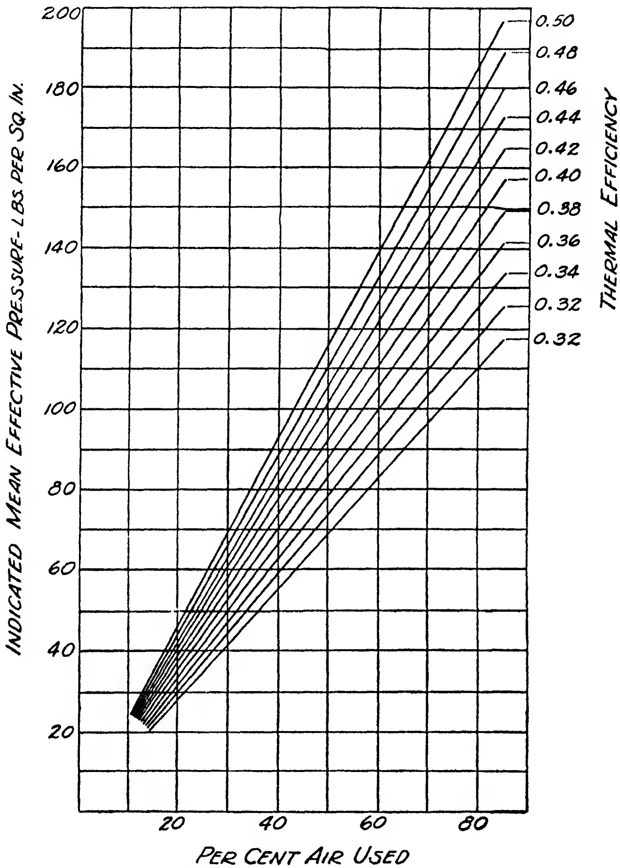


Fig. 10.—Relation between Proportion of Air Used and IMEP for 85 Per Cent Volumetric Efficiency and Various Thermal Efficiencies.

bulence, too slow a rate of, or too late injection. The presence of thick smoke when only a moderate proportion of the air has been consumed indicates that the fuel did not find its oxygen, that some parts of the combustion chamber are receiving an excess, while others are not receiving enough fuel, and that an increase in turbulence or a change in nozzle design is called for.

In obtaining the data for the plots here discussed, the air consumed is determined from the results of exhaust-gas analysis by means of the well-known Orsat apparatus. If the air charge of the cylinder and the heat value of the fuel are assumed to be fixed (a volumetric efficiency of 85 per cent can be figured with), then the slope of the characteristic line denotes the thermal efficiency. For instance, with a fuel having a heat value of 19,300 Btu per lb, an indicated mep of 110 psi, with 80 per cent of air consumed, represents a thermal efficiency of 30 per cent, while 160 psi with the same air consumption represents slightly more than 43 per cent.

Spread of Combustion—Although ignition undoubtedly occurs first at a particular point of the combustion chamber, it must not be supposed that there is only one point of ignition from which (as in the spark-ignition engine) the flame spreads gradually throughout the chamber. The sprays normally are symmetrical, and there is every reason to believe that at, say, opposite sides of the jet at a given distance from the nozzle, the conditions for spontaneous ignition at any given time are substantially alike. Even if they are not absolutely alike, after ignition has taken place at one side of the spray, the opposite side will be brought to the condition necessary for spontaneous ignition by heat transfer from air to fuel much quicker than it can be ignited by direct flame propagation from the first point of ignition. Thus, instead of a single ignition point, as in the carburetor-type engine, we have really a considerable area over which—and perhaps even a considerable volume throughout which—ignition occurs practically simultaneously, and through the same agency of heat transfer from air to fuel.

Combustion phenomena in the case of the Diesel engine are complicated by the fact that two processes are going on in the combustion chamber simultaneously—the distribution of fuel throughout the air charge and the heating of this fuel to the ignition point. Photographs of fuel sprays caught in a dispersion liquid show that when the injection pressures are comparatively low, there are great differences in the sizes of the droplets. Although the photos of high-pressure sprays do not show this so plainly, probably because the finest droplets cannot be clearly distinguished in the pictures, there is reason to believe that with such sprays there are size variations of the same relative order. Undoubtedly the surfaces of the smaller droplets reach the ignition temperature first, because in their case there is less liquid volume behind unit surface to carry off the heat absorbed at the surface, and the smallest droplets therefore burn first.

Vaporization and Combustion—There has been considerable discussion in the past as to whether it is necessary to vaporize liquid fuel before it can be burned, or whether combustion can take place directly from the liquid state. Most of the older writers on the subject held the view that the fuel first must be atomized by air friction, then heated and vaporized, and finally mixed with air before it can be burned. Some of the later investigators, however, reached results which convinced them that combustion can take place directly from the liquid droplets. With the application of high-speed photography to spray and combustion phenomena by the National Advisory Committee for Aeronautics, definite evidence on this point has been produced. Results obtained by Rothrock (N.A.C.A. Report No. 435) show that even at temperatures below the self-ignition point of the fuel the entire spray disappears (becomes invisible) in from 0.001-0.002 sec, and it is now generally believed that the fuel injected first is practically completely vaporized when ignition takes place.

Time Required for Combustion—Whereas in a carburetor engine the time required to complete the combustion depends upon the average speed at which the flame travels from the spark points to the remotest part of the combustion chamber, in a Diesel engine it depends on the speed at which the oil molecules can be brought into contact with molecules of oxygen. Combustion of the droplets starts at their surface. Unfortunately, as soon as its combustion has begun, the droplet is surrounded with an atmosphere of gases of combustion, which tend to interfere with the continuance of the combustion process. However, throughout the combustion period there is relative motion between the fuel droplets and the air, which tends to sweep away the gases of combustion from the surface of the droplets as soon as they are formed, and to bring fresh air in contact with the surface. This effect increases with the relative velocity of fuel and air.

Effect of Engine Speed on Combustion—A method of plotting combustion phenomena in a Diesel engine against crankshaft speed was given in an S.A.E. paper by F. G. Shoemaker and H. M. Gadebusch of General Motors Detroit Diesel Engine Division. Fig. 11 is reproduced from this paper. Time in milliseconds is plotted along the horizontal axis, and the light curved lines to both sides of the "dead-center" line represent constant crank angles before and after dead center. In the engine to which this diagram applies fuel injection began at 20 deg ahead of dead center. The various heavy curved dotted, dashed, dash-dotted, etc., lines to the left of the dead-center line represent points of ignition with different fuels at different engine speeds. The quality of each fuel is represented by the (cetane) number at the lower end of each curve. Ignition lag with any of these fuels at any engine speed is measured by the horizontal distance between the 20-deg curve (the lower end of which has been cut off) and the heavy line corresponding to the particular fuel. It will be seen that the poorest fuel (22) gives the longest ignition lag.

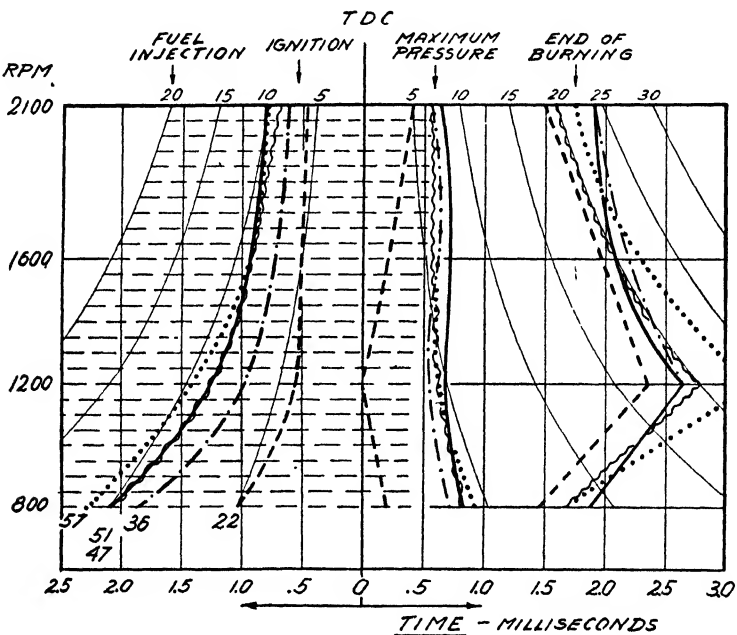


FIG. 11.—EFFECT OF CRANKSHAFT SPEED ON COMBUSTION PHENOMENA.

To the right of the dead-center line there are two sets of curves corresponding to the same five fuels, those nearest it representing the time of maximum pressure in the cylinder, the other set the end of combustion. It will be seen that with fuel 22, the one of lowest quality, the cylinder pressure reaches its maximum value at dead center at a speed of 1200 rpm, and only shortly after dead center at other speeds. The horizontal distance between the ignition and maximum-pressure curves for any fuel is an inverse measure of the rate of pressure rise, and with fuel 22 the pressure-rise curve is very steep. With this same fuel combustion in the cylinder also comes to an end earlier than with any other. The data from which the curves were drawn were obtained from pressure-time diagrams taken from the engine. To determine the end of combustion the pressure-time diagrams were converted into logarithmic pressure-volume diagrams in which the combustion line joins the compression and expansion lines at fairly pronounced angles.

Exhaust Components—If the fuel consisted entirely of pure hydrocarbons and burned completely, the exhaust gases would consist of nitrogen, carbon dioxide, water vapor and (excess) oxygen. However, all ordinary Diesel fuels contain small quantities of impurities, chiefly sulphur and nitrogen compounds, and, besides, combustion of the fuel is never complete. The sulphur and nitrogen compounds are oxidized during combustion, and sulphur dioxide, sulphur trioxide, and oxides of nitrogen are found in the exhaust in small quantities. Owing to incomplete combustion of the fuel, the exhaust also may contain hydrocarbons (especially methane), hydrogen, free carbon, carbon monoxide, and aldehydes. All components of the exhaust with the exception of oxygen and nitrogen may be said to vitiate the atmosphere, but carbon monoxide, aldehydes, and the oxides of sulphur and nitrogen are the most objectionable.

Carbon monoxide is a colorless, odorless and highly poisonous gas, of which not more than one part by volume in 10,000 can be tolerated in the air we breathe. Fortunately, the carbon-monoxide content of the Diesel exhaust is normally quite low, far lower than that of the average exhaust of gasoline engines. Within the range of air/fuel ratios with which a Diesel engine can be operated economically and without objectionable smoke, the CO content is usually below 0.1 per cent. Aldehydes are compounds of carbon, hydrogen and oxygen intermediate between alcohols and organic acids. Alcohol combines with the oxygen of the air under proper conditions to form an aldehyde and water, and further addition of oxygen converts the aldehyde—which is very unstable—into an acid. The aldehydes are chiefly responsible for the characteristic odor of the Diesel exhaust. They seem to be formed particularly when the temperature in the cylinder is relatively low, as at idling, when the aldehyde content of the exhaust may be as high as one volume in 2000, while in normal operation it is very much less. Tests conducted by the Bureau of Mines have shown that odor becomes perceptible at an aldehyde concentration of 0.2 to 0.3 parts per million, and that nasal and eye irritation by the exhaust begin at a concentration of about one part per million.

Oxides of sulphur and nitrogen are undesirable in the exhaust also because in the presence of water they form corrosive compounds, which property they share with carbon dioxide.

Problems connected with the harmful effects of the exhaust on health are of special importance in connection with the use of Diesel engines in tunnels and mines, and the U. S. Bureau of Mines has investigated these effects and the engine, fuel, and operating variables that influence the formation of harmful compounds. This study was dealt with in a paper entitled "Composition of Diesel Exhaust Gas," by Martin A. Elliott and Rogers F. Davis, which appeared in S.A.E. Quarterly Transactions for July, 1950.

Smoke in Diesel Exhaust—When a Diesel engine is being fed a fuel charge greater than it can burn properly, there will be unburned carbon in the exhaust, which manifests itself in a dense black smoke. As the injection quantity is decreased, the smoke becomes less dense, and at a certain point it vanishes practically entirely. At still lower loads there may be a whitish or grayish smoke in the exhaust, which is due to the presence of vaporized, unburned fuel. Under otherwise similar operating conditions, there generally is less smoke in the exhaust if the engine is being operated on fuel of high cetane number.

A comprehensive experimental investigation of the causes of Diesel smoke was made by Dr. P. H. Schweitzer of The Pennsylvania State College.* While hot smoke or sooty smoke always is due to incomplete combustion, the latter in turn may be due to a variety of primary causes, and one of the most interesting of Dr. Schweitzer's findings was that in order to eliminate or minimize smoke, a number of design and operating factors must be held within limits. If either the upper or the lower limit is passed, smoke is produced or increased. Thus the exhaust will be smoky if injection occurs either too early or too late, if the fuel used has either too low or too high a cetane number, if the compression is either too low or too high, and if the engine is either too lightly or too heavily loaded.

Incomplete combustion frequently is the result of too late ignition, and too late ignition may be due to too late injection, too low compression, or the use of a fuel of too low cetane

* Must Diesel Engines Smoke?, S.A.E. Quarterly Transactions for July, 1947, page 476.

number. The reason why too high a cetane number also is productive of smoke is not so obvious, but it is a fact that fuels of over 60 cetane usually produce a smoky exhaust, and the same effect is obtained with ordinary fuels if the compression is too high. In both cases the ignition lag is unusually short, and early ignition seems to interfere with the dispersion or distribution of the fuel in the air charge. Too heavy a load means that so much fuel is being injected that not all of it can readily find the oxygen needed for its complete combustion. At very light load, as the point of misfiring is approached, both cold and hot smoke are produced, the first because some of the fuel will not be ignited and appears in the exhaust as a fog, and the second because some of the remainder will be ignited so late that it will burn incompletely. According to Dr. Schweitzer, in a normal engine there should be relatively little smoke in the exhaust when operating between 25 and 75 per cent of full load, and the smoke should rise rapidly between 75 per cent and full load. The smoke, of course, is not wanted for its own sake, but because without it the "lugging" power of the engine most likely would be inadequate.

Fuel-fog or cold smoke (failure of all or part of the fuel to ignite) is due to such causes as too light load, too early injection, too low compression, too low volatility of the fuel (as represented by the 50 per cent point), and impingement of the fuel spray on cooled combustion chamber walls. The latter factor is or may be related to some of the others. Both types of smoke often are produced simultaneously.

Smokemeter—The quality of combustion in a Diesel engine is indicated by the color of its exhaust. When the exhaust is clear, the combustion is practically complete, while a smoky exhaust indicates the presence of unburned carbon particles, and therefore incomplete combustion. Ordinarily the exhaust is observed directly with the unaided eye, but such observations lack accuracy, and it is impossible to give any figures for the relative density of the smoke. At present, therefore, observations are often made with a smokemeter.

A simple smokemeter developed in the Diesel Laboratory of The Pennsylvania State College and known as the CRC Photovolt smokemeter, is shown in Fig. 12. Light from the 6-volt lamp enters the 2 x 18-in. glass tube at one end and strikes the light-sensitive layer of the photocell at the other. Current from the cell passes through a microammeter furnished with the instrument. Before a smoke-density determination is made the tube is scavenged with air by the suction fan shown. With clean air in the tube the instrument is adjusted so that the pointer of the microammeter is deflected the full range of the scale, which latter has 100 divisions. When the tube is then filled with exhaust gas, the pointer moves to the left and shows the density of the smoke directly in per cent.

Exhaust Odor—Diesel exhausts have a characteristic odor which, however, changes with the load on the engine. The odor is most objectionable when the engine is idling or operating under light load. Wetmiller and Endsley, of the Texas Company, who made a study of the conditions affecting smoke and odor in Diesel exhausts, describe the odor when operating with rich mixtures (full load) as "strong and heavy but not particularly objectionable," while the odor when operating with lean mixtures (idling or light load) is characterized by them as "very pungent, sharp, acrid and objectionable."

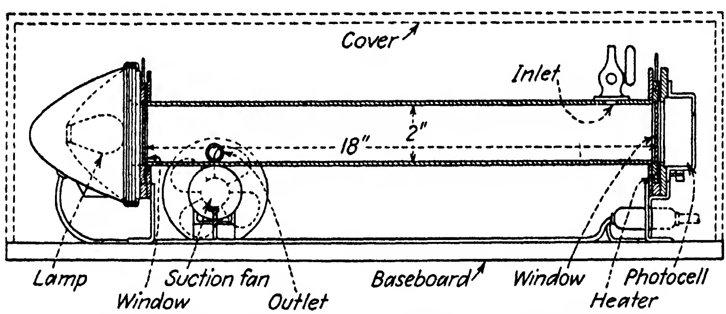
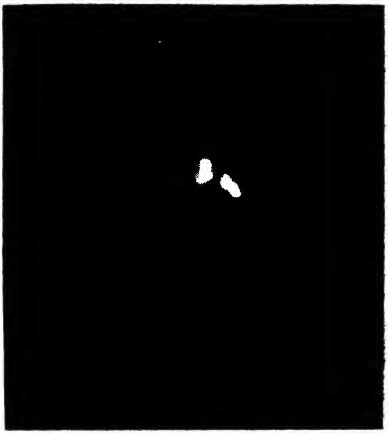


Fig. 12.—Sectional View of Photovolt Smokemeter.

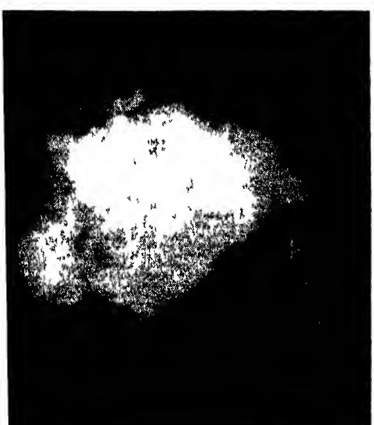
Photographic Study of Combustion in Engine—The National Advisory Committee for Aeronautics has succeeded in taking photographs of the spray and flame in an actual engine, by means of a special high-speed photographic apparatus capable of taking 1850 pictures per second, which at the speed of 570 rpm of the engine corresponds to intervals of less than two degrees of crankshaft rotation. The actual time of exposure was about one-third of the interval. Some of the pictures obtained are reproduced herewith (Figs. 13 to 20).

In the tests, the fuel was injected from the top of the explosion chamber, which is at the right side of the flame pictures shown. In most cases a six-orifice nozzle was used, and one of the series of pictures shown here are of flames from this nozzle, while the other series are of flames from a single-orifice nozzle. During each cycle, 0.00025 lb of Diesel oil was injected into the 5 x 7-in. cylinder, which made the air factor 1.3. A compression ratio of 13.6 was used.



1st frame
Start of flame

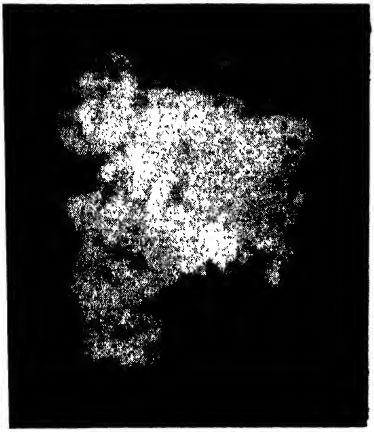
2nd frame
Arrow points to fuel spray



3rd frame
Fuel spray still visible

10th frame
Flame starting to decrease

Figs. 13 to 16.—Flame Spread with Injection from Single-Orifice Nozzle. Injection Starts at Top Center. 570 RPM.

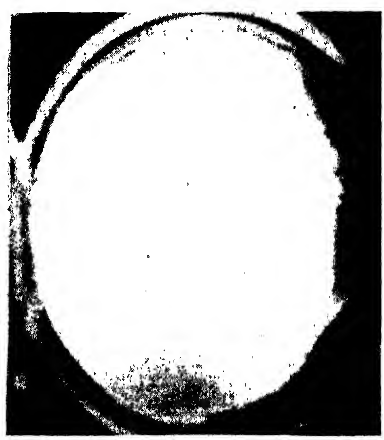


15th frame
Flame separating into small areas

25th frame
Flame marked by numerous small areas

Figs. 17 and 18.—Flame-Spread from Single-Orifice Nozzle. Injection Starts at Top Center. 570 RPM.

3rd frame
Silhouette of spray



8th frame
Maximum spread of flame

Figs. 19 and 20.—Flame Spread with Multi-Orifice Nozzle. Injection Starts at Top Center. 570 RPM.

With injection starting at top center, ignition occurred 5 or 6° after top center, at several different places at once. At the next exposure, 2° of crankshaft motion farther on, the isolated areas of burning had become connected and the flame had spread through about two-thirds of the combustion chamber. The flame filled the chamber until more than 20° after top center, became smaller steadily after 25°, and lasted about 70° of crank motion more, a few tiny isolated spots being still noticeable 90° after top center.

With injection starting 20° before top center, the pictures showed that the flame first appeared in the cylinder about 12° before top center. The ignition lag therefore was slightly greater than when injection started at top center, but flame spread was more rapid. Again the flame started in more than one place. Only two tiny isolated flames appeared in the picture taken 12° before top center. At top center the flame began to die out, but it continued to be visible in the combustion chamber until 60° later.

When the injection-advance angle was increased to 40°, the ignition lag was greater than with either of the two timings previously used. The first sign of combustion was two small flames that appeared 15° before top center. Two degrees of crankshaft motion later the flames had spread to the edge of the chamber in almost all directions. At top center the flame was dying out rapidly, and 10° later it had died out completely. Flame continued in the top of the chamber for some time after it had disappeared in the bottom, which indicated that the fuel injected did not form a homogeneous mixture with the air even when injected 40° ahead of top center and 25° before ignition started.

CHAPTER IV

Fuels for Diesel Engines

Three general types of liquid fuels have been used in Diesel engines in the past, either in regular service or experimentally. By far the most common are products of crude petroleum, and in the United States these are used exclusively in Diesel engines of all types. In Europe, tar oils or creosote oils, produced from the tars of lignite and bituminous coals, are being burned in large Diesels. Finally, in France and Belgium, both of which countries possess extensive African colonies, there has been a good deal of talk of the use of vegetable oils in compression-ignition engines, and some experiments along this line have been carried out. Palm, peanut, cottonseed and poppy-seed oil have been tried, but none is as satisfactory as gas oil. Their viscosities are rather high, which makes it difficult to pass them through the filtering equipment of engines at low temperatures; they usually result in a rapid building up of deposits in the combustion chambers, and most of them give an exhaust of very objectionable odor.

Petroleum fuels may be divided into four classes, viz., the crude petroleum as it comes from the ground, "straight-run" fractional distillates of this petroleum, fractional distillates of petroleum that has been subjected to treatment in thermal or catalytic cracking stills, and the residual oils which remain in the stills after the more volatile fractions have been driven off. Diesel engines sometimes are referred to as crude-oil engines, and some of them, no doubt, can be run more or less satisfactorily on crude oils that are fairly free from asphaltic residues, if the oils are carefully centrifuged or filtered. It must be remembered, however, that the high-speed Diesel, owing to the great rapidity with which combustion must take place in it, and the fineness of metering passages of its auxiliary equipment, cannot make use of all of the fuels that may be satisfactory for the larger engines. The automotive Diesel, moreover, must approach the standards of ease of starting and smoothness of the automotive gasoline engine, and these requirements place further limits on the range of fuels that can be satisfactorily used.

Diesel Fuel Oils—In the United States high-speed Diesel engines are operated for the most part on gas oil or fuel oil. Like gasoline, this fuel does not have a definite chemical composition. Its composition and properties are subject to changes from time to time in accordance with market conditions in the field of petroleum products. When high-speed Diesel engines first came into practical use they were operated mostly on distillates of paraffin-base (Pennsylvania) crude oil having a relatively narrow boiling range slightly above that of kerosene. This fuel ensured easy starts and smooth operation, and was generally recognized as the best for this particular purpose. But as the use of high-speed Diesels increased it became difficult to produce enough fuel of this type. The prices of petroleum products depend to a large extent on the relative demand for them. Up to now the demand for gasoline has been greatest, and that fraction has had to bear most of the cost of crude oil production and refining, which accounts for the lower price of gas oil. Refiners naturally strive to produce the largest possible quantity of gasoline from a given amount of crude, because of the very large market for this fraction and its relative independence of price within reasonable limits.

As long as gasoline was produced only by straight distillation, the maximum yield of even the best crudes was little more than 10 per cent. It was then found that by subjecting the crude petroleum to heat and pressure (the so-called thermal cracking process) some of its heavier and less volatile constituents could be converted into the more volatile compounds of which gasoline is composed, and the yield in gasoline thus greatly increased. Now that fraction of the paraffin-base crude oil which constituted the original gas oil makes excellent charging stock for cracking stills, and the high-speed Diesel engine therefore has entered into competition with the gasoline engine for this petroleum fraction. Since the chief reason for the existence of the Diesel engine is the saving on fuel cost which it makes possible, this is a rather serious matter. It may be pointed out in this connection that the same general type of petroleum product which is suitable for Diesel engines also is burned in domestic oil heaters, the number of which is growing rapidly.

The Diesel-fuel situation became more critical in the course of World War II, when the need for large quantities of high-octane aviation gasoline arose. Many petroleum companies then established refineries in which the petroleum is subjected to a catalytic cracking process. This process transforms compounds of low ignition temperature into others of higher ignition temperature, thereby increasing the anti-knock value of the fuels in carburetor engines. The process, however, transforms not only the low-boiling constituents which on distillation are included in the gasoline fraction, but also the higher-boiling ones which are included in the gas-oil fraction. Thus gas oils derived from the cracking still have relatively high ignition temperatures, which makes them rather poor Diesel fuels. The large increase in the number of high-speed Diesels during World War II, together with the fact that many of them had to be operated in cold climates and sometimes at high altitudes, far from the sources of the superior paraffin-base gas oils, brought the whole problem of fuels for high-speed Diesel engines prominently to the front once more. Automotive Diesel fuels have been under investigation by a division of the Cooperative Fuel Research (C.F.R.) Committee for some years; a number of reports have been issued by this division, and these have thrown considerable light on the subject.

Taxation of Fuels—When the Diesel engine was first applied in road transportation, its fuel was tax-free in most countries, the laws governing the taxation of motor fuels usually being so worded as not to cover oils of low volatility. In most European countries the tax on gasoline was quite high, of the same order of magnitude as the price paid for the fuel in this country. Freedom from taxation of its fuel encouraged the rapid development of the automotive-type of Diesel engine, for its use made possible savings on fuel costs which in some instances ranged as high as 80 per cent. Obviously, this situation could not endure, and as soon as Diesel trucks appeared on the highways in considerable numbers, laws were passed to force their owners to contribute their proper share to the cost of road maintenance. In Great Britain the tax was applied stepwise, being first placed at one penny per Imperial gallon, and some years later raised to 8 pennies, which put Diesel fuel on a par with gasoline from the standpoint of tax-liability.

At the time of this writing, most of the states of the Union levy the same tax on Diesel fuels as on gasoline, the rate usually ranging between 3 and 7 cents per gallon. Owing to the higher efficiency of the Diesel engine, the return to the state for a certain volume of traffic is less from Diesel than from gasoline trucks, and some of the states have sought to make up for this loss by imposing a higher registration fee on Diesel trucks. Diesel fuel used for other purposes than road transportation (in farm tractors, for instance) is tax-free in most states. In addition to the State tax there is a small Federal tax on all fuels used in road transportation.

Ignition Temperature—One of the most important characteristics of fuels for high-speed Diesel engines is the temperature of spontaneous ignition. This is so not only because under certain operating conditions it is difficult to develop a sufficiently high temperature by the compression of the air charge to effect ignition, but also because the ignition temperature of the fuel has a predominant effect on the smoothness of operation of the engine.

It was observed quite early that Diesel engines ran much more smoothly on some fuels than on others, and for a long time there was much speculation as to just what physical or chemical properties of the fuels this difference might be due to. An explanation became easy once the succession of events during combustion in a Diesel engine had been clearly established. Ignition does not occur instantaneously at the beginning of fuel injection, but only after a certain small amount of the fuel injected has been raised from the temperature at which it is being handled by the injection equipment to the temperature of self-ignition of the particular fuel under the density conditions obtaining in the engine.

When ignition takes place, a considerable amount of fuel is already in the combustion chamber, and this fuel burns at an explosive rate. The time interval (ignition lag) between the beginning of injection and the moment of ignition increases with the ignition temperature of the fuel. The rate of pressure rise during the period immediately after ignition depends upon the amount of fuel in the combustion chamber when ignition takes place, and hence upon the ignition lag. It follows that the smoothness of combustion is dependent primarily on the ignition quality of the fuel. The term "ignition quality" as here used has reference to the general behavior of the fuel when burned in an engine, but it is dependent mainly on the temperature at which the fuel ignites spontaneously with a time lag corresponding to from 10° to 15° of crank rotation.

Ignition Temperature and Charge Density—It was formerly believed that each fuel had a definite ignition temperature regardless of the pressure or density of the air into which it was injected. Tables of ignition temperatures of various fuels were published many years ago, but while these serve to range the fuels in the proper order with respect to ease of ignition, they give no idea of the exact temperatures to which the various fuels must be raised in the combustion chamber of the engine in order to effect their ignition. F. A.

Foord has pointed out that these are the lowest temperatures at which the particular fuels will ignite after an indefinite delay, and that the method by which they are obtained in reality gives only one point of a curve relating ignition lag to temperature for the particular fuel. This lowest ignition temperature may not bear any important relation to the performance of the fuel in a Diesel engine.

That compression of the air in the engine cylinder has the effect of lowering the ignition temperature of fuel was confirmed by an investigation made by Tauss and Schulte in 1925. These experimenters assumed that it was the pressure that lowered the ignition temperature, and they plotted the observed ignition temperatures against the air pressures. It is now generally agreed, however, that the factor which has the most direct effect on the ignition temperature is the density of the air.

In Fig. 1 the author has replotted some of the results of Tauss and Schulte on a density basis. The compression-temperature curve in the chart is calculated on the assumption that the temperature at the beginning of the compression stroke is 180 F, that the cylinder is filled to atmospheric pressure, and that the exponent n has a value of 1.32. The points of intersection of the ignition-temperature curves with the compression-temperature curve are the temperatures at which the fuels actually ignite in the combustion chamber of the engine. It will be seen that these ignition temperatures are substantially as follows for the different fuels: Gas oil, 540 F; gasoline, 625 F; coal-tar oil, 825 F; benzol, 900 F. If fuels of different spontaneous ignition temperatures are blended, the ignition temperature of the blend is always considerably closer to that of the one with lowest ignition temperature than might be expected from the proportion of the blend. This suggests that Diesel fuels may be materially improved with respect to ignition quality by the addition of small quantities of "ignition oil." In Germany such "ignition oils" are much used with coal-tar oils in larger, low-speed engines, but they are usually injected separately.

Cetane-Number System—The ignition quality of a Diesel fuel is expressed in cetane numbers. The cetane number is the percentage by volume of cetane in a mixture with alpha-methylnaphthalene which matches the fuel in ignition quality in a standardized test. This system of expressing ignition quality is due to G. D. Boerlage and J. J. Broeze of the Bataafsche Petroleum Company's laboratory in Delft, Holland. They used cetene ($C_{16}H_{32}$) and alpha-methylnaphthalene ($C_{11}H_{10}$) as reference fuels, but cetene, which was derived from fish oil, proved rather unstable, which made it impossible to obtain strictly reproducible results, and for this reason the American petroleum industry a few years later substituted cetane ($C_{16}H_{34}$) for cetene. These two hydrocarbons, although belonging to different series, have nearly equal ignition qualities, the cetane number of pure cetene being 96.

Cetane is a petroleum derivative and belongs to the paraffin series of hydrocarbons. It has a specific gravity of 0.775, a boiling range of 544 F-553 F, and a freezing point of 61 F. Alpha-methylnaphthalene is a coal-tar product and a hydrocarbon of the naphthalene series (C_nH_{2n-12}); it has a specific gravity of 1.025, a boiling point of 470 F, and a freezing point of −7 F. Since both cetane and alpha-methylnaphthalene are

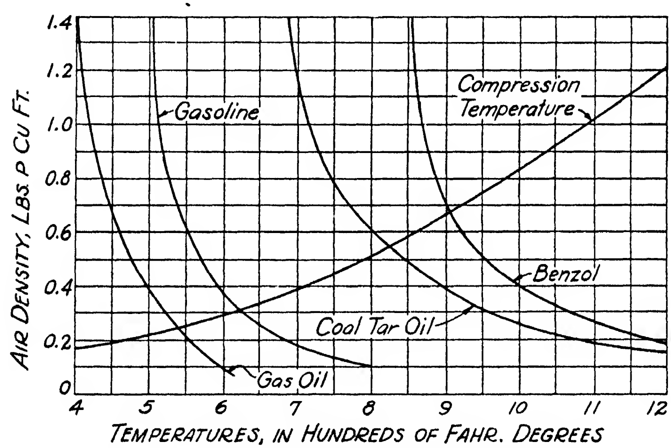


Fig. 1.—Variation of Ignition Temperature with Air Density.

quite expensive, secondary reference fuels are commonly used in routine testing. Shell Oil Company, Inc., of Wood River, Ill., furnishes Diesel secondary reference fuels which are straight-run petroleum distillates. The high reference fuel has a cetane number of 72.5, the low of 22. Commercial Diesel fuels usually range between 30 and 60 cetane number, and the cetane numbers of these secondary reference fuels therefore are far enough apart to meet the usual requirements.

Fuel-Test Engine—Tests for ignition quality are made in the C.F.R. (Cooperative Fuel Research) Diesel fuel-testing engine, which is a modification of the C.F.R. engine for testing carburetor fuels. The test for ignition quality of Diesel fuels has been standardized by the American Society for Testing Materials. The engine used is a single-cylinder, vertical one of 3.25-in. bore and 4.50-in. stroke, and Waukesha Motor Company of Waukesha, Wis., has been licensed to manufacture it. It has provisions for varying the compression ratio while it is running. Fig. 2 is a section through the upper part of the cylinder and the cylinder head, and shows that the piston comes very close to the cylinder head at the end of the up-stroke. The combustion chamber *H*, which is of horizontal cylindrical form, is in the cylinder head and communicates with the cylinder through a tangential passage, which latter ensures a high degree of turbulence in the chamber beginning toward the end of the compression stroke. Secured eccentrically into the end of the combustion chamber is the injector

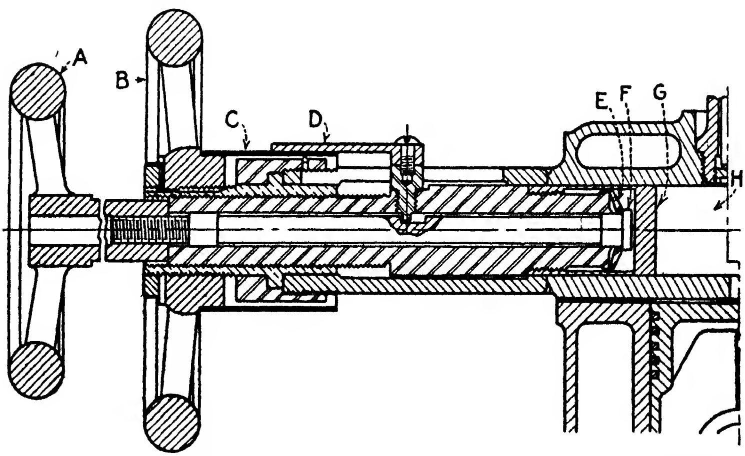


FIG. 2A.—SECTION THROUGH ONE-HALF OF CFR-ENGINE COMBUSTION CHAMBER.

A, locking wheel; B, compression ratio-adjusting wheel; C, micrometer-scale sleeve; D, micrometer-scale arm; E, locking washer; F, locking head; G, expansion-plug head; H, combustion chamber.

N, while into the top of the chamber is fitted a "combustion indicator" *J*—a pick-up device which serves to register the incidence of ignition.

Fig. 2 clearly shows the means provided for varying the volume of the combustion chamber, that is, for varying the compression ratio. They comprise a plug which can be moved endwise into and out of the cylindrical chamber by means of a screw equipped with handwheel *B*, and a locking wheel *A* and expanding washer *E* for locking the mechanism when the adjustment has been made.

Test Procedure—In making a fuel test, the engine is run at 900 rpm, the jacket water is kept constant within one degree at any point between 209 F and 215 F, and the crankcase oil, after being heated to 150 F by an electric heater, is allowed to attain the natural temperature level due to steady operation. As shown in Fig. 3, two neon bulbs are mounted on the rim of the flywheel, behind an aperture plate with narrow slits 13° apart. One of the slits is centrally under a sight tube when the crankshaft is in top dead center, the other 13° forward of that position in the direction of crankshaft rotation. Current is carried to these lamps by means of slip rings and brushes bearing thereon, and returns to the source through the engine block. The circuit through the first bulb is closed by means of the ignition indicator (Fig. 4), which is screwed into the wall of the combustion chamber. When ignition takes place, the sudden rise in pressure, acting on a

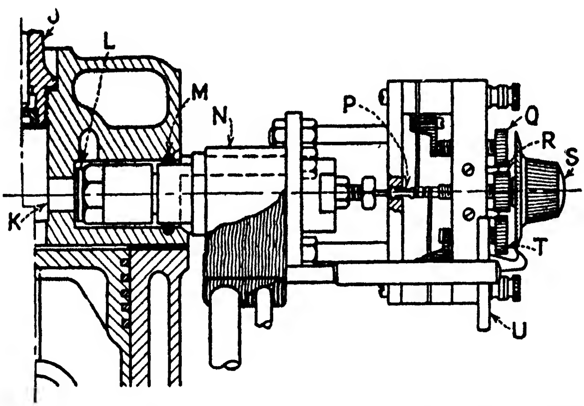


FIG. 2B.—SECTION THROUGH ONE-HALF OF CFR-ENGINE COMBUSTION CHAMBER.

J, combustion indicator; K, injector tip; L, injector gasket; M, rubber ring; N, injector; P, injector pin; Q, lower-leaf adjusting screw; R, center adjusting screw; S, bumper-spring adjusting screw; T, upper-leaf adjusting screw; U, by-pass valve handle.

steel diaphragm *H*, closes a pair of contacts. In a similar manner, the circuit through the second bulb is closed by an "injection indicator," which is operated directly by the injection valve as it rises from its seat. Fig. 5 shows the neon-lamp equipment installed on the test engine.

In making an ignition-quality test on a Diesel fuel, the engine is operated under specified conditions, which cover the injection quantity and many other factors, and the compression ratio is adjusted until the ignition lag amounts to 13° of crank angle. As injection takes place 13° ahead of dead center, if the lag is 13°, ignition occurs in dead center, and the ignition bulb flashes when it is directly under the sight tube. When injection begins, the engine is still 13° ahead of dead center, but since the slit over the injection bulb is 13° in advance over the ignition bulb, the injection bulb, too, will flash when directly under the sight tube. While there is a small time interval (0.0024 sec) between the two flashes, owing to the persistence of vision they appear to the observer as two simultaneous flashes side by side. When this effect is obtained it proves that the ignition lag is equal to 13°, as required. The length of the combustion chamber is then read

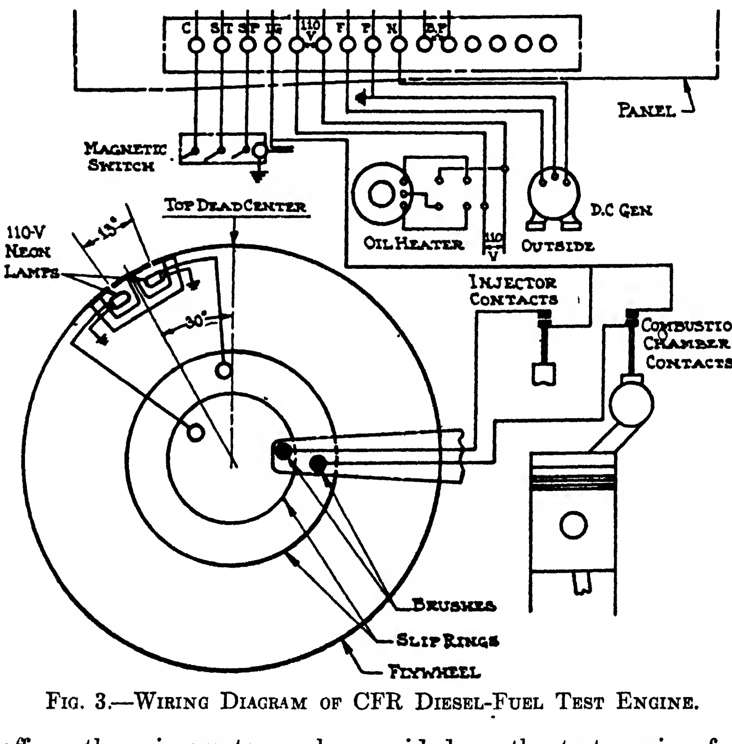


FIG. 3.—WIRING DIAGRAM OF CFR DIESEL-FUEL TEST ENGINE.

off on the micrometer scale provided on the test engine for that purpose.

From the combustion-chamber length thus determined the cetane number of the sample is estimated, and two reference blends are selected, differing by not more than 8 cetane numbers, which will bracket the sample. After two reference fuels have been found which show a shorter and longer ignition delay than the sample, respectively, tests are run on the sample and the reference fuels in the following order: Sample, reference fuel No. 1, reference fuel No. 2, sample, reference fuel No. 1, reference fuel No. 2. In each run the combustion chamber length required to cause the two neon-light flashes to coincide in top center is recorded, and from the six combustion-chamber-length readings the cetane number of the sample is

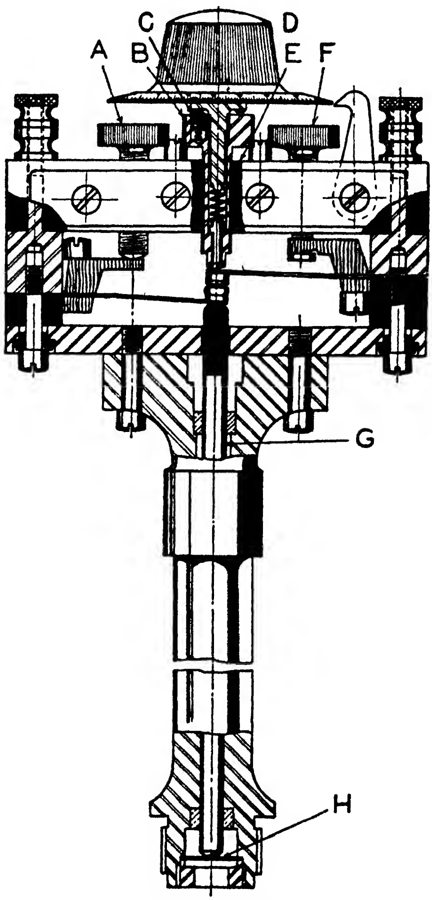


FIG. 4.—IGNITION INDICATOR FOR CFR DIESEL-FUEL TEST ENGINE.

determined by interpolation. It is reported to the nearest whole number, and in case the calculated value falls midway between two whole numbers, the nearest even number is reported.

Cetane Valve—Another method of determining the ignition quality of a Diesel fuel is based on the principle that if the air inlet of the test engine is throttled, the volumetric efficiency at a given speed is lowered, and the maximum pressure and temperature reached by the air during compression are reduced, with the result that eventually a point is reached where the engine fails to ignite. A "master cetane valve" based on this principle has been developed by the research department of the Caterpillar Tractor Company and is illus-

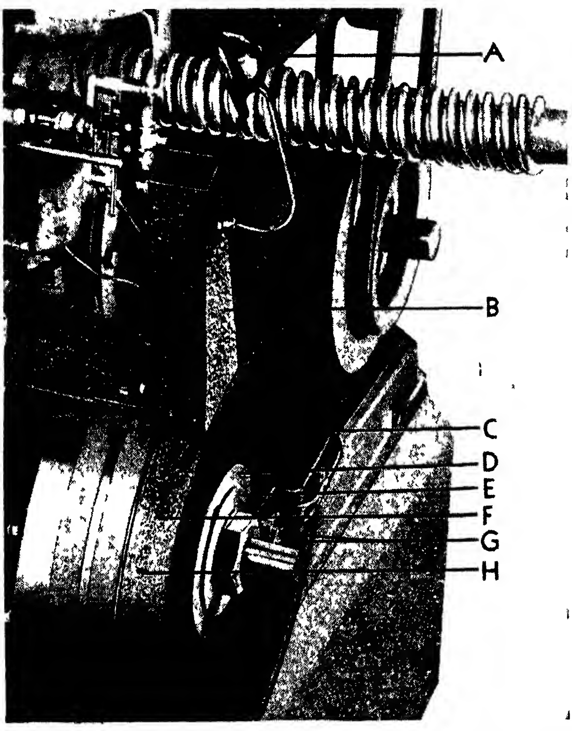


FIG 5.—SHOWING NEON-LAMP AND OTHER EQUIPMENT ON TEST ENGINE

A, mirror; B, sight tube; C, bracket for slip-ring brush; D, slip-ring disc of combustion indicator; E, slip-ring disc of injection indicator; F, combustion slot in neon-tube housing; G, slip-ring brush holders; H, injection slot in neon-tube housing.

trated in Fig. 6. The valve has a rectangular orifice, with means for adjusting the width of the orifice to compensate for changes in atmospheric conditions. A gate valve with calibrated profile can be moved across this orifice at a constant rate by means of an electric motor, and a scale of cetane numbers is secured to the valve in such a way that it can be adjusted in the direction of its length, for calibration purposes.

In making a test, with the engine operating under specified conditions, the valve is gradually closed by the gate moving across its orifice. When firing ceases it is signaled by a puff of white smoke accompanied by an audible noise from the exhaust tap (a small tube connected to the exhaust pipe about 6 in. from the cylinder head). Before making a test, the instrument is calibrated with a reference blend of known composition, to give a correct reading at the lower end of the scale. Once the scale has been correctly adjusted, the test can be proceeded with and the cetane number of the test fuel read directly on the scale. It was found that under the fixed conditions of the test (engine speed, jacket temperature, injection quantity, etc), ignition fails with 25-cetane fuel when

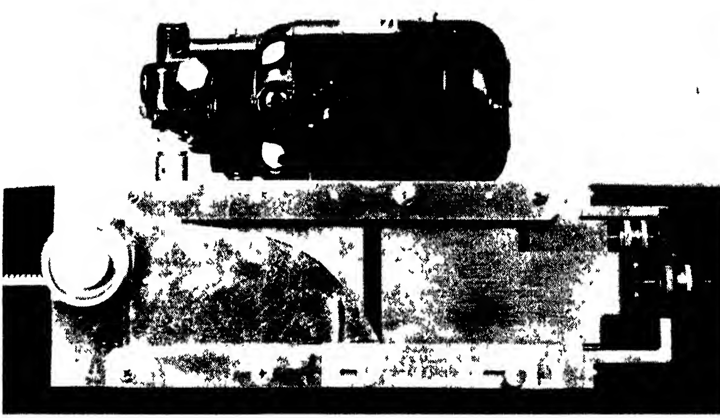


FIG 6—CATERPILLAR CETANE VALVE.

the compression drops to 395 psi, with 45-cetane fuel at 250 psi, and with 65-cetane fuel at 195 psi. Ratings by the cetane valve are said to correlate well with those obtained by the coincident-flash method. The cetane valve has been used for the most part on a Caterpillar 5¾ x 8-in. engine, on which it gives ratings averaging 1 cetane number higher than those obtained by the coincident-flash method.

Correlation Between Ignition Quality and Other Properties.—There has been a good deal of speculation regarding the possible dependence of ignition quality on other properties of the fuel, such as its density (A.P.I. gravity), its carbon/hydrogen ratio, volatility, heat value, viscosity, and aniline point, and extensive statistical studies have been made with a view to correlating these various properties. It has been found, however, that the ignition quality, that is, the readiness of a fuel to ignite, depends more on the molecular structure of its constituents than on any other factor. There is a close relation between the anti-detonating value of a fuel when used in a spark-ignition engine and its ignition quality when used in a compression-ignition engine. In a spark ignition engine a fuel gives rise to detonation if it ignites too readily, while in a Diesel engine a fuel is apt to cause harsh or rough operation if it does not ignite sufficiently readily and must be raised to an abnormally high temperature to cause it to ignite. Therefore, an anti-detonating fuel for carburetor engines—a fuel of high octane number—is the same as a Diesel fuel of low ignition quality—one of low cetane number. A fuel of high cetane number is one which ignites spontaneously at a relatively low temperature.

Molecular Structure of Hydrocarbons—All of the motor fuels are made up principally of hydrocarbon compounds, and so far as ignition quality is concerned, there are essentially four types of such compounds, viz., straight-chain compounds, branched-chain compounds, ring compounds, and branched-ring compounds. Molecular-structure diagrams for one compound of each type are given in Fig. 7. Carbon has a valency of four, and each carbon atom in the molecule is linked to other atoms by four bonds. Hydrogen has a valency of one, and each hydrogen atom connects to another atom by a single bond. Heptane, the straight-chain compound represented in Fig. 7, is a member of the paraffin or saturated series of hydrocarbons, and in the heptane molecule there are seven carbon atoms arranged in a straight line, each linked to one or two other carbon atoms and to two or three hydrogen atoms.

Before there can be combustion (chemical union of carbon and hydrogen atoms with oxygen atoms), the hydrocarbon molecule must be broken up, and it has been found that the straight-chain molecule breaks up most readily. That is to say, straight-chain hydrocarbons have the lowest ignition temperature, or, in Diesel-fuel terminology, they are of high ignition quality. The longer the chain, that is, the more carbon atoms there are in the molecule, the lower the temperature of self-ignition and the higher the ignition quality.

The branched-chain type of hydrocarbon is represented in Fig. 7 by iso-octane. This molecule, being more compact, is harder to break up than the straight-chain type, and its ignition temperature is much higher than that of normal heptane. In fact, these two hydrocarbons are so far apart in respect to ignition quality that they furnish the extremes for the scale of octane numbers used in connection with carburetor fuels, heptane having an octane number of zero and iso-octane of 100. The cetane number of heptane is approximately 60, that of iso-octane 22.

Ring-type hydrocarbons are represented in Fig. 7 by benzene (C_6H_6), a member of the aromatic series and the chief constituent of benzol. This molecule also cannot be broken up readily, and benzol has a high ignition temperature (a high octane number), which makes it an excellent carburetor fuel. In fact, benzol often is added to gasoline to

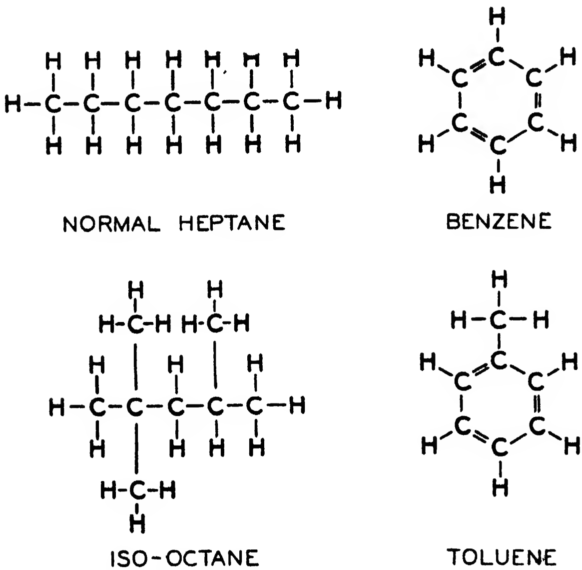


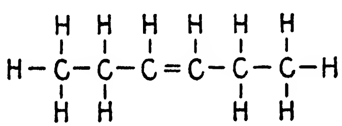
Fig 7—Molecular Diagrams of Four Types of Hydrocarbons.

improve the anti-knock qualities of the latter. For the same reason it is a poor fuel for compression-ignition engines; its high ignition temperature results in a long ignition lag and in rough operation.

The branched-ring type of hydrocarbons is represented by toluene (C_7H_8), another member of the aromatic series. This molecule is still more difficult to break up than that of benzene, and at one time toluene was used as a reference fuel in determining the anti-detonating qualities of carburetor fuels.

Olefins and Naphthenes—Two other series of hydrocarbon compounds, known respectively as the olefin and the naphthene series, are found in smaller or larger proportion in a good many petroleum fuels. The paraffin series, already mentioned, has the general molecular formula C_nH_{2n+2}, there being two more than twice as many hydrogen as carbon atoms in the molecule. These are called saturated hydrocarbons, because it is impossible to add hydrogen atoms to the molecule. The olefin and naphthene series, on the other hand, are unsaturated, and both have the same general molecular formula, C_nH_{2n}. But while both are made up of carbon and hydrogen in the same proportion, the molecular structures of the two are entirely different. Olefins have straight-chain molecules, while those of the naphthenes are of the closed-ring type. Molecular diagrams of one member of each series —hexene of the olefin and cyclohexane of the naphthene series—are shown in Fig. 8.

Ignition Qualities of Pure Hydrocarbons—Generally speaking, the paraffins have the highest and the aromatics the lowest ignition qualities. In each series, of course, the igni-


HEXENE

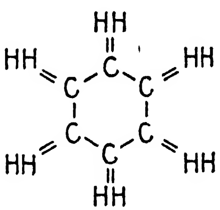

CYCLOHEXANE

FIG. 8—MOLECULAR DIAGRAMS OF AN OLEFIN AND A NAPHTHENE HYDROCARBON.

tion quality varies with the number of carbon atoms in the molecule, and it is interesting to note that while in the paraffin series the cetane number goes up and down with the number of carbon atoms, in the aromatic series an increase in the number of atoms entails a decrease in the ignition quality, and vice versa. Cetane numbers for three members of each of the two series are tabulated below:

Paraffin Series	*Aromatic Series*
(C_nH_{2n+2})	(C_nH_{2n-6})
Heptane, C_7H_{16}, 60	Benzene, C_6H_6, 0
Dodecane, $C_{12}H_{26}$, 80	Toluene, C_7H_8, −5
Cetane, $C_{16}H_{34}$, 100	Xylene, C_8H_{10}, −10

The naphthenes, which resemble the aromatics in that they also are closed-ring compounds, also have low ignition qualities. Cyclohexane, for instance, of which the molecular structure is shown, is said to have a cetane number of 13. The iso-paraffins have the same low ignition qualities as the ring compounds, iso-octane, C_8H_{18}, having a cetane number of 24. Of the various cetane numbers here given, that of cetane, 100, is, of course, fixed by definition. The others should be regarded as approximations rather than absolute values, as relatively few cetane-number determinations on pure hydrocarbons have been published so far.

Differences between Diesel fuels with respect to ignition quality are largely the result of differences in aromatics content. Crude oils from certain fields, such as that of southern Texas, are rich in aromatic constituents, and while these crudes yield excellent gasoline, they give a gas oil of low cetane number which gives poor results in engines. Pennsylvania crude oils, on the other hand, which are of paraffinic base, give an excellent gas oil, and the best Diesel fuels, those of highest cetane number, are straight-run distillates from paraffinic crudes.

Proportions of Hydrocarbon Groups in Diesel Fuels—A research project relating to the composition and properties of Diesel fuels has been under way for some time at the Bartlesville, Okla., Petroleum Experiment Station of the U. S. Bureau of Mines, in cooperation with a number of technical and industrial organizations and the Navy. Early tests were made on a straight-run and a catalytically cracked Diesel fuel, and it was found that whereas the former contains about 80 per cent of paraffins and naphthenes, the proportion of these hydrocarbons in the latter is only about 61 per cent. Table I gives the composition in terms of hydrocarbon series or groups, and some of the principal physical properties of a number of Diesel fuels used in SAE combustion tests in 1947. Fuel A is a blend consisting mainly of catalytically cracked stocks; B, a straight-run fuel of Mid-Continent base; C, a heavy, catalytically cracked stock prepared to Navy specifications, and D, a straight-run fuel corresponding to Navy specification 702-E.

TABLE I—COMPOSITION AND PROPERTIES OF TYPICAL DIESEL FUELS

Fuel	*A*	*B*	*C*	*D*
Aromatic content *	24.0	13.0	31.0	16.0
Naphthene content	17.0	22.2	32.0	24.2
Olefin content	10.5	4.5	8.5	3.5
Paraffin content	48.5	60.3	28.5	56.3
Specific gravity	0.856	0.823	0.891	0.834
Cetane number	41.0	54.0	29.0	52.0

* In per cent of carbon atoms.

It can be seen from the table that the fuel containing the greatest proportion of paraffins has the lowest specific gravity and the highest cetane number, while the one containing the greatest proportion of aromatics has the highest specific gravity and the lowest cetane number.

Ignition Quality vs. Volatility—For the normal paraffins the ignition temperature decreases with the volatility, octane (C_8H_{18}), for instance, having a lower ignition temperature than hexane (C_6H_{14}) and decane ($C_{10}H_{22}$) a lower ignition temperature than octane. When a crude oil or a gas oil cut is broken up into small fractions by distillation, there is a similar relation between the volatility (the distillation temperature) and the cetane number. Fig. 9 shows results obtained by T. B. Hetzel at Pennsylvania State College with a particular gas oil by distilling off small fractions and testing them for ignition quality. In this particular case the first fraction showed a cetane number of 42, and the ignition quality rose continually until toward the end three successive cuts showed 62.5 cetane numbers each, but the very last cut dropped to 53 cetane number. In tests with another fuel from a different source, the cetane number rose rapidly at first, but remained practically constant after about 17 per cent had been distilled off.

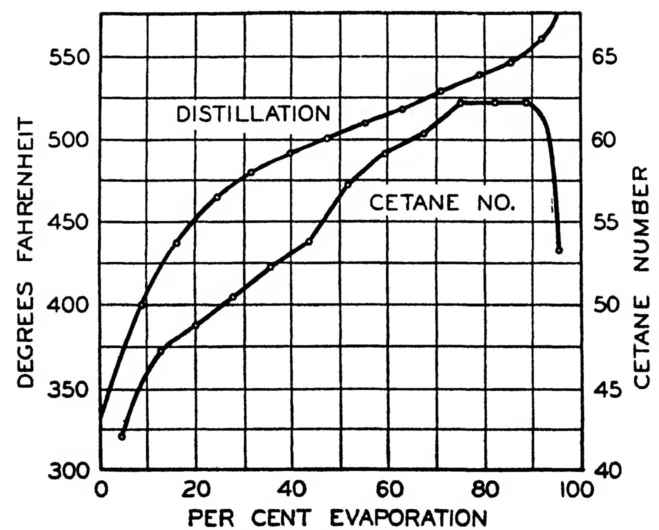


FIG. 9.—CETANE NUMBERS OF SMALL FRACTIONS OF A DIESEL FUEL.

Diesel Index—Determination of the cetane number involves rather laborious tests necessitating expensive equipment, and the idea naturally suggested itself that this characteristic of Diesel fuels might be definitely related to some of the physical properties which are easily determined by simple means. Some work along this line was done by the Standard Oil Development Co., and A. E. Becker and H. G. M. Fischer of that company in 1934 suggested the use of a "Diesel Index" which they defined as follows:

$$\text{Diesel Index} = \frac{\text{Aniline point (Deg F)} \times \text{A.P.I. gravity}}{100}.$$

The aniline point is determined by heating a mixture consisting of equal volumes of aniline and the fuel sample, in a jacketed test tube until it forms a clear solution, and noting the temperature at which turbidity appears on cooling. This factor is dependent on the proportion of aromatic and non-saturated hydrocarbons in the fuel, varying inversely thereto.

The A.P.I. gravity (formerly known as Baumé gravity) is related to the specific gravity by the following equation:

$$\text{Specific gravity} = \frac{141.5}{131.5 + \text{A.P.I. gravity}}.$$

The Diesel Index is a figure which usually is not very different in magnitude from the cetane number of the same fuel. That the ignition quality of a Diesel fuel varies rapidly with the aniline point and in the same sense is clearly shown by Fig. 10, where aniline-point and cetane-number data for 17 different straight-run gas oils are plotted. From the median line drawn through the observation points it appears that when the A.P.I. gravity is increased from 120 to 180, the cetane number increases substantially from 30 to 61.

A weakness of the Diesel-Index system of evaluating ignition quality is apparent from Fig. 11 in which the aniline point is plotted against the A.P.I. gravity for a large number of Diesel fuels from different crudes. It will be seen from this that if we confine ourselves to fuels from a given crude, the A.P.I. gravity, instead of varying directly and almost in the same proportion as the aniline point, actually varies in the opposite sense. It is true that if all of the different observation points are considered, and a median line is drawn through them (the curved, dashed line in the chart), the aniline point varies in the same sense and at nearly the same rate as the A.P.I. gravity. But it is obvious from the plot that for fuels from the same crude, of minimum and maximum gravity respectively, figures given by the Diesel-Index formula cannot possibly correlate well with the cetane numbers. Let us take, for example, the heaviest and lightest gas oils from the California crude. Since there is little difference between their aniline points, we would expect their cetane numbers to be nearly alike, and so they are, the heaviest, with an A.P.I. gravity of 23.4, having a cetane number of 43.5, and the lightest, of 40.2 A.P.I. gravity, having a

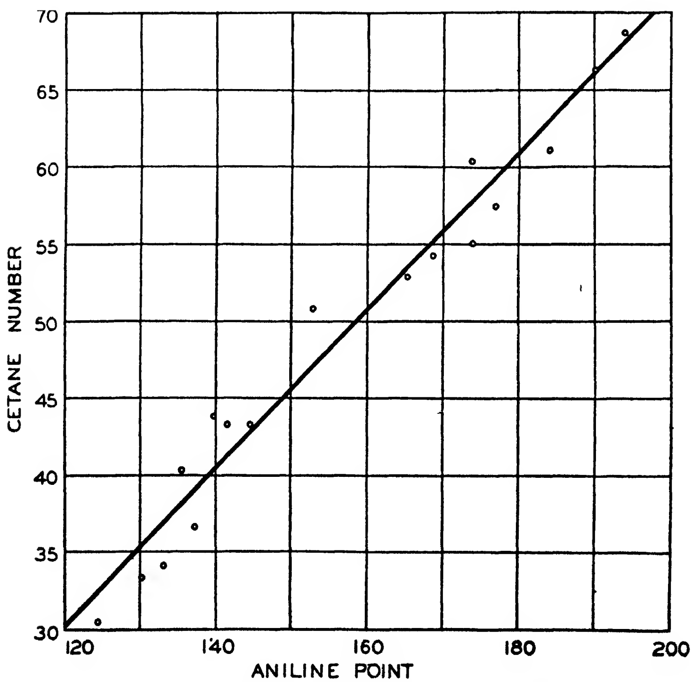


FIG. 10.—RELATION BETWEEN ANILINE POINT AND CETANE NUMBER.

cetane number of 40.4. However, owing to the great difference in the A.P.I. gravities, the Diesel-Index formula gives widely different values, 34 for the heaviest and 54.5 for the lightest fuel.

The straight median line in Fig. 10 corresponds to the equation

$$\text{Cetane number} = \frac{\text{Aniline point}}{1.95} - 31.5.$$

A check on data for 33 different Diesel fuels given in a paper by W. G. Ainsley, H. D. Young, and M. L. Hamilton (*S.A.E. Journal*, April, 1942) showed that this formula gives results that correlate considerably better with test results than the Diesel-Index formula.

As there is substantially a straight-line relation between the aniline point and the cetane number, the aniline-point scale in Fig. 11 could be replaced by a cetane-number scale which would have uniform divisions and extend from 30 to 71. From this it is apparent that as far as straight-run Diesel fuels are concerned, those from crudes from Pennsylvania have, on the whole, the best ignition qualities, followed by those from crudes from the Mid-Continent, California and South-Texas fields, in the order named.

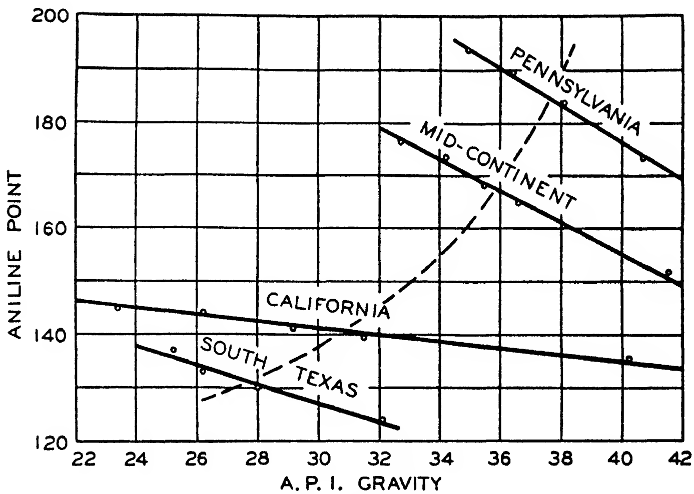


FIG. 11.—RELATION BETWEEN A.P.I. GRAVITY AND ANILINE POINT.

Heat Content—An average gas oil has a lower heat value of about 18,600 Btu per pound and a higher heat value of 19,800 Btu per pound. The lower heat value is the amount of heat obtained if a pound of the fuel is burned in an excess of air and the products of combustion are cooled to 212 F without condensation of the steam contained, while the higher heat value is the amount of heat obtained from the same combustion if the products are cooled to 60 F. Most of the difference between the two values is due to the latent heat liberated by condensation of steam in the products of combustion when the latter are cooled to 60 F.

Test engineers usually express heat values on the weight basis, because they determine the fuel consumption by weighing in lb per bhp-hr. However, since fuels are bought on a volume basis, and also because injection systems deliver a definite volume of fuel per cycle regardless of its density, the heat value per gallon enters into many calculations. This value increases with the density of the fuel, and therefore as its A.P.I. gravity decreases. A number of the physical properties, including the heat value per unit of volume, the A.P.I. gravity, the cetane number, the viscosity, and the volatility as represented by the mid-boiling point, are interrelated, and the relationships are shown by Fig. 12, a chart which originated in the Esso Laboratories and which is here reproduced from a report of the C.F.R. Automotive Diesel Fuels Division. This chart shows that for a given mid-boiling point or a given viscosity the cetane number varies directly with the A.P.I. gravity, and that for a given cetane number the A.P.I. gravity varies inversely as both the mid-boiling point and the viscosity.

Effect of Gravity on Power and Consumption—Most engines are provided with a fuel stop which definitely limits the maximum volume of fuel that can be injected per cycle. As there is always an excess of air in the cylinder, more power is developed with the heavier or denser fuels, a charge of which contains more heat units. For the same reason, a lower specific fuel consumption on the volume basis can be expected with the denser (low A.P.I. gravity) fuels. However, since a unit volume of the denser fuels weighs more, on the weight basis the specific consumption is greater with the denser fuels. In tests made for the C.F.R. Automotive Diesel Fuels Division the specific consumption dropped from 0.62 pint per bhp-hr with 44 A.P.I. gravity fuel to 0.58 pint with 32 A.P.I. gravity fuel, or about 6.5 per cent, but it increased from 0.509 lb per bhp-hr to 0.517 lb, about 1.57 per cent.

Viscosity and Pour Point—Since the fuel must pass through the small-diameter tubes and the fine-mesh filters at the lowest operating temperatures, its viscosity is of importance. There must be no excessive resistance to flow under these conditions, which makes it necessary to set a maximum limit on the viscosity. However, there is need also for a minimum limit, because the fuel is depended upon for the lubrication and the sealing of pump and injectors, and the more viscous fuels have better lubricating qualities. With fuels of very low viscosity there is generally some loss of engine power, because these fuels have lower densities and heat values. The loss of power may be aggravated by injection-pump leakage, which naturally will be greater with fuels of low viscosity. Owing to the higher viscosity of gas oil as compared with gasoline, ordinary automobile-type fuel pumps are not sufficiently powerful, and special Diesel fuel transfer pumps are being used. For a specification covering Diesel fuels suitable for use under all ordinary conditions it has been proposed to set the minimum viscosity at 33 and the maximum at 43 sec Saybolt Universal, both at 100 F.

The lowest temperature at which the fuel will flow through a pipe is known as the pour point or freezing point. Most

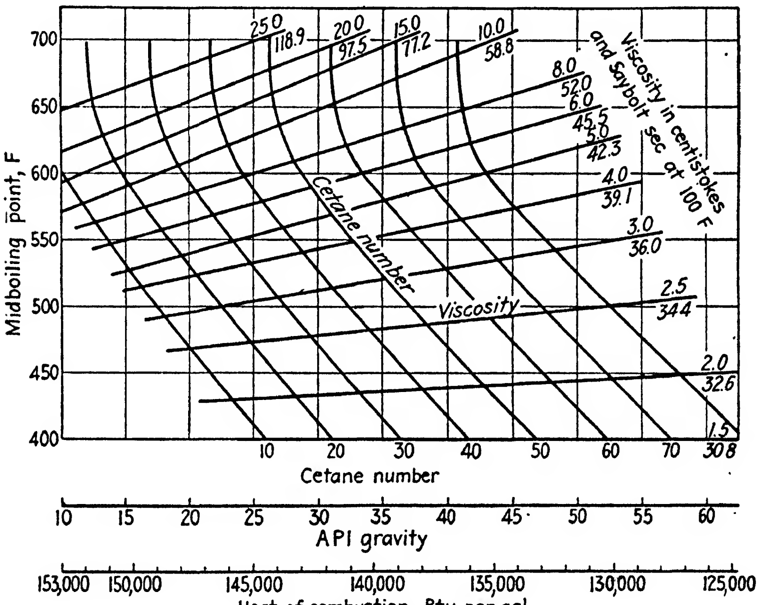


FIG. 12.—RELATIONSHIP BETWEEN VARIOUS PHYSICAL PROPERTIES OF DIESEL FUELS.

furnace oils have a freezing point well below zero F. In writing specifications it is advisable not to set the pour point any lower than absolutely necessary, so as not to exclude fuels which are among the most desirable from other points of view. Viscosity and pour point usually go up and down together (though there is no definite relation between the two), and fuels of high viscosity and high pour point have the greatest heat value.

Volatility—The volatility of a fuel is indicated by its distillation curve, in which percentages of fuel distilled are plotted against temperature. Fig. 13 shows such distillation curves for four different fuels and for a half-and-half mixture of two of them, which were used in the experimental investigations of the C.F.R. Automotive Diesel Fuels Division. These fuels cover a wide range of volatility and, incidentally, a wide range of ignition quality. Volatility has no fixed relation to ignition quality, as shown by the fact that fuels B and D, which are extremes with respect to volatility, have practically the same cetane number, while fuels A and C, which differ only slightly with respect to volatility, have widely

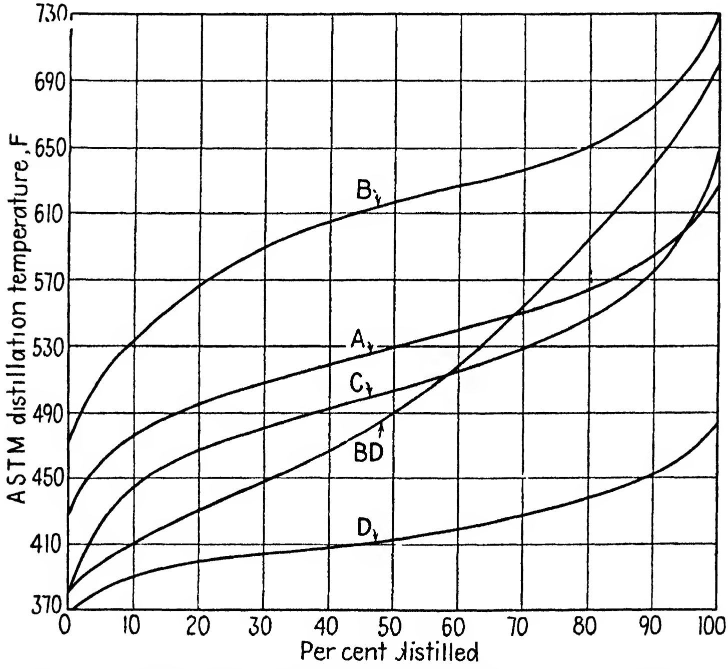


FIG. 13.—DISTILLATION CURVES OF A NUMBER OF DIESEL FUELS.

different cetane numbers. Fuel B, of course, has a much higher viscosity and a much higher pour point than fuel D. The experiments showed it to be desirable to limit the 90 per cent point to about 650 F and the end point to about 700 F, because if these points are raised farther, both engine deposits and smoke in the exhaust increase rapidly.

In evaluating the significance of different fuel qualities, greatest importance usually is attached to their effects on power, economy, and smoothness of operation under normal conditions. However, most engines are difficult to start in cold weather and operate rather erratically when under light loads for considerable periods; and it has been found that fuel volatility has a decided effect on engine behavior under these conditions. F. G. Shoemaker and H. M. Gadebusch in an S.A.E. paper reported that in the General Motors two-stroke Diesel engine, Western stove oils, which usually have a cetane number of less than 40 but an end point as low as 560 F, give as good operating results as No. 3 fuel oil with 50-55 cetane number and 675 F to 700 F end point. They say that in city bus operation, owing to frequent low load factors and drastic restrictions on exhaust smoke and odor, it is necessary to use special fuels, and experience has shown that in such service high volatility has the same beneficial effects as high ignition quality. To throw further light on the relationship between the effects of volatility and ignition quality, some tests were run with a blend consisting of 75 per cent aviation gasoline and 25 per cent of lubricating oil. This had a very high volatility, but its cetane number was only 21. All of the test engineers who used it agreed that this blend gave very smooth operation for a fuel of such low cetane value. It was found that with this blended fuel the ignition lag was the same as with a regular Diesel fuel of the same low cetane number, but that combustion extended over a period 42 per cent longer, which would seem to warrant the conclusion that the pressure rise was less rapid.

Sulphur Content—The presence of sulphur in Diesel fuel is objectionable, as it increases piston-ring and cylinder wear, as well as the formation of varnish on piston skirts and of oil sludge in the crankcase. It is therefore necessary to put a limit on the maximum sulphur content.

During the early post-war period some users of Diesel engines experienced trouble from excessive liner wear, especially in districts where the fuel available had a rather high sulphur content. The matter was brought to the attention of engine manufacturers and oil companies, and its seriousness was quickly recognized, because high sulphur fuels were very common in the foreign market and were being offered in increasing quantity in the home market. While it is possible to remove the sulphur from the fuel by chemical treatment, the process is a rather expensive one.

Nitrogen Compounds—High-sulphur fuels often also contain considerable quantities of nitrogen compounds, and there seemed to be a possibility that the high wear rate was at least partly due to the nitrogen. To get at the root of the matter, two research projects were instituted, one by the Union Oil

Company of California, the other by Caterpillar Tractor Company, and papers relating to both of these investigations were presented at engineering meetings.*

Both investigations showed that the nitrogen content has only a minor effect on the wear rate. Union Oil Company ran a series of tests in which a fuel containing 0.17 per cent of nitrogen and no sulphur and a fuel containing 0.7 per cent of sulphur and no nitrogen were compared with a fuel practically free from both contaminants. The nitrogen content of 0.17 per cent is abnormally high, twice that of a high-nitrogen commercial fuel, and was obtained by adding nitrogen bases from Diesel fuel. It was found that the nitrogen increased the wear rate by less than 50 per cent, whereas the sulphur increased it by more than 500 per cent.

Fuels with less than 0.5 per cent sulphur produce only light deposits in the engine, but when the sulphur content is raised beyond this point the pistons become progressively dirtier, the ring grooves become packed with carbon, and the rings become more sluggish in action. Ring sticking is common with a sulphur content of more than 1.00 per cent. An analysis of a compounded oil that had been drained from the crankcase after 480 hr operation on high-sulphur fuel showed a greater loss of additive than experienced with fuel free from sulphur. It can be laid down as a general rule that the lubricating qualities of an oil deteriorate more rapidly the greater the sulphur content of the fuel.

When sulphur is burned in the atmosphere it forms sulphur dioxide, but it has been shown by C. H. Cloud and A. J. Blackwood of Esso Laboratories, by exhaust gas analyses, that most of the sulphur in fuels burned in Diesel engines forms sulphur trioxide, which latter combines with water to form sulphuric acid. They also showed that the trioxide is responsible for the increase in wear and in varnish and sludge formation observed when high-sulphur fuels are used, by feeding sulphur compounds with the intake air while the engine was being "motored." Since there was then no combustion in the engine, the nature of the sulphur compounds did not change. Sulphur trioxide fed under these conditions increased the wear and sludge formation the same as the use of high-sulphur fuel, while sulphur dioxide supplied in the same manner had no such effect.

* Effect of Nitrogen and Sulphur Content of Fuels on Diesel-Engine Wear, by C. C. Moore and W. L. Kent, Union Oil Co. of California, S.A.E. Quarterly Transactions, October, 1947.

Effect of Diesel Fuels on Deposits and Wear, by L. A. Blanc, Caterpillar Tractor Co., S.A.E. Quarterly Transactions, April, 1948.

Antidote for High Sulphur—Investigations have shown that compounded or E.P. oils reduce the rate of wear with high-sulphur fuels, but that the effects of different additives vary widely. In recent years a number of oil companies have come out with special lubricants for Diesel engines operating on high-sulphur fuel. One of these lubricants is described as a highly refined mineral oil with additives giving it extreme-pressure characteristics and enabling it to give protection under boundary-lubrication conditions. The oil, moreover, has a negative neutralization number, which tends to reduce acid formation.

Diesel Fuels Classification—A classification of Diesel fuels has been published by the American Society for Testing Materials at Philadelphia, and includes one grade, 2-D, to cover the requirements of all mobile and high-speed Diesel powerplants. This specification sets a limit of 1 per cent on the sulphur content. Since the adoption of this classification the use of high-speed Diesels has increased widely in many different fields, and the fuel requirements therefore vary widely. The subject has been under continuous study of the C.F.R. Automotive Diesel Fuels Division, which in 1945 adopted a new Diesel fuels classification covering four grades, as follows: 1-D, a distillate fuel of high volatility for use in high-speed engines (1200 rpm and over); 2-D, a distillate fuel of medium volatility for high-speed engines (1200 rpm and over); 3-D, a distillate fuel of low volatility for medium-speed engines (500-1200 rpm), and 4-D, a viscous fuel for low-speed engines (below 500 rpm). The sulphur limits for these four groups are 0.5, 1.0, 1.5 and 2.0 per cent respectively.

Other items usually included in fuel specifications include the flash point (which has no material effect on the behavior of the fuel in the engine but affects its safety in storage); the water and sediment content, which must be kept low for obvious reasons, and the Conradson carbon residue, which affects the formation of deposits in the combustion chamber and on the fuel nozzles. This latter value is determined by an A.S.T.M. test applied to 10-per cent bottoms or residues of the fuel in question.

The following table gives the specifications for three of the four fuel grades classified by the C.F.R., the 4-D grade being of no interest here because it is intended solely for use in low-speed engines. It may be noted here that the C.F.R. grade 2-D is identical with the A.S.T.M. grade 2-D, except as regards the following items, which in the A.S.T.M. grade have the values given in parentheses: Carbon residue (0.20 per cent), pour point (20 F) and cetane number (45).

C.F.R. SPECIFICATIONS FOR FUELS FOR HIGH- AND MEDIUM-SPEED DIESEL ENGINES

Grade	*1-D*	*2-D*	*3-D*
Flash point, min., F	100	140	140
Water and sediment, % by vol.	0.05	0.05	0.10
Carbon resid., % by weight, max.	0.05	0.25	0.25
Ash, % by weight, max.	0.01	0.01	0.02
90 per cent point, max., F	550	650	......
End point, max., F	.....	700	......
Kin. visc., centistokes			
min.	......	2.0	......
max.	......	6.0	12.0
Saybolt Universal visc., sec			
min.	......	32.6	......
max.	......	45.5	65
Sulphur, 5 by weight, max.	0.5	1.0	1.5
Corrosion	Pass	Pass	......
Alk. and min. acids	None	None	None
Cetane number, min.	40	45	35

Pour points are not fixed, but it is recommended that in order to avoid flow restrictions in cold weather this point be specified as 10 per cent below the lowest temperature reached by the fuel in service. In a footnote to the specifications the Committee note that although information regarding the absolute maximum limits of sulphur is still inconclusive, recent evidence indicates that the limits recommended in the specifications may be excessive.

Ignition Accelerators—It was first shown by Harry Ricardo in England that the ignition quality of a Diesel fuel can be improved by the addition of small proportions of certain chemical substances, the same as the anti-detonating qualities of carburetor fuels are improved by the addition of tetra-ethyl lead. In the case of carburetor fuels the anti-detonating agents or dopes have the effect of lowering the rate of combustion, and are sometimes referred to as negative catalysts. What is required in the case of Diesel fuels is an agent which will expedite the ignition of the fuel, as by reducing its temperature of self-ignition. Ricardo found that the addition of a small percentage of amyl nitrate (which when added to gasoline causes severe knocking) has the effect of decreasing the ignition lag in Diesel engines. Another similar catalyst is ethyl nitrate, and the effect of this fuel dope on the ignition lag is clearly shown by the superposed indicator diagrams reproduced in Fig. 14. In both cases the engine was run at 1000 rpm, and the load was adjusted to a bmep of 83.5 psi. The full-line curve was obtained with an

Asiatic fuel in its natural state, while the dashed-line curve was obtained with the same fuel after 5 per cent of ethyl nitrate had been added. It will be seen that the ignition lag was cut down from 13° to 10° of crankshaft motion. Beneficial effects of this reduction in the ignition lag are apparent in the less abrupt rise of the pressure line upon ignition and the lower maximum pressure (remembering that the bmeps are the same for both diagrams). It was said, moreover, that the engine ran much more satisfactorily on the doped fuel.

Study of Accelerators by the Navy—The U. S. Navy has explored the field of Diesel-fuel dopes (or ignition-quality improvers, as it calls them) ever since the early 1930's, when it appeared that Diesel engines might be suitable for naval air-

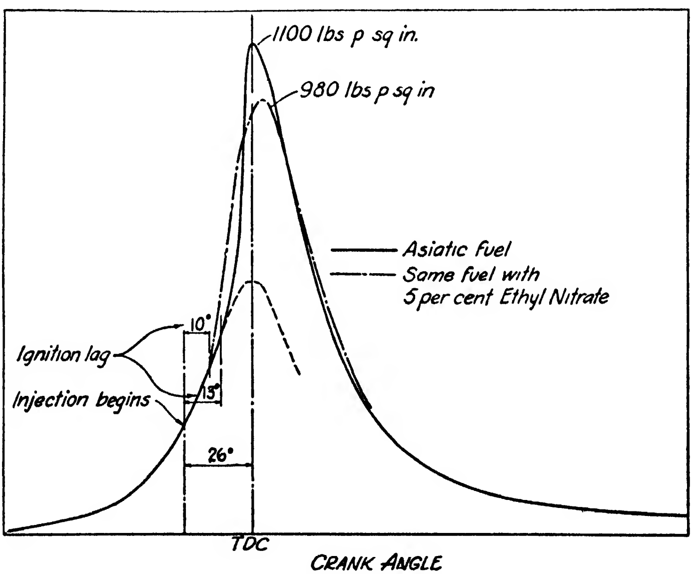


FIG. 14.—EFFECT OF FUEL DOPE ON IGNITION LAG.

craft of the lighter-than-air type. Up to the time of this writing, however, the Navy had never used the improvers. The subject is of particular interest to the Navy because Diesel-equipped craft may have to be operated in the Arctics, where only fuel with a pour point far below 0 F is practical in winter time, and for petroleum fuels of the same origin the cetane number and pour point go up and down together. More than 70 different additives or dopes were tested, but later work was confined to four types, three of them nitrogen compounds and one a peroxide.

It was found that the effectiveness of the additive, that is, the increase in the cetane number due to a certain quantitive addition, varies with the base fuel. When the cetane number is raised by the use of an additive, the rate of pressure rise and the maximum combustion pressure are lowered, and as a rule the engine will start more easily under unfavorable conditions, and produce less smoke. On the other hand, the maximum power is usually reduced, and the specific fuel consumption somewhat increased. Some of the additives have a tendency to promote ring sticking. The ignition-quality improvement due to additives generally decreases during storage. An account of the Navy's investigation is given in "Performance and Stability of Some Diesel-Fuel Ignition-Quality Improvers" by W. E. Robbins, R. R. Audette, and N. E. Reynolds III, in the S.A.E. Quarterly Transactions for July, 1951.

Accelerators Not Yet Commercialized—Although, in addition to the Navy, a number of chemical and petroleum firms have explored the field of possible chemical ignition accelerators, the latter so far have not come into commercial use. The situation in this respect is rather different from that in the field of carburetor fuels. As the octane number of commercial gasoline was increased by the addition of "dope," engine manufacturers raised the compression ratio and thereby increased the thermal efficiency. Sometimes the compression ratio was increased too much, and further improvement in fuels by the addition of more dope was then called for. No equivalent gain is to be expected from ignition accelerators for Diesel fuels, though it is generally admitted that if the cetane number of a Diesel fuel drops below a certain value, it not only leads to roughness in operation, but also makes starting more difficult, and increases the smokiness and odor of the exhaust and the deposits on cylinder walls.

Any increase in the cetane number of the fuel due to the addition of an ignition accelerator would make it possible to work with a lower compression ratio in the engine, but that generally would result in a decrease in thermal efficiency. It now looks as though little if anything were to be gained by increasing the cetane number of a Diesel fuel above 50, and that figure can be obtained in straight-run distillates from different crudes without the addition of chemical agents. The cetane numbers of cracked fuels, of course, run considerably below this figure. The large increase in the demand for fuel for high-speed Diesel engines and the simultaneous increase in the production of fuel by the catalytic cracking process is making it necessary to turn more and more to this type, and improvement of its ignition quality is therefore highly desirable. This already has led to a revival of interest in ignition accelerators. Among the substances which have been found to improve the ignition qualities of Diesel fuels are amyl and ethyl nitrate and acetone peroxide. A pamphlet on Ignition Accelerators for Compression-Ignition Engine Fuels, by J. S. Bogen and G. C. Wilson, was published by Universal Oil Products Company of Chicago in 1946. It contains an extensive bibliography on the subject.

One objection to the use of some of the accelerators now known is that in order to produce a worth-while effect, they must be added in considerable quantity, and they then add materially to the cost of the fuel. It has been found, for instance, that a certain cracked fuel from Pennsylvania crude, which in its natural state has a cetane number of 35, requires the addition of 0.55 per cent amyl nitrate by volume to bring its cetane number up to the specification minimum of 45; similarly, a Mid-Continent cracked gas oil, with a cetane number of 32, requires the addition of 1.30 per cent, and an Illinois cracked gas oil, of 26 cetane number, 2.75 per cent. The effect of the ignition accelerator drops with increase in the addition; that is to say, an addition of 2 per cent will not raise the cetane number twice as much as an addition of one per cent.

If Diesel-fuel ignition-quality improvers should come into commercial use, it probably will be because the rapid increase in the demand for such fuel has made it necessary to resort to fuels which cannot be burned satisfactorily without the use of the "improvers."

CHAPTER V

Fuel Injectors and Sprays

The fuel injector or spray nozzle serves two principal functions: It must finely divide (atomize) the fuel, and it must distribute the same through the air charge in the combustion chamber. For the first function dependence is placed on the injector alone, while dispersion of the fuel through the air charge may be assisted by causing the air to flow in the combustion chamber during the period of injection, in a direction either opposed to the spray from the nozzle or at an angle thereto.

In a common-rail system the injector serves also as a metering device. In such a system the fuel in the lines to the injectors is maintained under a constant pressure, and with a constant injection pressure the quantity of fuel injected depends on the sectional area of the orifice or orifices and on the duration of the injection period.

Types of Nozzles—There are three different types of nozzles, as follows:

1. Open nozzles, which are merely plugs with a small axial hole through them.
2. Valve-closed nozzles which are opened hydraulically, that is, by the pressure to which the pump subjects the fuel in the line at the time of injection.
3. Valve-closed nozzles the valves of which are held open by cam mechanism during the injection period.

Open nozzles and nozzles with hydraulically-actuated valves are adapted for use on engines equipped with the jerk-pump injection system, while the third type, the mechanically-controlled nozzle, is suitable for use on engines with air injection and engines with mechanical injection by the common-rail system. As air injection and the common-rail system are not being used on high-speed Diesel engines, we will not consider the third type of nozzle or injector.

Valve-closed nozzles in which the valves are actuated by the fuel pressure may be divided again into two sub-classes, viz., nozzles with outwardly-opening and inwardly-opening valves, respectively. Actually there is a third sub-class, in which the valve is fixed relative to the nozzle body and the valve seat consists of a number of superposed, perforated discs of spring steel which yield under the pressure impulses imparted to them by the fuel, causing the seat to move away from the valve. This type, however, has been used only in experimental work. Fig. 1 shows a nozzle with outwardly-opening valve, which latter is held to its seat by a coil spring (not shown). When a pressure impulse from the pump arrives at the nozzle, the pressure of the fuel against the valve head compresses the spring and causes the valve to open, whereupon injection begins. When injection is completed, the pressure in the fuel line drops and the valve is closed by the spring.

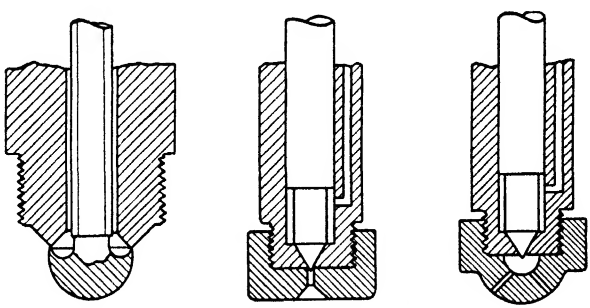
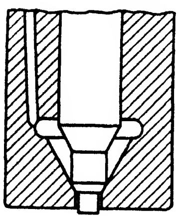


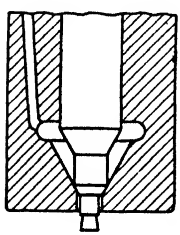
FIG. 1 (*Left*).—SPRAY NOZZLE WITH OUTWARDLY-OPENING VALVE.
FIG. 2 (*Center*).—SINGLE-ORIFICE NOZZLE.
FIG. 3 (*Right*).—MULTIPLE-ORIFICE NOZZLE.

Fig. 2 shows a nozzle of the type with inwardly-opening valve. Here, also, the conical valve is held to its seat by a coil spring. The valve stem is lapped into the bore of the nozzle with a very close fit, and its lower end is reduced in diameter and forms a chamber, to which the fuel from the pump has access through a passage drilled in the nozzle holder. With the valve closed, the fuel pressure acts on the shoulder of the stem and tends to open the valve. As soon as the pressure on the shoulder overcomes the spring pressure, the valve opens and injection begins. The nozzle shown in Fig. 3 also is of the inwardly-opening valve type. It is identical with that of Fig. 2 except for the fact that it has several orifices instead of a single one. Multi-orifice nozzles are used where the injector is depended upon to disperse the fuel throughout the air charge.

Pintle and Throttling Nozzles—Fig. 4 shows the so-called pintle nozzle, in which a pin or pintle extends from the valve through the spray orifice. Pintle nozzles are used primarily in turbulence-type engines, but also to a certain extent in precombustion-chamber engines. The spray pattern depends on the forms of both the orifice and the pintle. In one design the pintle is a rather close fit in the orifice, and the fuel therefore has to pass through a narrow annular opening. Jets from such nozzles are of the form of a hollow cone, the outside angle of which can be made anything desired up to 60 deg. A valuable feature of the pintle nozzle is its self-cleaning property, which prevents carbon deposits from building up in and around the orifice.

A further development of the pintle-type is the throttling nozzle (Fig. 5), in which there is an inverted-cone-shaped extension to the pintle. With this nozzle the area of valve opening increases less rapidly during the early part of the valve-lift period, with the result that when ignition occurs there is less fuel in the combustion chamber, and the rate of pressure





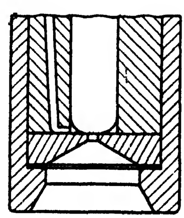


Fig. 4 (*Left*).—Pintle Nozzle.
Fig. 5 (*Center*).—Throttling Nozzle.
Fig. 6 (*Right*).—Nozzle with Flat Valve Seat.

rise following ignition therefore is lower. During this part of the injection period the nozzle produces a wide-angled, thoroughly atomized jet, which probably tends to reduce the ignition lag, while during the latter part of the period it produces a more nearly solid jet. Throttling nozzles are used chiefly in precombustion-chamber engines.

Fig. 6 shows a nozzle with flat contact surfaces on the valve and the nozzle plate or valve plate.

Nozzle Leakage—One difference between nozzles with outwardly-opening and inwardly-opening valves, respectively, is that whereas the former can be completely closed except for the fuel inlet from the pump and the valve-controlled orifice or orifices, the latter cannot. A certain amount of fuel leaks past the valve stems in these nozzles, and provision must be made to return this leakage fuel to the fuel tank or to the inlet side of the injection pump. To minimize the leakage, the valve stem must be made a very close fit in the nozzle body (with a clearance of the order of 0.0001 in.), and this close clearance occasionally gives rise to valve seizures. Such seizures are due either to heat distortion of the nozzle or the valve, or to the entrance of minute dirt particles into the clearance space.

Theories of Liquid Sprays—When a liquid is forced under pressure through a small orifice, it is divided up into fine particles (globules or droplets) which gradually recede from each other, so that the stream issuing from the orifice assumes the form of a cone. There has been a great deal of speculation as to why a "solid" stream of liquid issuing from an orifice should be thus broken up into fine particles. Some have claimed that the chief cause is the air resistance encountered by the stream. While air resistance undoubtedly has an effect on some of the characteristics of the spray, it has been shown that sprays are formed also when a liquid is injected into a vacuum, and air resistance therefore cannot be the primary cause.

Spray Formation in Vacuum—In experiments on spray formation made by Professor K. J. De Juhasz at the Pennsylvania State College, oil was injected into a chamber in which the air pressure was reduced to one-fiftieth atmosphere (0.6 in. of mercury), so that the air resistance was reduced to a very low value. It was found, however, that the liquid was atomized and dispersed approximately the same as in air at atmospheric pressure. An increase in the air density had the effect of increasing the spray angle, making the distribution pattern more nearly even, and reducing the penetration. This result, Professor De Juhasz observed, is strong evidence against the supposition that atomization and dispersion are due solely to air friction.

Similar tests were made by Dana W. Lee of the National Advisory Committee for Aeronautics, and Fig. 7 is a reproduction of photographs obtained by him of sprays into vacuum, into air at atmospheric density, and into air at 14.5 times atmospheric density respectively. The injection pressure in all cases was 4000 psi. These photographs bring out the different characters of the sprays very clearly.

Turbulence of Jet—The most plausible explanation of the breaking up of the "solid" stream into fine particles and of the divergence of the paths of these particles seems to be as follows: When a liquid flows through an orifice of the form generally employed for fuel injection, its individual particles do not move in straight lines parallel with the axis of the orifice; the general motion, of course, is in that direction, but in addition the particles move transversely to the axis of the orifice. If a liquid flows slowly through a tube of very small bore as compared with its length, its particles move parallel to the axis of the tube. This is known as streamline flow or laminar flow. But when the length of the orifice is small compared with its diameter, and especially when the velocity of flow is high and the viscosity of the liquid is low, the motion of the fluid particles is of the irregular form above described, which is known as turbulent flow.

This subject was first investigated by Osborne Reynolds, an English physicist. He caused water to flow through a

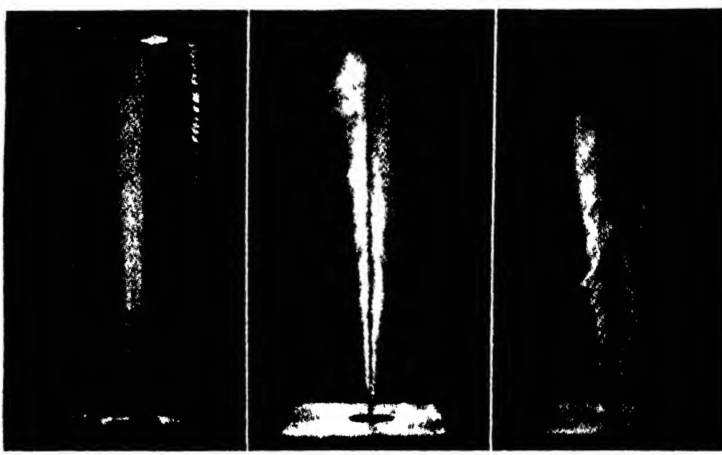


FIG. 7.—PHOTOGRAPHS OF SPRAYS INTO A VACUUM (*Left*), AIR AT ATMOSPHERIC DENSITY (*Center*) AND AIR AT 14.5 TIMES ATMOSPHERIC DENSITY (*Right*).

glass tube, and at the center of the stream he introduced a colored liquid. Under certain conditions, and especially if the velocity of flow was small, the colored liquid would form a distinct straight line throughout the length of the tube, showing that there was streamline flow. Under other conditions, and particularly when the diameter of the tube was large and the velocity of flow high, the colored liquid completely intermingled with the water, showing that there was then turbulent flow. Reynolds found that turbulent flow occurs when the value of the expression $\rho v\delta/\mu$ exceeds a certain number (now known as the Reynolds number), where

ρ is the density of the fluid; v, the velocity of flow; δ, the diameter of the tube; and μ, the absolute viscosity of the fluid.

With turbulent or quasi-turbulent flow through the nozzle orifice, the particles of the fluid have a transverse motion in addition to their motion in the direction of the orifice axis; therefore, when they emerge from the orifice and are no longer constrained by the walls thereof, the particles separate, thus forming the spray cone. Individual droplets assume the form of spheres, owing to the surface tension of the liquid, which tends to make each droplet assume that form which has the least surface area for its volume, that is, a sphere.

Schweitzer's Theory of Jet Disintegration—The foregoing is generally in agreement with a theory propounded by Dr. P. H. Schweitzer in Pennsylvania State College Bulletin No. 40 (1932). Says Dr. Schweitzer:

"If the emerging jet is truly laminar, no immediate disintegration will occur. The liquid particles flow in parallel streams inside the orifice, the layer next to the wall having zero velocity, with the velocity increasing in a parabolic manner approaching the tube axis. Since the wall exerts no direct influence on either the periphery or the inside of the jet, the velocity distribution immediately past the orifice will be the same as inside the orifice. Consequently, no appreciable velocity difference exists between the air and the adjacent liquid close to the orifice, the skin of the jet being virtually stationary. It is clear that air friction cannot atomize a laminar jet as long as the velocity distribution adjacent to the skin remains parabolic, dropping to zero at the periphery. After a certain distance, however, the combined effect of air friction and surface tension gives rise to surface irregularities which lead to disintegration of the jet. . . .

"If the flow in the orifice is turbulent in its entire cross section, the orifice wall is bombarded by liquid particles having radial velocity components. As soon as the restraint imposed by the orifice wall ceases, such particles are only kept in bounds by the surface film. This may soon break under the impact of radial particles, and a general disruption of the jet will occur. A laminar layer may retard this disintegration only for a short period, if at all. Therefore, disintegration due to turbulence will take place shortly after passing the nozzle."

Recent investigations have shown that if a section is taken of a spray cone at some distance from the nozzle, the density of the spray (the mass of fuel passing through a small unit area in unit time) decreases rapidly from the axis of the cone outward, and that the mean diameter of the droplets is smaller near the outside than at the axis of the cone.

Size of Nozzle Orifice Required—To assure thorough atomization, the fuel must be sprayed into the compressed air under high pressure, and this pressure is produced by causing the pump to force the fuel through a small orifice. Assuming the length of the orifice to be less than four times its diameter, which under the conditions of velocity of injection and viscosity obtaining in oil-engine practice assures turbulent flow, there is a definite relation between the diameter of the orifice, the time of injection, the pressure behind the jet, and the quantity of fuel injected. The diameter of orifice required to produce a certain pressure when the quantity to be injected and the time of injection are given may be determined as follows:

The standard equation for velocity of flow from an orifice is

$$v = f\sqrt{2gh},$$

where v is the velocity of flow in inches per second; f, the coefficient of discharge of the nozzle, which depends on the form of the orifice; g, the constant of gravity (386 ipsps), and h, the head on the orifice, in inches of a column of the liquid which is being handled. That a coefficient of discharge must be applied is due largely to the fact that most of the liquid approaches the orifice (whose leading edge is rounded or chamfered) at an angle to its axis, and therefore has a radial velocity component, with the result that just inside the orifice the stream contracts and does not fill the orifice completely. Therefore, the minimum cross-sectional area of the stream is less than that of the orifice.

By applying the above fundamental equation for velocity of flow through an orifice to the problem of an injection nozzle with a single orifice we find that the orifice diameter required is

$$d = \sqrt{\frac{nQ}{19.16f\theta}\sqrt{\frac{s}{P_1 - P_2}}}\ \text{in.}$$

where d is the diameter of the orifice in in.; n, the engine speed in rpm; Q, the amount of fuel required per cycle at full load, in cu in.; f, the coefficient of discharge of the nozzle orifice; θ, the duration of injection in degrees of crank motion; s, the specific gravity of the fuel; P_1, the injection pressure and P_2, the pressure within the combustion chamber. The equation applies only to orifice-controlled nozzles, that is, nozzles with orifices whose total sectional area is quite small compared with the area of the valve passage.

An equation for the determination of Q from the cylinder displacement and from "experience values" for the mean effective pressure and the specific fuel consumption at full load was given on page 41.

The above equation appears quite simple, but, unfortunately, when we attempt to apply it to the case of an injection valve in an actual engine we meet with great difficulties, for the reason that so many of the factors involved are not constant throughout the period of injection. In fact, the equation is based on the assumption that the velocity of flow is constant. However, a velocity always starts with a value of zero and it is hard to conceive of its coming to an end in any other way than by dropping to zero more or less gradually. Moreover, at the very beginning and the very end of injection through a closed nozzle it is not the area of the fixed orifice but that of the valve passage which mainly controls the flow. P_2 is always variable throughout the injection period, and with pump injection systems P_1 also is variable, since the velocity of the pump plunger varies throughout the stroke. The value of f is normally determined for the condition of constant flow, and it is doubtful whether it is exactly the same when the flow is the result of a sudden impulse, rising rapidly to a maximum and almost immediately decreasing again.

The equation, however, gives serviceable results when applied to constant-pressure injection systems. Relations between orifice diameter, fuel-line pressure, and flow velocity in jerk-pump systems will be discussed in the chapter on Injection Pumps.

Coefficients of Discharge—A great deal of research work on the factors affecting the coefficient of discharge from injector nozzles has been done for the N.A.C.A. at Langley Memorial Laboratory in Virginia by Joachim, Gelalles and others, and the results have been published in Technical Notes and Technical Reports of the N.A.C.A. (obtainable from the Superintendent of Documents, Washington, D. C., at nominal prices) and in the Proceedings of the National Oil Power Conference of the American Society of Mechanical Engineers. The factor which has the greatest influence on the coefficient is the general form of the orifice.

Fig. 8 shows a nozzle plate on which tests were made for the N.A.C.A., the 0.015-in. orifice being modified in various

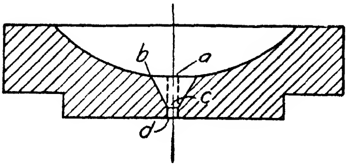


FIG. 8.—NOZZLE PLATE USED TO DETERMINE COEFFICIENTS OF DISCHARGE.

ways for successive tests. For injection pressures of 1000 psi and over the coefficients of discharge for various forms of the orifice were found to be as follows:

Leading edge *a* sharp, 0.67.
Leading edge *a* rounded to about $\frac{1}{64}$-in. radius, 0.81.
Leading edge *a* rounded to $\frac{1}{32}$-in. radius, 0.88.
Leading edge chamfered 60°, corners *b*, *c*, and *d* sharp, 0.85.
Leading edge chamfered 60°, corner *b* slightly rounded, *c* and *d* polished, 0.88.
Leading edge chamfered 60°, corners *b*, *c*, and *d* slightly rounded, 0.94.

For injection pressures of less than 1000 psi the coefficients increased. These values of the coefficient of discharge were obtained when discharging into air at atmospheric density.

Effect of Length/Diameter Ratio—It was found that the coefficient did not change appreciably with the length/diameter ratio for values of this ratio from 1 to 3 inclusive. For larger ratios the coefficient decreased slowly, as shown in Fig. 9, which represents results from a nozzle with a plain stem and 0.008 in. orifice diameter, discharging into air at atmospheric density. With this orifice the coefficient of discharge is the same for injection pressures ranging from 2000 to 6000 psi.

Effect of Reynolds Number—In Fig. 10 results obtained by Bird in England and by the N.A.C.A. staff in this country, with regard to the effect of the Reynolds number on the coefficient of discharge, are plotted. Bird used rather viscous fuels, and his results fall in the region where the change-over from laminar to turbulent flow takes place. He found that with an increase in the Reynolds number there was at first a rapid increase and then a decrease in the coefficient of discharge, the coefficient reaching its maximum value for a Reynolds number of about 2500. While the work of the N.A.C.A. also showed an increase in the coefficient with the Reynolds number, it was more gradual, and the coefficient reached a maximum, constant value only at Reynolds numbers of 4000-6000, depending on the length/diameter ratio.

In making determinations of the coefficient of discharge, use is made of the equation

$$f = \frac{R}{146.5A\sqrt{\frac{P_1 - P_2}{s}}},$$

where R is the rate of flow in cu in. per sec; A, the cross-sectional area of the orifice in sq in.; P_1, the injection pressure and P_2, the chamber pressure, in psi; and s, the specific gravity of the fuel.

High-Speed Spray Photography—Numerous researches on fuel sprays have been made at Langley Memorial Laboratory,

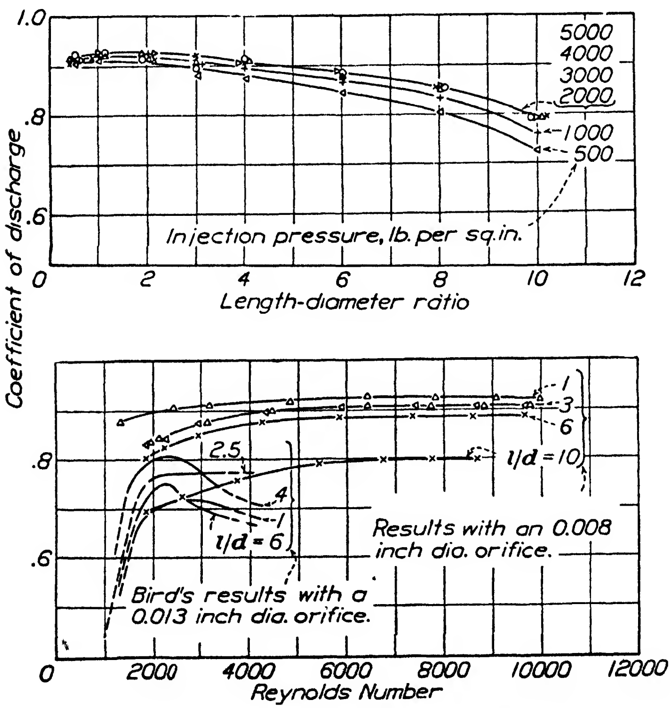


FIG. 9 (*Above*).—RELATION BETWEEN ORIFICE LENGTH/DIAMETER RATIO AND COEFFICIENT OF DISCHARGE.

FIG. 10 (*Below*).—RELATION BETWEEN REYNOLDS NUMBER AND COEFFICIENT OF DISCHARGE.

Langley Field, Va., with a photographic apparatus by means of which 4000 exposures can be made per second. In one series of tests, by Harold E. Miller and Edward G. Beardsley (N.A.C.A. Report No. 222), fuel was sprayed into a chamber with glass walls, which was filled with nitrogen either to

FIG. 11.—PHOTOGRAPHS OF SPRAY DEVELOPMENT.

Receiver Pressure: 300 psi throughout; Injection Pressures: Top, 3000 psi; Center, 5000 psi; Bottom, 8000 psi.

atmospheric pressure or to pressures of 100, 200 and 300 psi. The reason for filling the chamber with nitrogen instead of air was that it was desired to prevent the occurrence of ignition when fuel was injected into it while it was under high gas pressure. Fuel was sprayed through a nozzle with a single round orifice of 0.015 in. diameter. During each injection the injection pressure remained constant, but this pressure was varied between the limits of 2000 and 8000 psi. The injection valve was operated by a cam which, by means of a special clutch, could be given a single rotation at a speed corresponding to 90 rpm.

One series of photographs obtained, with injection pressures of 3000, 5000 and 8000 psi, respectively, against a receiver pressure of 300 psi, is reproduced in Fig. 11, the photos having been retouched before the half-tone plate was made. From these photographs the angle of the spray cone can be measured, and, since the time interval between successive "exposures" is known, the mean velocity of the spray tip during the interval can be calculated. The tip velocity, of

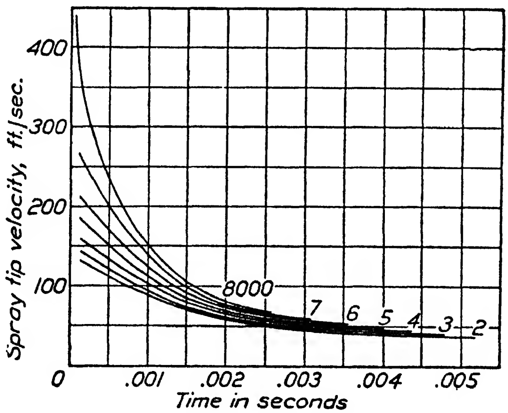


FIG. 12.—EFFECT OF INJECTION PRESSURE ON SPRAY-TIP VELOCITY.
Injection Pressures, 2000-8000 psi; Receiver Pressure, 800 psi.

course, is highest at the beginning, that is, near the orifice; and it decreases more rapidly the higher the pressure in the receiver into which the fuel is injected. Naturally, the tip velocity increases also with the injection pressure.

Fig. 12 shows the influence of the injection pressure on the spray-tip velocity and the decrease of this velocity with time, the fuel being sprayed into nitrogen under 300 psi pressure. Analysis of the experimental results obtained by Miller and Beardsley revealed that the deceleration of the spray tip is approximately proportional to the 1.5th power of the instantaneous velocity of the tip. It was also found that, regardless of the initial velocity of the spray tip, after a period of 0.003 second from the beginning of injection it had been reduced to about 50 fps, but since the initial velocities were much higher with high injection pressures, these produced by far the greatest penetration in 0.003 sec.

Spray Penetration—At the moment the spray begins to form, the tip has a high velocity, but owing to the resistance encountered from the dense air in the receiving chamber, its velocity drops off rapidly. Deceleration of the spray tip is not constant, but varies with the velocity, and even more rapidly than the latter, as stated in the preceding paragraph. This means that while the tip velocity decreases rapidly at first, its rate of decrease diminishes, with the result that the tip still retains a certain velocity and the jet still keeps on growing a comparatively long time after it was started. In

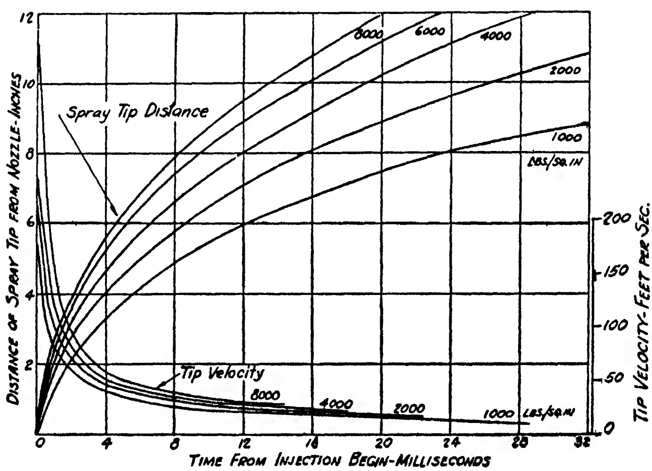


FIG. 13.—INCREASE OF SPRAY-TIP DISTANCE FROM NOZZLE AND VARIATION OF TIP VELOCITY WITH TIME.

normal engine operation the spray is never fully developed, simply because the injection pressure is not maintained long enough. The problem therefore is not to find to what depth the jet would penetrate if it continued indefinitely, under the given conditions of injection pressure, chamber pressure, coefficient of discharge, Reynolds number, etc., but rather to find the penetration during the injection period.

Effect of Injection Pressure on Tip Velocity and Penetration—The effect of the pressure behind the fuel at the nozzle on the velocity of the tip and on the penetration of the jet are clearly brought out by Fig. 13, which is reproduced from a paper by P. H. Schweitzer read before the National Oil Power Conference in 1934.* The results were obtained experimentally from a nozzle with an orifice of 0.0135 in. diameter and 0.027 in. length, injecting into cold air under a pressure of 200 psi (about the same density as obtained in an engine cylinder with a compression ratio of 15). Injection was effected at various pressures from 1000 to 8000 psi. The tip-velocity curves are in fair agreement with the similar curves of Fig. 12. Fig. 13 confirms the statement made in the foregoing that the spray tip, if the pressure behind it were

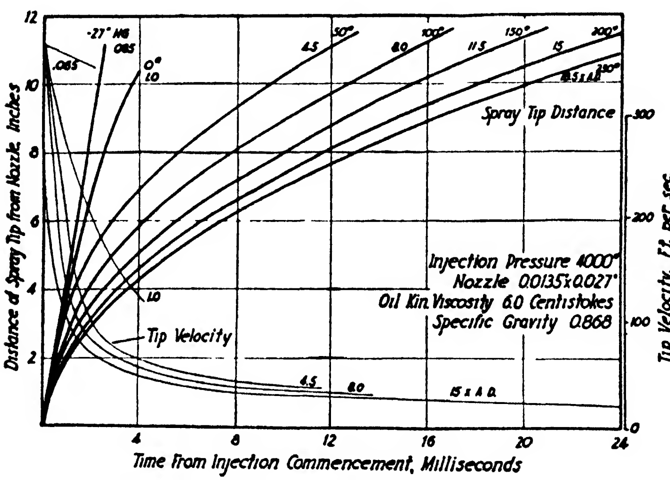


FIG. 14—EFFECT OF AIR DENSITY ON SPRAY PENETRATION.

continued, would keep on advancing after a period several times as long as the normal injection period.

Effect of Chamber Pressure—Fig. 14, which also is from Dr. Schweitzer's paper, shows the effect of chamber pressure, or, rather, chamber air density, on tip velocity and penetration. The injection pressure in this case was 4000 psi. It is interesting to note that to reach a penetration of 10 in. takes nine times as long when injecting into air at 18.5 times atmospheric density as when injecting into air at atmospheric density. In engine operation, of course, the chamber density is limited rather closely by the required compression ratio,

* Proceedings of this and previous Oil Power Conferences have been published by Pennsylvania State College and are obtainable from it.

and penetration therefore must be controlled mainly by the injection pressure and the orifice diameter.

Effect of Length/Diameter Ratio on Penetration—From the equation for nozzle velocity given on page 114 it will be seen that this velocity is dependent on the coefficient of discharge, which latter in turn has been shown to vary slightly with the length/diameter ratio of the orifice. It follows from this that the penetration of the spray tip in a given time must vary with the length/diameter ratio. The results charted in Figs. 13 and 14 were obtained with the same length/diameter ratio of 2. The effect of variations in the ratio on penetration was studied by the N.A.C.A., and some of its results are plotted in Fig. 15, which shows the penetration during one one-thousandth of a second. The receiver pressure was 200 psi.

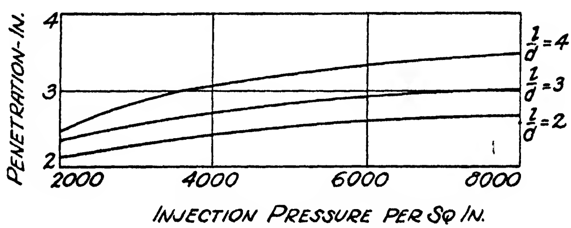


FIG. 15.—SPRAY PENETRATION DURING 0.001 SEC., WITH A CHAMBER PRESSURE OF 200 PSI.

That the length/diameter ratio of the orifice has a marked effect on the penetration is evidently due, primarily, to the fact that with increasing proportional length of orifice the nozzle gives a more nearly laminar flow, which results in a spray cone of smaller angle and consequently reduced air resistance.

Real Velocity of Oil Droplets—Spray velocities at different distances from the nozzle, in air at atmospheric and 15 times atmospheric density, as determined by Dr. Schweitzer, are plotted in Figs. 16 and 17 respectively. He called attention to the fact that the tip velocity of the spray must not be taken as representing the velocity of the individual oil droplets. The velocity of the oil when it leaves the nozzle is usually several times as large as the velocity of the spray tip. The reason is that the spray-tip velocity merely represents the rate at which the "hole in the air" grows. At the beginning this is much slower than the velocity of the individual droplets, because the leading droplets are quickly stopped by the air and are overtaken by others pushing up from behind. Determinations with a ballistic pendulum show the initial velocity of the spray under 4000 psi pressure into air at 15 times atmospheric density to be about 835 fps (see Fig. 17), while the initial velocity of the spray tip under similar conditions (see Fig. 13) is only 230 fps.

The effect of the orifice diameter is rather pronounced. If the diameter is increased from 0.007 in. to 0.025 in. the penetration is approximately doubled.

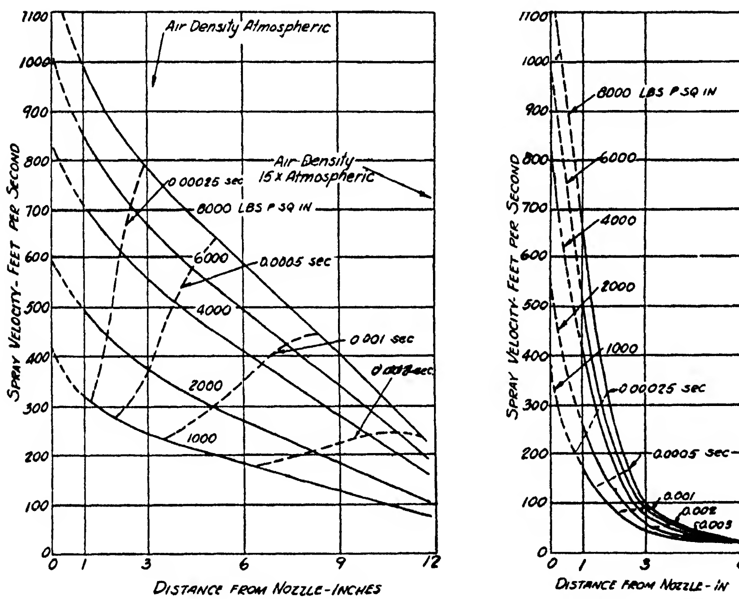


FIG. 16 (*Left*).—VARIATION OF SPRAY VELOCITY WITH DISTANCE FROM NOZZLE IN AIR AT ATMOSPHERIC DENSITY.

FIG. 17 (*Right*).—VARIATION OF SPRAY VELOCITY WITH DISTANCE FROM NOZZLE IN AIR AT 15 TIMES ATMOSPHERIC DENSITY.

In Dr. Schweitzer's experiments the amount of fuel in the spray which reached different distances from the nozzle was measured by means of a tipping cup. This cup is so placed in the pressure chamber that its opening faces the spray and is concentric therewith. The cup is pivotally mounted on cup-and-cone bearings and is held in an upright position by a counterweight. It catches the spray, and after a certain amount has been collected, its moment exceeds that of the counterweight, and it tips, thereby opening an electric circuit.

By counting the number of injections necessary to tip the assembly, the amount of oil which gets as far from the nozzle as the opening of the cup can be determined.

Size of Spray Droplets—In a general way, the higher the pressure on the fuel at the spray nozzle, the more effective the atomization, that is, the smaller the individual droplets. It also seems reasonable to assume that the smaller the droplets, the shorter the time in which the fuel can be burned. If the droplets are too large, they will be burned incompletely, the exhaust will become smoky, and the specific fuel consumption will increase.

Several different methods of determining the mean diameter of the droplets in oil sprays under various conditions have been worked out. The Woeltjen method consists in throwing the spray into an absorption liquid in which it remains unchanged for a considerable time. The absorption liquid used consists of 70 per cent distilled water and 30 per cent of a special tanning fluid.

In order to simulate the conditions encountered in engine operation as closely as possible, the fuel is injected into a steel bomb which is partly filled with the absorption fluid mentioned and which is pumped up to a gas pressure equal to that in the combustion chamber of an engine. Thus when the fuel is injected into the bomb, it must pass through the highly compressed gas before it is taken up by the absorption liquid, and any influence of gas resistance on the atomization is taken account of. The result of the injection is an emulsion of oil fuel and absorption liquid which can be studied under the microscope. The diameters of individual droplets can be ascertained by means of a measuring ocular, and the relation between droplets of different sizes can be determined by means of the statistical method.

The Woeltjen method has been applied in this country by H. F. Bryan of the International Harvester Company, who used the following technique: A receiving vessel containing 300 ml of a gum solution serving as the absorption liquid is placed ¾ in. from the discharging nozzle. Approximately ten "shots" are allowed to enter the liquid, the number being varied according to operating conditions, as suggested by experience. Three different solutions were used as absorption liquid. The first consists of 5 g gum tragacanth powder in 1000 ml of distilled water. Small portions of the gum tragacanth are ground in water in a mortar to a thick, smooth paste. The paste is then added to the remainder of the distilled water and the whole is thoroughly shaken. Another solution consists of 1 g of white soap (Swift's Amazone Floating), 100 ml ethyl alcohol, and 400 ml distilled water. The third is made up of 285 ml of the first and 15 ml of the second solution, the two being combined shortly before the experiment is to be made. A small quantity of the emulsion formed when fuel is sprayed into the absorption liquid is placed in a microscope slide cell. This is photographed and the drop count is made on a print from the negative.

Other experimenters have caught the spray on smoked glass plates. With this method, photomicrographs are taken of representative areas, and the impressions in the lampblack in these areas are classified according to diameter range, and those in each diameter range counted. Those within the diameter range 0.00025-0.00075 are considered as one group

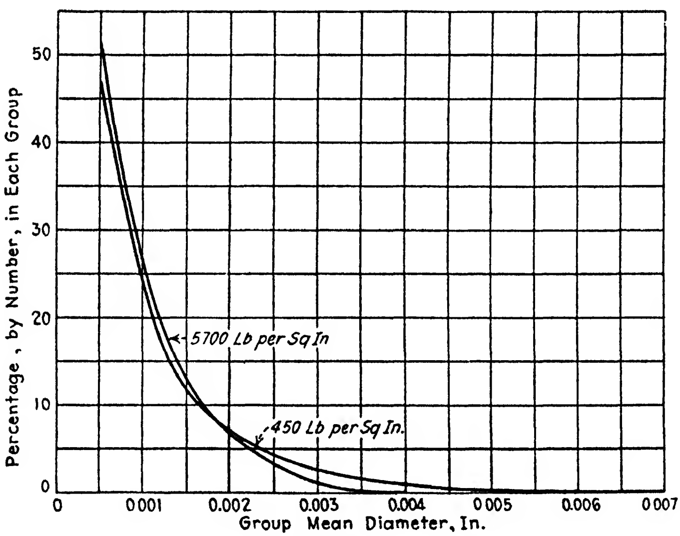


FIG. 18.—DROPLET-SIZE FREQUENCY CURVES.

of mean diameter 0.0005 in., those within the diameter range 0.00075-0.00125 in. as a second group of mean diameter 0.0010 in., and so on. The numbers of droplets counted in the different groups are plotted as ordinates against the group mean diameters as abscissas, and a smooth curve is drawn through the points so obtained.

Droplet-Size Frequency—Such curves—known as frequency curves—show that the smallest droplets are present in the largest numbers. But these small droplets represent only a small proportion of the total amount of fuel in the spray, and a better picture of the degree of atomization may be obtained by plotting the percentage of the total volume represented by the droplets in any mean-diameter group against the group mean diameter. Ordinates for such a curve can be readily found by multiplying the ordinates of the frequency curves by the cubes of the corresponding abscissas (mean diameters of droplets).

A frequency curve and a volume distribution curve, both based on the work of Dana W. Lee at the N.A.C.A. Laboratory, are shown in Figs. 18 and 19. It may be seen from Fig. 19 that with low injection pressures the diameters of the droplets cover a much wider range; in other words, atomization of the fuel is less uniform.

Fig. 20 shows some photomicrographs of spray droplets prepared by the Woeltjen method at Pennsylvania State College by Lieut. Jos. P. Thew. The pressures of the air into which the oil was sprayed are noted on the photographs. The oil was sprayed under 4000 psi pressure through an orifice 0.0135 in. in diameter by 0.027 in. long.

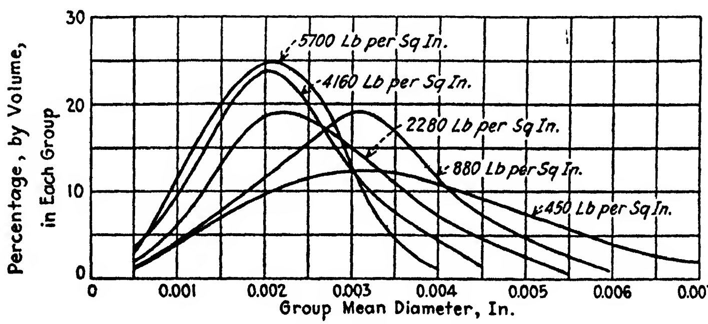


FIG. 19.—QUANTITY DISTRIBUTION ACCORDING TO DROPLET SIZE.

Among the points learned from atomization studies are that the injection pressure has a rather important effect on the mean droplet diameter, and that the chamber pressure has less influence thereon. Sass' experiments showed practically no difference in mean droplet diameter for injection into air at atmospheric pressure and at five times atmospheric pressure respectively, but when the chamber pressure was increased from five to ten atmospheres, the mean droplet diameter (for about 4000 psi injection pressure and an orifice diameter of 0.023 in.) decreased from 0.00047 to 0.00038 in.

Effects of Orifice Dimensions—The mean droplet diameter is dependent to a slight extent on the diameter of the orifice, and N A C.A. tests showed that with an injection pressure of 4000 psi it decreases with the orifice diameter for values of the latter below 0 015 in. Experiments made at the Arcueil Laboratory of the French Air Ministry indicate that for given values of the injection pressure, chamber pressure and orifice diameter, the fuel is most finely atomized when the length/diameter ratio of the orifice is between 4 and 5. With orifices having a length only twice the diameter, the

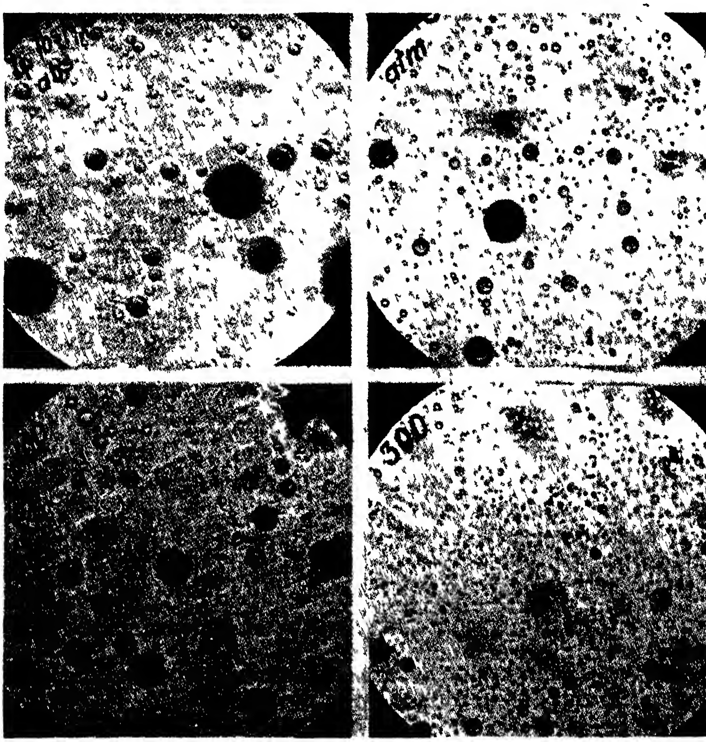


FIG 20—SPRAY PHOTOGRAPHS OBTAINED AT PENN STATE COLLEGE.
(The figures on the photographs represent the chamber pressure in psi)

mean droplet diameter is very considerably greater than with orifices having a length/diameter ratio of 4 or slightly more.

The higher the viscosity of the sprayed fuel, the greater the mean diameter of the droplets in the spray.

Fuel Dispersion—Fuel injected into the combustion chamber must not only be finely atomized, but its particles must be dispersed throughout the air charge as nearly uniformly as possible. Fine atomization and the highest degree of dispersion are necessary particularly in the case of engines with combustion chambers of a quiescent type.

Tests made at Pennsylvania State College (and described in Engineering Experiment Station Bulletin No. 40) showed that the dispersion of the oil droplets becomes more nearly uniform as the injection pressure and the air density in the chamber increase and as the oil viscosity decreases. The cone angle increases with the injection pressure and the chamber density, and Professor Schweitzer has pointed out in this connection that with an injection pressure of 4000 psi, if the injection takes place into air at atmospheric density, 57 per cent of the total discharge will still be confined within a circle of 1-in. diameter at a distance of 12 in. from the nozzle, whereas if the discharge takes place into air at a density corresponding to 200 psi pressure, the percentage will be only 8.7. It was also shown that the percentage of the sprayed oil which reaches a certain distance from the nozzle increases as the cone angle decreases. With very viscous oils the cone angles were much smaller and the dispersion was poor.

Differential Action of Valves—With hydraulically-actuated valves, up to the time the valve leaves the seat the fuel pressure acts on an annular surface whose outer diameter is equal to the diameter of the valve stem and whose inner diameter is equal to the outer diameter of the valve seat. As soon as the valve is off its seat, the pressure acts on a circular surface of a diameter equal to the diameter of the valve stem. With an increase of the area on which the pressure acts and a simultaneous increase in pressure due to the increase of the speed of the pump plunger, the valve is lifted very rapidly. Valves of this type open with a popping sound. In most designs the lift is limited by a stop.

As the fuel pressure acts on a larger area when the valve is open, it follows that the pressure must drop to a value materially lower than the opening pressure before the valve can close again. This is referred to as the differential action of the valve.

Valve-Opening and -Closing Pressures—The fuel pressures at which the valve opens and closes have an influence on the spray characteristics. One effect of an increase in the valve-opening pressure is to increase the injection lag. With an increase in the opening pressure the injection pressure is raised not only at the beginning, but throughout the whole period of injection. In fact, the characteristics of the injection system can be materially changed by a change in the valve-opening pressure, and means for changing the opening pressure are generally provided.

A high closing pressure tends to prevent objectionable secondary discharges. If the difference between the opening and closing pressures is small, there is greater assurance that the valve will be opened by the first pressure wave striking it, which is desirable from the standpoint of smooth operation. The difference between the opening and closing pressures depends, of course, on the ratio of the diameter of the valve seat to the diameter of the valve stem.

Minimum-Injection Limit—With hydraulically-operated, inwardly-opening injection valves the smallest quantity of fuel that can be injected per cycle is determined by the difference between the fuel pressure required to open the valve and the fuel pressure which just allows the valve to close, as well as by the elasticity of the fuel and the fuel line. As explained in the preceding paragraph, a larger area of the valve is exposed to the fuel pressure when the valve is off its seat than when it is closed. As the spring pressure on the valve is substantially the same whether the latter is closed or just off its seat, it follows that the unit pressure of the fuel must be greater when the valve is being opened than when it is merely being held open or is closing.

Let us call the fuel pressure which just allows the valve to close, P_1, and the fuel pressure required to open the valve, P_2. Also, let the normal capacity of the fuel line be V and the modulus of elasticity of the fuel and line, E (equal to about 280,000 psi for ordinary fuel oil). At the time the pump begins to deliver fuel the pressure in the line is P_1, and fuel will not leave the line until the pressure therein has reached the value P_2 and the injection valve begins to lift. While the pressure in the line rises, the fuel therein compresses and the tube expands, and the quantity of fuel required to make up for this is

$$Q = \frac{V}{E}(P_2 - P_1) \text{ or } 0.0000035V(P_2 - P_1).$$

If the pump delivers less than this quantity into the line, the injection valve is not lifted at all; but if the quantity delivered to the line slightly exceeds this amount, the valve lifts, and as much fuel is injected into the engine as is delivered to the line. Hence the minimum quantity of fuel which can be injected per cycle in an engine with inwardly-opening, hydraulically-operated valves is $0.0000035V(P_2 - P_1)$, where V is the normal capacity of the line in cu in. and $P_2 - P_1$ is the difference between the hydraulic pressures required to open the valve and to just allow it to close, in psi.

Under certain conditions, owing to the presence of pressure waves in the line, it is possible to inject still smaller fuel quantities than the foregoing equation would indicate. Under such conditions it may also occur that fuel will be injected every second or every third cycle, no injection taking place during the intermediate cycles.

American Bosch Injector—An axial section of one type of American Bosch injector is shown in Fig. 21, where the names

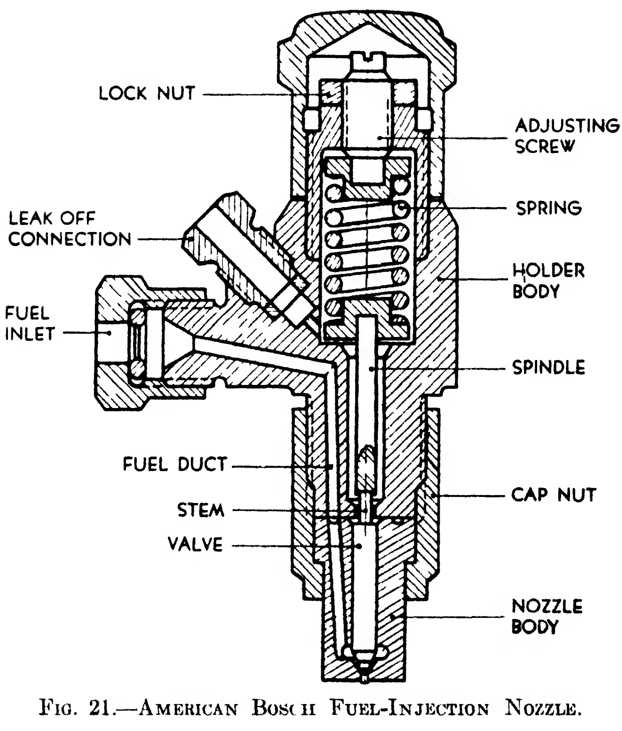


FIG. 21.—AMERICAN BOSCH FUEL-INJECTION NOZZLE.

of the different parts are lettered in. The nozzle proper consists of two parts—the nozzle body and the nozzle valve. These parts are made of alloy steel and heat-treated. Body and valve of all nozzles are lapped to a very close fit, and the two parts cannot be interchanged individually. At its upper end the nozzle body is provided with a flange, and it is secured to the nozzle holder by a cap nut. Fuel enters the nozzle holder through a fitting shown at the left, and passes through a drilled duct to a chamber directly above the valve seat. Any fuel leaking past the valve into the nozzle holder can return to the injection pump through a leak-off tube, for which a fitting is provided on the nozzle holder. Abutting faces of the nozzle body and nozzle holder are accurately ground, and in reassembling these parts, great care must be taken that they are absolutely clean. The fuel pressure at which the valve opens can be varied by means of the adjusting screw at the top, which is rendered accessible by removing the cap nut.

American Bosch injection nozzles are made in three basic types, viz., pintle nozzles, throttling nozzles, and hole nozzles. In the hole-type nozzle there are one or more spray orifices in the form of straight, round holes through the tip of the nozzle body below the valve seat. The spray from the individual orifice is relatively dense, and the general spray pattern, which may or may not be symmetrical, is determined by the number and arrangement of the holes. Orifices can be made of as small a diameter as 0.004 in., and in large nozzles there may be as many as 18 orifices. Hole-type nozzles are used mostly in engines with undivided combustion chamber.

Scintilla Injector—Fig. 22 shows a Scintilla injection nozzle. The nozzle tip and the nozzle-valve assembly are held to the holder body by an assembly nut. The nozzle valve assembly includes a stop plate. Spring pressure is transmitted to the valve by a long pressure pin. The pressure on the pin can be adjusted by means of an adjusting screw which, after adjustment, is locked in position by a lock nut. Other parts of the unit include an inlet nipple, a filter, a leakage-return fitting, and a bleeder valve.

Ruston & Hornsby Injector—In this injection device (Fig. 23) the nozzle body is clamped between the nozzle holder and a cap, in accordance with a very common practice, this facilitating the machining operations on the nozzle body. The fuel passage down the nozzle holder is of annular form, between the outer wall of the valve guide and the bore of the nozzle holder. Fuel enters the nozzle holder through the fitting at the side, and any leakage drains off through the fitting at the top. The method of holding the nozzle holder in place in the cylinder head by means of a clamp plate and studs is clearly shown.

Hints on Injector-Valve Design—To ensure oil-tightness, injection valves must be so designed that the unit pressure on the seat is quite high. On the other hand, the pressure must not be too high, as that tends to injure the seat by chipping, etc. The N.A.C.A. has recommended that the widths of valve seats be calculated on the basis of 15,000 psi. One authority recommends making the seats on the valve and in the valve body of the same width, so that no ridge can form on the wider seat, but in general the seat on the movable part is made somewhat wider.

For proper seating it is essential that the valve be accu-

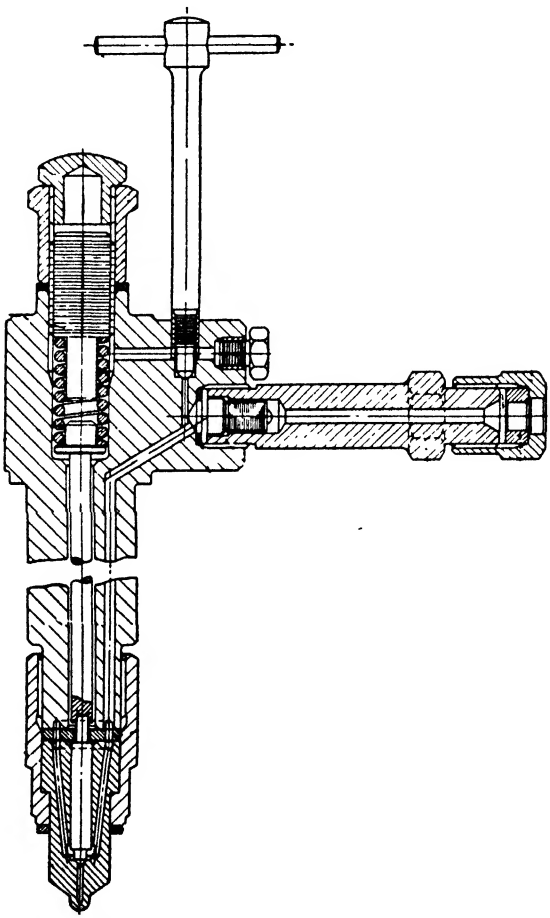


FIG. 22.—SCINTILLA INJECTION NOZZLE.

rately guided. Packing is impractical, the same as in pumps, and the stem and guide therefore must be made of considerable length, equal to at least 5 or 6 diameters, and the stem accurately lapped in. It has been pointed out already that rounding of the up-stream edge of the spray orifice considerably improves the coefficient of discharge. If the edge were left sharp in the manufacturing process, it would soon be worn away by abrasive material in the fuel, and this would alter the characteristics of the nozzle. It is therefore advisable to always round the up-stream edge. Pointers on this subject are given in the publications of the N.A.C.A. It is advisable not to bring the valve spring too close to the nozzle tip, as that might result in its temper being drawn. The

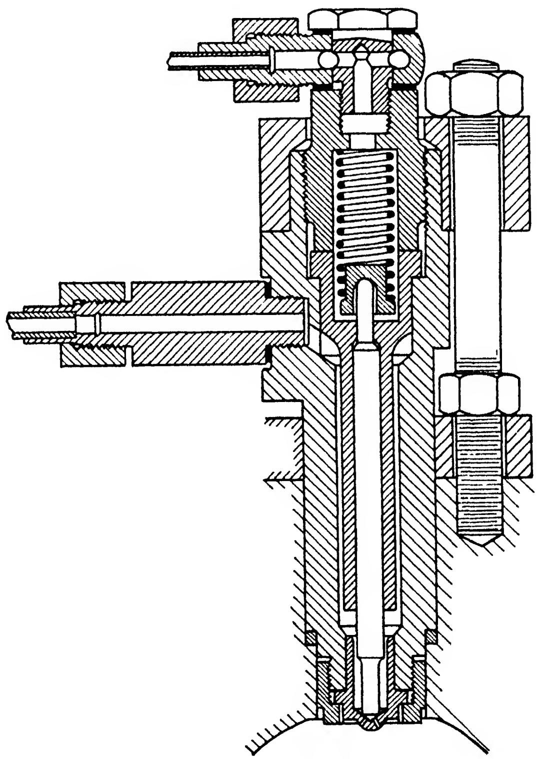


FIG. 23.—RUSTON INJECTION NOZZLE.

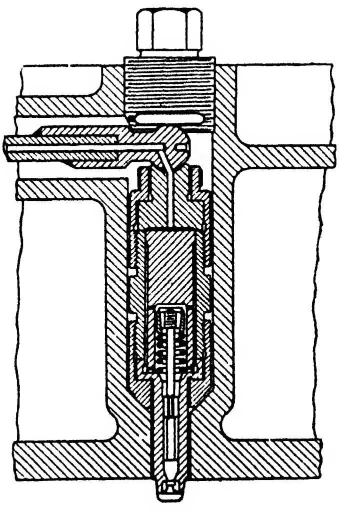
nozzle fittings must be made quite substantial, so that they may be tightened without fear of injuring them. In some designs a vent screw is provided at the highest portion of the injection valve, so that any air getting into the injector may be allowed to escape, and interference with the proper operation of the valve prevented.

Injector Mounting—Injectors should be secured to the cylinder or cylinder head in such a manner that they can be easily removed for inspection and tests. A familiar method of mounting is by a clamping plate or yoke with a circular hole through it, which is passed over the outer end of the injector body and rests against a shoulder thereon. This plate has the general form of a two-bolt pipe flange and is drawn against the shoulder by means of two studs and nuts.

Quite a unique method of injector mounting has been worked out by the Swiss firm of Saurer. As shown in Fig. 24, the injector is completely submerged in a long drilled boss in the cylinder head. The fuel pipe connecting to it is provided with a brazed-on fitting with a spherical head, which is clamped between the top of the injector and a set screw screwing into the head casting. Attention may be called to the fact that the shank of the fitting is cut off at an angle of 45°, which is intended to reduce concentration of stress in the tube where it enters the fitting. This type of mounting, of course, makes for a very "clean" exterior of the engine. The injector is of the type having an outwardly-opening valve, and it needs no leak-off tube. It gives what is called an umbrella-shaped spray.

Nozzle-Operating Temperatures—A good deal of the trouble experienced with injection nozzles is due to the high temperatures attained by them in severe service. The nozzle tips reach a temperature of between 600 F and 650 F, which is approximately the end temperature (end point) of the fuel. Unless injection terminates with a very sharp cut-off, a small amount of dribbling will take place at the end of the period. The rapidity of the cut-off depends on a number of factors of the injection system, including the rapidity of opening of the pump relief valve. The surface of the nozzle tip is then covered with liquid fuel. Owing to the high temperature of the tip, this fuel evaporates, leaving a tarry deposit adhering to the nozzle. At still higher temperatures, some of the fuel on the tip will be "cracked," in which case gas carbon combines with the tarry deposits. Frequently this deposition of carbon and tarry material continues until a regular "crater" forms on the tip, which tends to deform the spray cone and may even choke the orifice. It is therefore advisable to so mount the injector that it will be effectively cooled by the cylinder cooling system. Alternately, internal cooling by means of a liquid circulating through cooling passages in the nozzle may be resorted to.

Nozzle Protector—In some injector designs the nozzle is held to the injector body by means of a gland nut, and a certain clearance is allowed between the head portion of this nut and the nozzle. If the nozzle were made a close fit in the nut, heat distortion of the latter might cause the injector valve to stick. If the clearance between nozzle and gland nut is of any importance, it further adds to the maximum temperature reached by the nozzle tip in service, because this clearance space will fill up with burning gases during each cycle. To eliminate this heating effect, the German Daimler Company places a heavy washer of soft copper underneath the injector, as shown in Fig. 25. The nozzle is so designed that its tip projects about 0.004 in. from the gland nut. When the mounting nuts of the injector are screwed down, they deform the copper washer plastically until the gland nut seats on it, and the clearance space between nozzle and gland nut is then



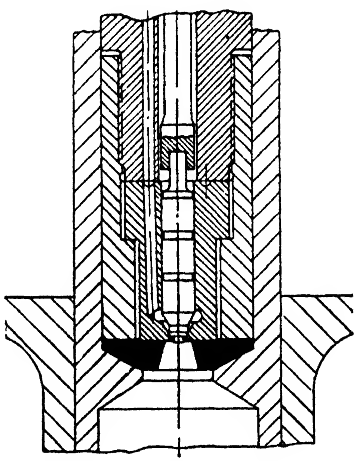
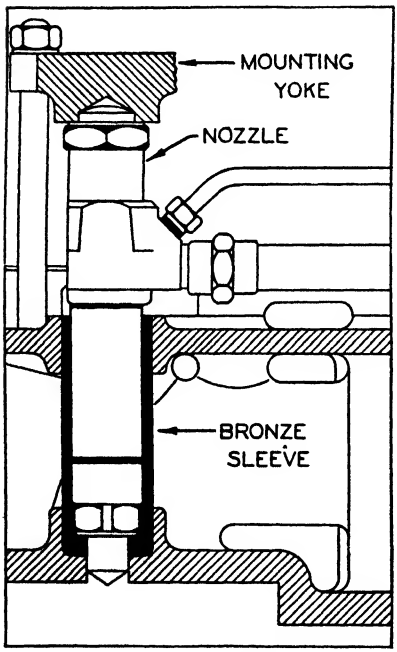


FIG. 24 (*Left*).—SAURER INJECTION NOZZLE AND MOUNTING.
FIG. 25 (*Right*).—DAIMLER-BENZ NOZZLE PROTECTOR.

completely shut off from the combustion chamber. This "nozzle protector" is said to greatly increase the life of nozzles.

Injector Cooling—In engines in which the injector is mounted in the cylinder head it is usual to provide a drilled boss which is completely surrounded by water. Owing to the presence of inlet and exhaust passages in the head, the water jacket there is always quite deep, and a considerable length of injector body can be brought into close proximity to the jacket water. In other engine types, where the injector has to pass through the cylinder wall, there is less depth of water jacket, and the cooling effect on the injector then can be increased by drilling through the jacket and combustion-chamber walls and expanding a bronze or copper sleeve in the drill holes, which the lower part of the injector fits with only little clearance. The walls of such a sleeve can be made thinner than the walls of a drilled mounting boss cast in the cylinder block and, besides, copper and bronze have higher heat conductivities than cast iron. One method of mounting the injector in a non-ferrous sleeve (in this case in the cylinder head) is illustrated in Fig. 26. Here the sleeve has an inward flange at its lower end, which serves as a gasket for the injector.



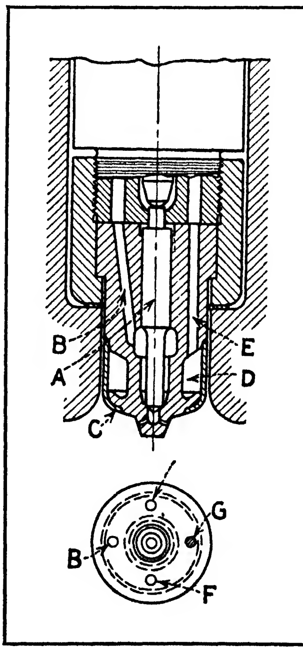


FIG. 26 (*Left*).—MOUNTING INJECTOR IN BRONZE SLEEVE.
FIG. 27 (*Right*).—SULZER BROTHERS LIQUID-COOLED NOZZLE.

Nozzle Cooling by Fuel Circulation—The most practical way to cool the nozzle effectively seems to be to circulate fuel through chambers or passages close to the nozzle tip. In the pump-injector units of the General Motors two-stroke Diesel engine, a cooling effect is produced by surrounding the pump barrel with an annular chamber through which fuel is circulated continuously by the transfer pump. The injector is located immediately below the pump barrel, and while the circulating fuel does not come close to the nozzle tip, the heat conductivity of the nozzle seems to ensure an adequate heat flow from the tip to the fuel chamber above the nozzle to prevent excessive heating of the tip.

Quite a number of liquid-cooled injection nozzles have been offered to engine manufacturers in recent years. In a design due to the Swiss firm of Sulzer Brothers (Fig. 27), the lower end of the nozzle has a circular recess *D* cut in it and is provided with a welded-on jacket *C*. A cooling liquid enters the jacket space through duct *E* and returns through the oppositely-located duct *F*. The intermediate duct *B* serves to convey fuel to the chamber surrounding the lower end of nozzle valve *A*, while *G* is a dowel pin provided to ensure proper registry of the ducts in adjoining parts.

The two views shown in Fig. 28 are of a Bosch liquid-cooled nozzle and are axial sections taken at right angles to each other. A cooling jacket is formed between the nozzle tip and a sleeve with internal and external flanges at opposite ends, which is set into the nozzle cap nut. The coolant flows toward and away from the jacket through oppositely-located coolant ducts *C*, which terminate in tapped holes at the upper end of the injector, opposite from the fuel-inlet and leak-off connections. In the left-hand view, *F* indicates the duct through which fuel is supplied to the nozzle. These liquid-cooled injectors are used mainly with engines having large-sized cylinders.

Open Nozzles—Open nozzles have been used principally in Germany, by such firms as M.A.N., Krupp and Junkers. The Packard Diesel aircraft engine, which was developed in this country, had nozzles of this type, and they are also used on the Hesselman spark-ignition engine formerly manufactured by the Waukesha Motor Co. The open nozzle has the advantage of simplicity, as it consists essentially of a plug fastened into the cylinder wall, through the axis of which there is a very fine hole. Another advantage that has been claimed for it is that any air entering the fuel line can readily pass out of the nozzle, whereas with a closed nozzle the air, owing to its compressibility, may prevent opening of the valve. Furthermore, the open nozzle is free from all trouble due to a leaky valve.

The chief disadvantage of the open nozzle is that the pressure producing the spray will be comparatively low at the beginning and end of injection, as a result of which atomization will not be so thorough during these parts of the injection period. This means comparatively large droplets in the spray, which tend to result in poor combustion and a smoky exhaust. At low engine speeds atomization is poor throughout the injection period, but owing to the greater time then available for combustion, it is not so essential that the fuel charge be introduced into the combustion chamber in a state of very fine division.

A sectional view of an open nozzle used in an early truck-type Diesel engine of the Maschinenfabrik Augsburg-Nürn-

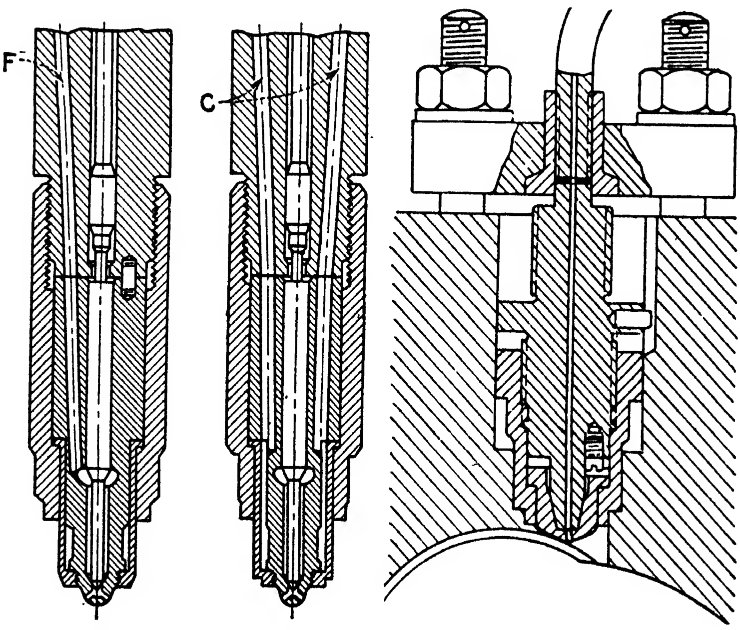


FIG. 28 (*Left and Center*).—BOSCH LIQUID-COOLED NOZZLE.
FIG. 29 (*Right*).—M.A.N. OPEN-NOZZLE ATOMIZER.

berg is shown in Fig. 29. Either one or two nozzles were used per cylinder, and were secured into the cylinder-head casting at the side, a flange around the head of the piston being cut away where the nozzles were located. The arrangement of the air inlet port and valve was such as to produce a swirling flow of air in the combustion chamber, and this was depended on in part to assure proper distribution of the fuel through the air charge.

CHAPTER VI

Injection Pumps, Their Principles and Characteristics

Practically all modern high-speed Diesel engines make use of the jerk-pump system of fuel injection. As a rule, there is one pump or one pump element for each engine cylinder, though in some cases a single pump injects fuel into all of the cylinders, being placed in communication with the different cylinders in turn by a distributor. Where a separate pump element is used for each cylinder, the different elements, of course, can be combined in a single unit and provided with a single drive, a single control linkage, and a single connection to the fuel tank or the transfer pump. Each of these pump elements, in addition to forcing fuel into the cylinder under high pressure, must accurately meter the fuel charge and time its delivery. Plunger pumps are used exclusively for this purpose.

In addition to the jerk-pump there are two other systems of fuel injection, the common-rail and the spring-pressure system. These, however, are rarely met with at present, and they are of interest chiefly from the historical standpoint. In the common-rail system fuel is maintained at a substantially constant pressure somewhere between 2000 and 8000 psi in a manifold which connects to injection valves in the individual cylinders. The fuel is put under pressure in this manifold (and in an "accumulator" in the form of a steel bottle connected to it) by a pump, and the pressure is kept uniform by means of a spring-loaded by-pass valve. In this system, which was first used by the British firm of Vickers Ltd. for submarine engines, metering of the fuel is accomplished by means of the injection valves. The valves of the different cylinders are opened mechanically in the proper sequence, and are held open for a given length of time relative to the time of one engine cycle. The amount of fuel injected per cycle by this system evidently depends upon the pressure on the fuel in the manifold, the number, diameter and form of the injector orifices, and the time the injector valve remains open, as well as upon some other factors, including the viscosity of the fuel if the injector orifice is quite long relative to its diameter. The common-rail system has been widely used on large marine engines, but it is poorly adapted to engines which must operate satisfactorily over a wide speed range, and it has not come into use in this field. Among other reasons why it is not suited to automotive-type engines are that it is rather complicated and that if the engine is overloaded and pulled down to a low speed, there is no definite limit to the fuel quantity injected. The engine then operates with a smoky exhaust and fouls up internally.

With the spring-pressure system a plunger is moved positively by means of a cam during the suction stroke of the pump and is returned by a spring—which has been compressed during the suction stroke—during the delivery or injection stroke. It is obvious that the speed of the plunger during the delivery stroke depends only on the pressure of the spring and the inertia of the masses moved by it, and it is therefore independent of engine speed. The amount of fuel injected does not vary with the engine speed, as with the common rail system, but the duration of injection in terms of crank angle varies greatly, which is almost equally objectionable.

Injection pumps operated by spring pressure have seen only limited use. An advantage of both the common-rail and the spring-pressure systems is that the pressure behind the jet, the initial velocity of the jet, and the fineness of atomization are not influenced by engine speed.

Ratio of Piston Displacement to Fuel-Charge Volume—The degree of accuracy with which the injection pumps of small, high-speed oil engines must function may be judged from the small quantities of fuel required by them per cycle. Commercial fuel oils need about 10,000 times their volume of atmospheric air for their complete combustion, theoretically, but these engines usually work with 50 per cent excess air even at full load, hence the volumetric ratio of air to fuel is about 15,000 to 1. In regular operation the engine cylinder, of course, is not completely filled with air at atmospheric pressure during the inlet stroke; assuming a volumetric efficiency of the engine of 75 per cent, the volume of fuel delivered by the pump would have to be 1/20,000 of the displacement of the engine cylinder.

Approximately this same proportion between engine-piston displacement and maximum volume of fuel required per cycle is arrived at by a calculation based on performance data of actual engines. An average high-speed Diesel engine will show a brake mean effective pressure of 80 psi and a specific fuel consumption of 0.45 lb per brake horsepower-hour. Assuming the mechanical efficiency of the engine under these conditions to be 80 per cent, the indicated mean effective pressure is 100 psi and the specific consumption per indicated hp-hr, 0.36 lb. Since one horsepower is equal to 33,000 foot-pounds per minute, with an indicated mep of 100 psi one indicated hp-hr corresponds to a piston displacement of

$$\frac{33{,}000 \times 12 \times 60}{100} = 237{,}600 \text{ cu in. per hr.}$$

One cu in. of Diesel fuel of 0.84 specific gravity weighs 0.030 lb, and the specific consumption of 0.36 lb therefore is equivalent to

$$0.36/0.030 = 12 \text{ cu in. per ihp-hr.}$$

Thus the ratio of engine-piston displacement to volume of fuel injected at full load is about

$$237{,}600/12 = 19{,}800$$

In the above calculation the assumed values are such as to give a rather large figure for the volumetric ratio between air and fuel charges. That ratio increases in direct proportion to the specific gravity ρ of the fuel burned and varies inversely as the mechanical efficiency η, the indicated mean effective pressure p, and the brake specific fuel consumption f. It is given by the equation

$$r = \frac{844{,}000\rho}{\eta p f}.$$

If we assume that the engine operates on lighter fuel, of 0.80 specific gravity, and that at maximum output the mechanical efficiency is 80 per cent, the indicated mep, 110 psi, and the brake specific fuel consumption, 0.50 lb per bhp-hr, the ratio comes out to 15,340. In practice engines usually are provided with pumps having a delivery between 1/15,000 and 1/18,000 the displacement of the engine piston. Pump deliveries usually are given in cu mm. One cu in. = 16,380 cu mm.

Injection-Pump Displacement—It must not be assumed that the figures given by the above equation represent the ratios of engine-piston to pump-plunger displacements. Only a small fraction of the plunger stroke can be used for injection, because for efficient combustion the fuel must be injected during the time corresponding to a small crank angle, and in a four-stroke engine a small crank angle corresponds to an even smaller pump-shaft angle. Besides, the first and last portions of the plunger stroke, which are necessarily at low speed, are not well suited to injection, as the injection pressure and the fuel atomization would be inadequate. In most designs of injection pump only one-fourth of the stroke or less is utilized. Assuming the ratio between engine displacement and pump delivery to be 16,000 to 1, and that one-quarter of the pump delivery stroke is effective, the total displacement of the pump plunger must be 1/4000 that of the engine piston. When the engine is idling, its fuel charge will be only about one-fifth as large, and the relation between the displacement of the engine piston and the volume of the fuel charge required then will be of the order of 80,000 to 1. Thus a 4¾ x 6-in. cylinder, having a displacement of 106 cu in., requires a maximum fuel charge of 0.0066 cu in. (about 110 cu mm), and an idling charge of 0.0013 cu in.

In addition to measuring such small charges of fuel accurately, the pump must work against pressures which in the case of engines with direct injection into the combustion chamber range between 5000 and 25,000 psi. In spite of the lapped fit, there is some leakage past the plunger, and this affects the accuracy of the metering function. The fuel quantities injected into individual cylinders of a multi-cylinder engine must be equal within 3 to 5 per cent. Leakage assumes greatest importance at low speeds; it naturally increases with wear, and with a pump in an advanced state of wear it is generally difficult to start the engine, because too little fuel is being injected.

Elements of Pump—An injection pump consists of a barrel, a plunger adapted to reciprocate therein, a valve or port through which fuel is admitted to the barrel during the suction stroke, a valve or port through which it is discharged during the delivery stroke, and means for reciprocating the plunger within the barrel. Owing to the high pressures against which the pump must work and the high degree of accuracy required in its operation over a long period of service, all of the parts must be made of high-grade material, designed to possess a relatively high degree of rigidity, and accurately machined or fitted. The pump barrel and plunger generally are made of forged steel and hardened. Bronze, semi-steel and the better grades of cast iron, such as the pearlitic irons and alloy irons developed in recent years, also may be used for the barrel. All parts are accurately ground, and the plunger is lapped into the barrel.

With plunger pumps for other purposes it is customary to use a stuffing box to prevent leakage past the plunger, but owing to the fact that fuel injection pumps are usually operated by a cam and spring, the cam moving the plunger positively during the delivery stroke and the spring returning it during the suction stroke, it is not practical to use a stuffing box. Dependence for oil-tightness therefore must be placed on the fit between plunger and barrel, and the sealing effect of the viscous fuel. To reduce the leakage to a minimum, the plunger must have a comparatively long bearing in the barrel. In order to reduce wear and minimize leakage, it is also desirable that motion be imparted to the plunger in such a way that there is relatively little or no side thrust of the plunger on the barrel, which can be accomplished by using a tappet. In most designs of injection pump the barrel is not the pump body but a sleeve inserted in the pump body.

Pump Valves—For the proper operation of the pump it is also necessary that the valves close tightly and open freely. The valves of an injection pump may be either of the automatic or positively-actuated type. Automatic valves may be either of ball or conical form; they are opened by a difference in pressure on opposite sides, and closed by a spring. Such valves have given quite satisfactory service at low and moderate speeds, but it is doubtful whether they can be made to function properly at really high speeds. This applies particularly to the suction valve, which must open under the influence of atmospheric pressure plus whatever pressure head there is on the fuel, hence its spring cannot be made very strong.

Conical valves can be actuated also mechanically, like the inlet and exhaust valves of engine cylinders. Mechanical actuation is always used when this type serves as a "return-flow" or spill valve, as will be explained farther on.

Finally, the valving functions may be accomplished by ports in the wall of the pump barrel which are controlled either by the pump plunger itself or by a separate valve plunger adapted to reciprocate in a bore parallel with the pump cylinder. The most widely used type of jerk pump has ports in the wall of the barrel which are opened and closed by the pump plunger.

A simple diagram of one type of injection pump is shown in Fig. 1. *A* is the pump barrel; *B*, the plunger; *C*, the fuel inlet; *D*, the inlet or suction valve; *E*, the delivery valve, and *F*, the delivery-line connector. The plunger is moved up in the barrel by the cam shown at the bottom, and returned by a coil spring.

Delivery Control—Fundamentally there are three methods of varying the amount of fuel delivered by the pump per stroke. The first (Fig. 1) consists in varying the stroke of the pump plunger and delivering the whole of the fuel charge displaced by the plunger to the engine cylinder; the second, in throttling the fuel flow into the pump barrel more or less, by providing an adjustable needle valve in the inlet passage; and the third, in by-passing more or less of the fuel moved by the pump plunger to the inlet side of the pump or to the fuel tank. This can be done either by holding the suction valve open during a part of the delivery stroke, by pro-

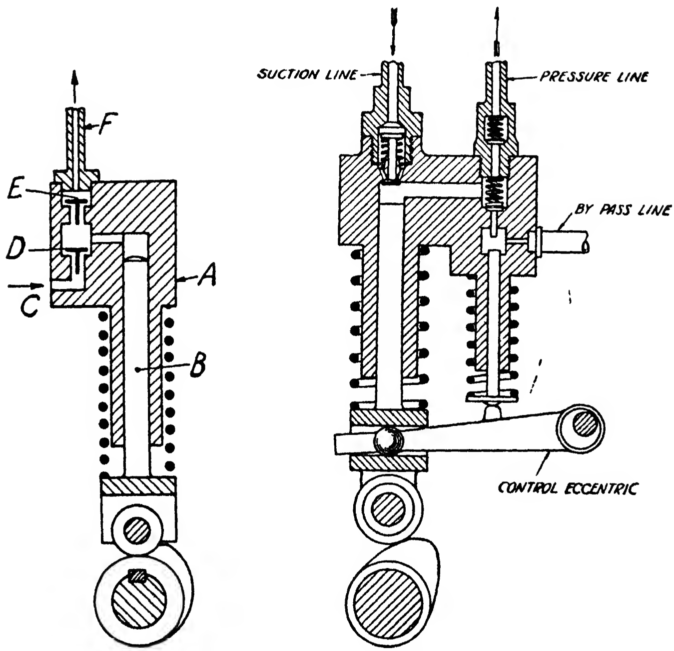


Fig. 1 (*Left*).—Pump with Delivery Control by Inclined Cam.
Fig. 2 (*Right*).—Pump with Delivery Control by Spill Valve.

viding a throttling spill valve remaining open continuously, or by providing a spill valve which is opened at some point of the delivery stroke, the point being variable, of course.

Control by Timed By-Pass—The system of delivery control by a timed by-pass or spill valve is illustrated in Fig. 2. With this system the pump has three valves, a suction valve, a delivery valve, and a spill valve. Delivery begins shortly after the beginning of the up-stroke of the plunger, and continues until the spill valve is lifted from its seat. Lifting of the spill valve is effected by a lever arm whose free end moves up and down with the plunger, with which its free end has a sliding connection. The fulcrum of the lever can be raised or lowered by means of an eccentric, and the time of opening of the spill valve relative to the plunger stroke thereby advanced or retarded. The earlier the spill valve is opened, the smaller the amount of fuel injected. Injection control may be effected also by means of a mechanically-actuated suction valve, which then is held off its seat throughout the suction stroke and during a variable portion of the delivery stroke. This, however, causes injection to begin later at light than at full load, and it is undoubtedly better to have the start of injection substantially independent of the load and control the engine by varying the point of cut-off.

In the type of injection pump in most extensive use today, fuel is admitted through ports in the wall of the barrel which are uncovered by the plunger toward the end of the downstroke or suction stroke, and is forced into the line through a delivery valve at the top of the barrel. The delivery is controlled by the plunger. Part of the cylindrical surface of the plunger is undercut. Delivery begins when the top of the plunger passes the top edge of the port, closing the latter, and it ends when the undercut portion of the plunger begins to register with a port. The undercut portion has a helical upper edge, and by rotating the plunger around its axis, the port can be uncovered earlier or later in the stroke. With the second and third methods of delivery control the stroke of the pump plunger is fixed.

Plunger-Controlled Port—The action of an injection pump in which the delivery is controlled by means of ports opened and closed by the plunger is illustrated by the three diagrams of Fig. 3. The view at the left shows the plunger at the bottom of the stroke, and fuel is being forced into the pump barrel through ports in the wall of the barrel by the transfer pump. In the view in the center, where the plunger has partly completed the up-stroke, the ports are closed by the plunger, and fuel is being forced from the pump barrel through the delivery valve at the top. The pressure of the fuel in the line has lifted the valve of the injection nozzle off its seat, and fuel is being injected into the combustion chamber. In the view on the right the plunger is shown farther up the pump barrel, and the land of the plunger with helical lower edge has uncovered one of the ports. This has relieved the pressure on the fuel in the pump chamber, and as a result the injection valve has closed and injection has ceased.

Injection Timing—In a spark-ignition engine the spark is advanced as the speed increases, because it takes time for the fuel to burn, and if the spark occurred so late that at low engine speeds the cylinder pressure reached its peak value only after the top dead-center position, then at high speed some of the fuel would burn so late during the expansion stroke that only a small fraction of its energy would perform useful work on the piston, and most of it would pass out with the exhaust. Combustion conditions are similar in the compression-ignition engine. First of all, there is an ignition lag. If this lag had a constant time value, its duration in crankshaft degrees would be greater at high than at low speeds. The same applies to the combustion period. Some time elapses between the moment ignition occurs and

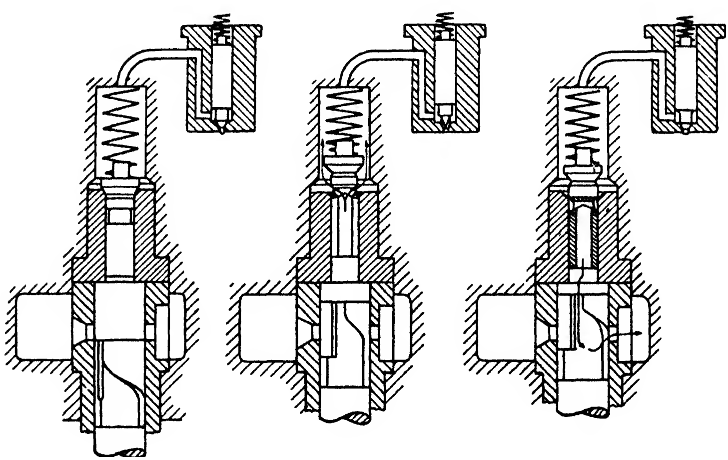


FIG. 3.—PHASES IN CYCLE OF JERK-PUMP OPERATION.

On Left—Plunger at Bottom of Stroke, Barrel Filled with Fuel.
In Center—Plunger Moving Upward, Fuel Being Injected.
On Right—Relief Port Uncovered by Plunger, Injection Has Ceased.

the moment the peak pressure is reached, and if this time were independent of engine speed, it would correspond to a greater number of crankshaft degrees at high than at low speeds. Actually both the ignition lag and the combustion period in time measure are reduced at high engine speeds by the greater turbulence then prevailing in the combustion chamber. Turbulence effects, however, vary greatly in different engines. In high-turbulence engines the ignition lag and the period of pressure rise, expressed in degrees of crankshaft rotation, may be nearly constant over a considerable speed range, and permit of fixed injection timing. This, of course, applies only to the beginning of injection, the end of injection, or the cut-off, being always varied in accordance with the load on the engine.

In engines with relatively quiescent combustion chambers the ignition-lag and pressure-rise periods in time measure are nearly constant, which makes it desirable to advance the beginning of injection as the speed increases. Both manual and automatic injection timing devices have been worked out, and an example of each type will be described later on. With the manual timing device the injection can be retarded for starting and idling, and advanced for normal operation, while with automatic timing, injection is advanced gradually as the speed increases, by means of a speed-sensitive device which usually comprises masses subject to centrifugal force.

It has also been proposed to make use of the pressure on the fuel in the supply line to the injection pump for injection-timing purposes. The fuel is caused to flow through a calibrated orifice, so that its pressure ahead of the orifice will vary with the rate of fuel flow, that is, with the engine speed. The oil pressure is made to act on a piston against the pressure of a spring, and the piston acts on the shifter fork of the injection-timing device. Most modern high-speed engines, however, have fixed injection timing.

Speed-Delivery Characteristics of Pumps—In the foregoing, methods have been described by which the quantity of fuel injected into the engine per cycle can be varied either by hand or automatically by means of a governor. The question now arises as to whether—and if so how—the quantity of fuel injected varies with the engine speed when the latter is changed by the torque load, the position of the control device remaining unchanged. But before discussing this question it may be well to consider how the quantity injected should vary with the engine speed, if at all.

It is obvious that for the "full-load" setting of the control device at least, that particular speed-delivery characteristic of the pump is best which for all engine speeds shows delivery of the maximum amount of fuel which can be burned with a clear exhaust. This amount of fuel is substantially proportional to the mass of air inducted into the engine at any given speed, and under given atmospheric conditions it is proportional to the volumetric efficiency. By volumetric efficiency is meant the ratio of the mass of air which enters and is retained in the cylinder during each cycle, to the mass of a volume of air equal to the piston displacement at the prevailing atmospheric pressure and temperature. The mass of air passing through the engine per cycle is given by the expression

DWe, where D is the piston displacement in cu ft; W, the weight in lb of one cu ft of air under the prevailing conditions of pressure and temperature, and e, the volumetric efficiency.

In Fig. 4 the upper one of the two curves is a volumetric efficiency curve of an actual Diesel engine, plotted on a base of "velocity-of-air-flow" through the inlet valve—which for a given design is proportional to the engine speed. It shows that the volumetric efficiency decreases continuously as the speed of the engine increases. This is due to the fact that the velocity of flow through the valves and passages increases with engine speed, and the resistance to air flow increases likewise. The lower curve in Fig. 4 is a composite volumetric-efficiency curve of about a dozen high-speed gasoline engines. That the volumetric efficiency of these engines at fairly low

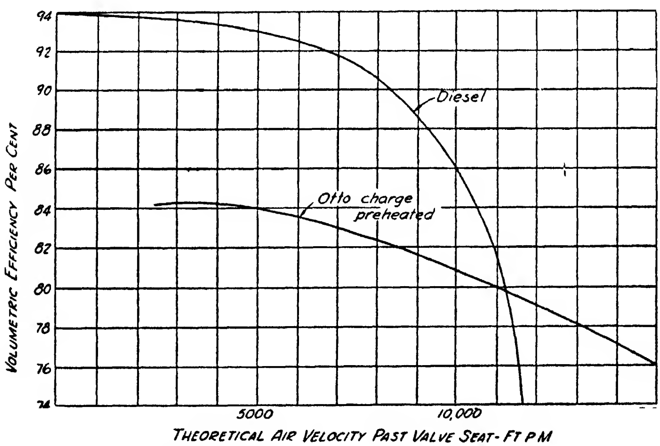


FIG. 4.—TYPICAL VOLUMETRIC-EFFICIENCY CURVES FOR DIESEL ENGINE AND OTTO ENGINE WITH PREHEATING OF THE CHARGE.

speeds is considerably lower than that of the Diesel is due partly to preheating of the carburetor air and partly to the resistance to flow through the carburetor throat.

Characteristics of "Valved" Pumps—The "speed-delivery" characteristic of the injection pump evidently should be of the same general form as the volumetric-efficiency curve of the engine. That is to say, the delivery per cycle should increase as the speed decreases. Pumps with spring-closed suction valves possess this general characteristic, though the effect of a change in speed on the delivery is likely to be greater than desirable. The straight-line characteristic in Fig. 5 is that of a pump with a spring-closed suction valve. It will be seen that as the speed of the pump drops from normal to 20 per cent of normal, the delivery per stroke increases from normal to 60 per cent above normal. The maximum increase in the mass of air drawn into the cylinder as the speed drops below normal is usually only about 25 per cent, hence with this particular pump there is a tendency for the cylinders to get too large a fuel charge at low speeds. This, of course, tends to increase the torque of the engine up to a certain point, and engines equipped with this type of pump are characterized by great "lugging" ability. The disadvantage of this kind of pump characteristic is that if the pump is adjusted to deliver exactly the right amount of fuel at normal speed, it will deliver too much at low speed, with the result that the exhaust will be smoky and the fuel economy low.

That the delivery of a pump with spring-closed suction valve increases as the speed decreases is evidently due to lag in the closing of the valve. Some of the fuel drawn into the pump barrel during the suction stroke is forced out again before the valve is fully closed, and the amount thus returned is greater at high speeds, because the valve will then close later in the stroke. With an inlet port closed by the plunger the timing of port closing and of the beginning of delivery is independent of the speed.

Characteristic of "Ported" Pump—The curved line in Fig. 5 shows the variation of delivery with speed for a pump in which the intake and return flow take place through ports controlled by the motion of the pump plunger. With most of these pumps the delivery has a tendency to increase with the speed, though this tendency is considerably more pronounced in some designs than in others, in which the delivery is substantially constant over the upper half of the speed range. One reason the delivery in this type of pump increases with speed is that immediately after opening of the port, and before the delivery valve has time to close, expansion of the fuel in the line causes a slight return flow past the delivery valve, and this is less at high speed when, on account of the higher line pressure, the delivery valve closes more rapidly. A second reason is that delivery begins before the inlet port is completely closed, because of the excessive throttling of the flow when the port is very nearly closed, and the consequent rapid building up of pressure in the barrel.

It is obvious that when the quantity of injection increases with speed, if the pump is set to give the proper quantity at high speed for maximum power with clear exhaust, it will not deliver quite enough fuel at low speed, and with such pumps the torque of the engine does not increase so rapidly with decrease in speed as with pumps with spring-closed suction valves. Where port-controlled pumps are used, it is customary to so adjust the delivery that the air inducted into the cylinders is used to best advantage at intermediate speeds. The amount injected at high speeds will then be slightly too great, and there is likely to be smoke in the exhaust at high speeds, indicating incomplete combustion. This, however, can be corrected by the driver by means of the control member. At low speeds, with the control set for maximum delivery, the amount of fuel injected will be slightly too small, and the engine in consequence will not have quite the "lugging" ability that it would have if the ratio of fuel to air were independent of engine speed. Fig. 6 shows torque curves of the same engine when fitted with the two pumps whose delivery characteristics are shown in Fig. 5, respectively. It

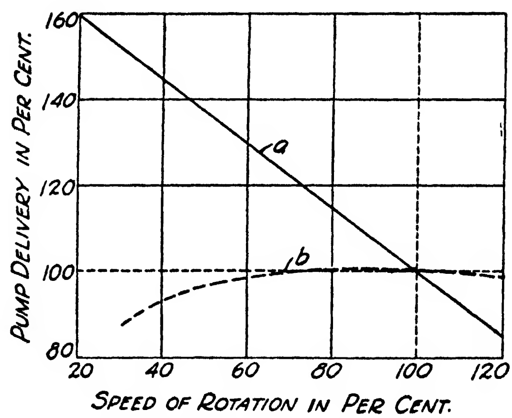


Fig. 5.—Speed-Delivery Characteristics of Pumps with Spring-Closed Inlet (*a*) and Plunger-Controlled Inlet (*b*).

will be seen that at lower speeds the torque is much greater with the pump having a spring-controlled suction valve.

Effect of Pump Characteristic on Engine Performance—There are two distinct disadvantages inherent in a pump-delivery curve such as shown by *b*, Fig. 5. Most injection pumps are provided with a stop for the control rack which limits the maximum fuel quantity that can be injected per cycle. If this stop is so set that it just permits tolerably smokeless operation at full speed, then at low speeds the engine does not receive enough fuel to take full advantage of the air charge inducted, and the torque is considerably less than it should be. The result is a practically flat torque curve, as shown by *b*, Fig. 6. Besides lacking "lugging ability," an engine with such a torque curve is not stable in the lower speed range. No engine in which the torque increases and decreases automatically with the speed can be stable in operation, unless the torque load also decreases automatically with speed, as in propeller work.

Referring to Fig. 6, assume that the engine is running at 60 per cent of normal speed and developing the maximum torque corresponding to that speed. If the load now increases for any reason, the engine will slow down, and owing to the pump characteristic, it will then receive a smaller quantity of fuel. This again will cut down the engine torque and speed, and further reduce the injection quantity, with the result that finally the engine stalls. Now let us suppose that the engine is running under light load at, say, 30 per cent

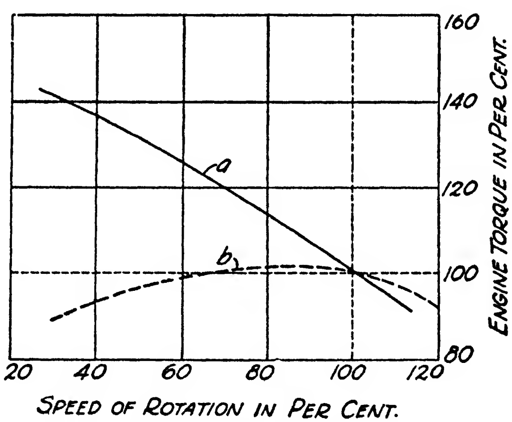


Fig. 6.—Torque-Speed Characteristics of Engines Equipped with Fuel-Injection Pumps Having Spring-Closed Inlets (*a*) and Port-Controlled Inlets (*b*).

normal speed. If for any reason the load is reduced further, the engine immediately will speed up, and this will result in an increase in the fuel quantity injected. The increased fuel charge will result in a further increase in speed, and finally the engine will race. To prevent such instability in operation, automotive-type Diesel engines sometimes are provided with a special type of governor which not only limits the maximum speed but also controls engine speed in other parts of the range. Some governors of this type will be described in the next chapter.

Improvement of Delivery Characteristics—Manufacturers of injection pumps have been at pains to improve their pumps so as to obtain a delivery-speed characteristic corresponding fairly closely to the engine volumetric-efficiency curve. The speed-delivery characteristic is influenced to some extent by the design of the delivery valve, because as soon as the pressure in the system is relieved by the opening of the port, fuel begins to flow back from the line into the pump barrel, a phenomenon sometimes referred to as "re-induction," and the amount of fuel which thus returns from the line to the pump depends on the speed with which the delivery valve can close. A change made in the design of the delivery valve of an early Bosch injection pump is illustrated in Fig. 7, where *A* and *B* represent the old and the new design respectively. It will be seen that four slots are cut in the stem of the valve, through which the fuel has to pass. In the older valve the section of the slot was uniform over practically the

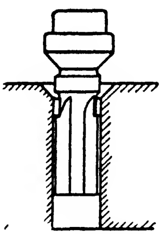


FIG 7A.

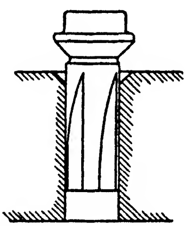


FIG 7B.

FIGS 7A AND 7B (*Above*).—OLD AND NEW DELIVERY VALVE OF BOSCH PUMP.

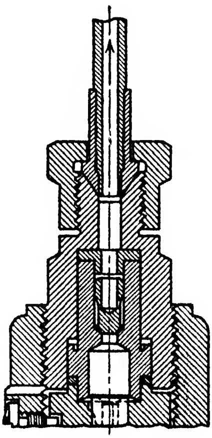


FIG. 8 (*Right*).—SIMMS DELIVERY VALVE.

entire length, while in the new one it tapers gradually. This evidently results in an increased lift of the valve for a given pump speed, and owing to the increased lift the valve takes a longer time to close, which increases the "re-induction" or return flow of fuel.

A valve specially designed to improve the pump delivery characteristics is used also in the Simms injection pump manufactured in England. As shown in Fig. 8, this is a poppet valve which, when off its seat, allows the fuel to flow into an annular space between it and its guide. From this space the fuel passes through a small radial hole into an axial hole in the upper part of the valve. The radial hole is the point of greatest restriction, instead of the passage through the valve, as in most other designs. This is claimed to make it possible to so design the valve that even at high speeds it will allow a certain amount of fuel to return from the line before the delivery valve closes following the cut-off action.

Governor Control of Fuel Delivery—In recent years the preferred method of improving the speed-delivery characteristic of the injection system (or the speed-torque characteristic of the engine) has been by modifications to the governor, rather than to the pump delivery valve. With an increase in speed, the centrifugal force on the governor weights increases, and they tend to move out from their axis of rotation. The weights are connected to the pump control rod in such a way that their outward motion cuts down the fuel delivery. With the conventional governor, designed to limit the maximum engine speed, the centrifugal weights begin to act on the control rod only as maximum speed is approached; but by providing the governor with softer springs (or springs less preloaded), it can be made to act on the control rod practically throughout the entire speed range. In that case, any increase in engine speed cuts down the injection quantity and, therefore, the engine torque, thus giving the drooping type of torque curve which ensures stability in engine operation and great "lugging power." A few governors based on this principle will be described in the next chapter.

Elasticity of Fuel and Tubes—It is evident that all of the fuel entering the line from the pump must leave it at the injection valve; but the rate at which fuel enters the line at any particular moment of the injection period is not necessarily the same as the rate at which it leaves the line. This is due to elasticity of the fuel and of the metal tube. It is customary to use tubing with rather heavy walls, as compared with the inside diameter, and the effect of the expansion of the tube is negligible compared with the compression of the fuel, even if the tubing is made of brass with a modulus of elasticity of 12,000,000, instead of steel with a modulus of elasticity of 30,000,000 psi.

The compressibility of fuel oil was investigated by D. H. Alexander (*Trans. Inst. of Marine Engineers,* 1927-8) for pressures up to 5000 psi, and his results are given in Fig. 9. The bulk modulus, it may be explained, is the inverse of the fraction representing the reduction in the bulk of fuel by a surface pressure of one pound per square inch. R. S. Jessup (*Bureau of Standards Journal of Research,* November, 1930) investigated the compressibility and thermal expansion of a number of petroleum oils, but although his temperature range was from 0 C to 300 C, the pressure was carried only to 700 psi. Fig. 10 shows the results from two gas oils of widely different densities and viscosities. He also found that the mean compressibility of gas oil decreases only slightly with increase in pressure, but, as the curves show, the compressibility is very susceptible to temperature changes and is widely different for the two oils. Jessup also found that petroleum oils of the same density and viscosity had, within narrow limits, the same thermal expansion and compressibility.

A ready rule for the compressibility of petroleum fuel is one twenty-thousandth for every additional atmosphere of pressure on it. The relative effect of the elasticity of tubing may be judged from the fact that if steel tubing is worked at

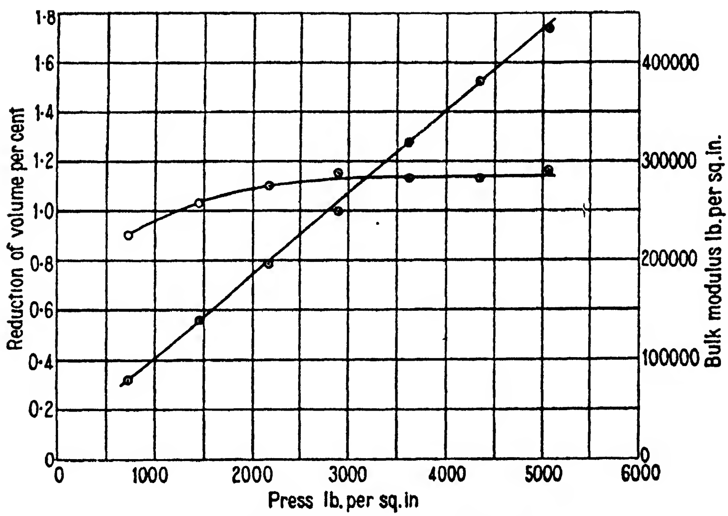


FIG. 9.—COMPRESSIBILITY OF FUEL OIL.

a fiber stress of 30,000 psi, or brass tubing at 12,000 psi, the diameter of the tube will increase only 0.1 per cent, and the area 0.2 per cent.

Injection Lag—When an injection valve is opened by the pressure of the oil in a tube connected to it, the pressure of the oil being controlled by a timing valve or a pump some distance away, there will be a time lag between the opening of the timing valve (or the delivery valve of the pump) and the beginning of injection by the nozzle, owing to the expansibility of the tube and the compressibility of the fuel. For instance, in some experiments reported in the paper by D. H. Alexander, referred to in the foregoing, although the pump stroke started 30° of crank motion before top center, injection began only 6° before top center.

The injection lag is in reality made up of two periods, one being that required by the pressure wave to travel through the line from the pump to the nozzle, the other that required by the valve to leave its seat after the pressure wave has arrived there. The first period in time units depends only on the length of the line (aside from characteristics of the fuel and line material) and in pumpshaft degrees is therefore directly proportional to the pump speed in rpm. The second period in pumpshaft degrees is independent of the pump speed and the length of the line, but is much greater if the line pressure is relieved between injections than when it is not.

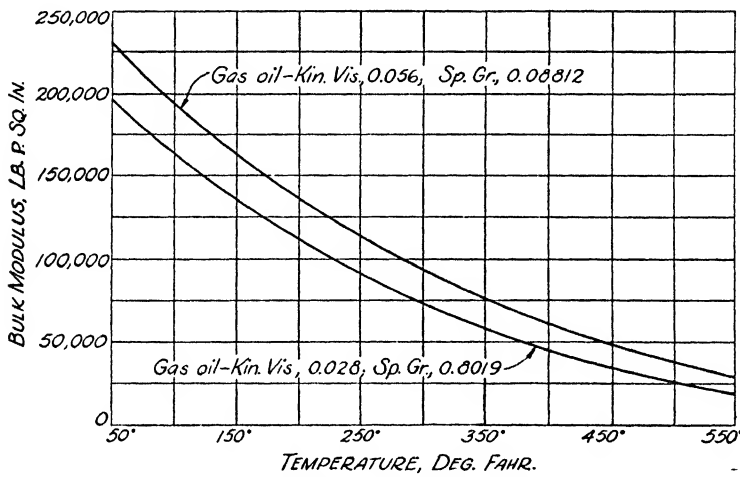


Fig. 10.—Effect of Temperature on Compressibility of Fuel Oil.

A comprehensive investigation of the subject of injection lag was made by Dr. Hans Heinrich, whose results were published in *Dieselmaschinen V* (Society of German Engineers, 1932). In his experiments the second part of the injection lag period was of the order of 1.5 pumpshaft degrees without line relief, and 5.0 pumpshaft degrees with line relief. It will be seen that the second part is very small when there is no line relief, and in that case, therefore, the total injection lag in pumpshaft degrees is substantially proportional to the length of the line.

Fig. 11 represents some of the results obtained by Heinrich with a jerk-pump injection system operating without line relief, the pump-plunger diameter being 0.315 in.; the internal diameter of the fuel line, 0.08 in., and the length of the line, 40 in. Fig. 12 shows results of the N.A.C.A. and covers the effects of both injection pressure and line length.

At very low pump speeds, when the amplitude of the pressure wave is low, it may happen that, although the valve is lifted upon the arrival of the wave, it seats again as a consequence of the reduction in pressure due to opening of the valve. The cross section of the fuel line has no appreciable influence on the injection lag. In Heinrich's set-up, the nozzle-opening pressure had no appreciable influence on the injection lag in degs at pump speeds of over 1000 rpm, while at lower speeds the lag increased somewhat with the opening pressure.

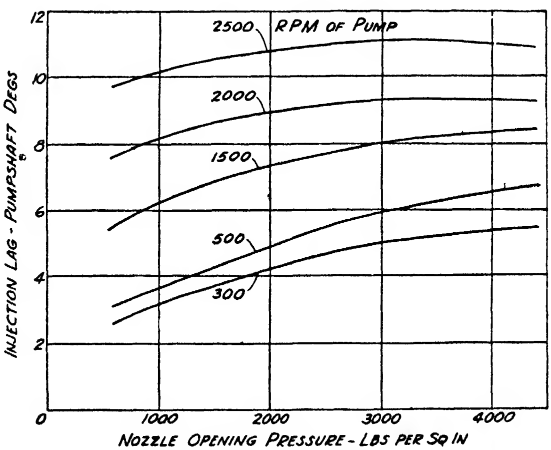


FIG. 11.—EFFECT OF NOZZLE-VALVE OPENING PRESSURE ON INJECTION LAG AT VARIOUS PUMP SPEEDS.

Line-Pressure Relief—With direct pump injection, as soon as the pressure in the pump barrel is relieved by the opening of the spill valve, the delivery valve closes. Delivery of fuel through the spray nozzle continues for a while, however, owing to expansion of the fuel in the line as the pressure in it decreases gradually, and to the simultaneous contraction of the tubing carrying the fuel. This is known as after-dripping. Such dripping is objectionable for various reasons. In the first place, since this fuel is sprayed under reduced pressure, the droplets are large and its combustion therefore is poor. Then, since this fuel burns rather late in the cycle, the efficiency of conversion of its heat energy is low. With an open nozzle, after-dripping is affected also by the fact that during the expansion stroke the pressure in the cylinder drops from 30 atmospheres or more to a few atmospheres.

Dripping due to compressibility of the fuel is considerably less with closed than with open nozzles. To reduce it still further, line-pressure-relief devices are extensively used, by means of which a small amount of fuel is withdrawn from the line just before the delivery valve closes. The manner in which this is accomplished in the American Bosch system is illustrated in Fig. 13. The delivery valve, which is of the conical type, has a collar on its stem which exactly fits the guide. During delivery the fuel passes through the hollow stem and through two inclined holes to a groove cut in the stem, and thence past the valve into the line, as shown in the right-hand view. When delivery into the line ceases,

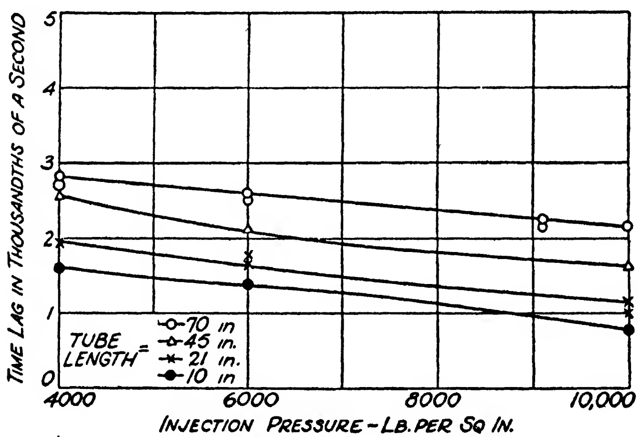


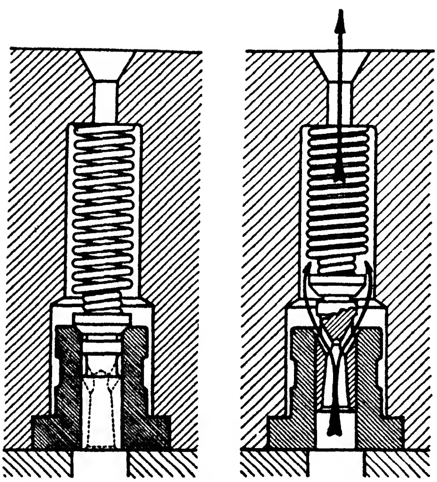
FIG. 12.—EFFECT OF INJECTION PRESSURE ON INJECTION LAG.

communication between the pump chamber and the line is shut off by the collar on the valve stem entering the guide. The rest of the seating motion of the valve increases the capacity of the line, thereby taking care of expansion of the fuel and contraction of the tube.

Besides preventing—or at least materially reducing—after-dripping, relief of line pressure (or unloading of the line as it is usually called by British writers) has some other effects, notably on the injection lag. The lower the pressure in the line between injections, the more the fuel in the line must be compressed before injection can start again, hence the longer the injection lag. This was confirmed by the experimental work of Heinrich already referred to.

It is obvious that with the system of line-pressure relief described in the foregoing, since the volume of the line is increased in a definite proportion at the end of injection, the line pressure is reduced in a certain proportion. Now, the injection pressure, or the pressure in the line at the moment the spill valve opens and injection ceases, varies greatly with the speed of the pump (and engine), and the residual line pressure therefore also will vary greatly. At high speed the residual pressure is higher and the injection lag shorter.

Constant Residual Line Pressure—A method of relieving the line pressure which—theoretically—results in constant residual pressure in the line was described in a paper presented to the Institution of Automobile Engineers in 1931 by



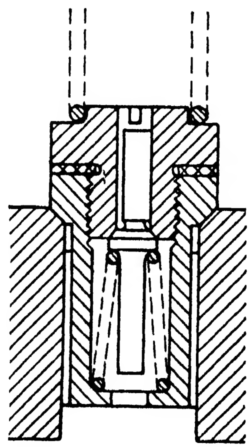


FIG. 13 (*Left*).—AMERICAN BOSCH LINE-UNLOADING DELIVERY VALVE IN CLOSED AND OPEN POSITIONS.

FIG. 14 (*Right*).—LINE-UNLOADING DELIVERY VALVE INDEPENDENT OF ENGINE SPEED.

S. W. Nixon, and is illustrated in Fig. 14. It makes use of a normal delivery valve with a small relief valve inside of it, which opens in the opposite direction to that in which the delivery valve opens. The relief valve is closed by a spring, the strength of which governs the pressure remaining in the line. As soon as the pump chamber is relieved of pressure by the opening of the port, the delivery valve seats and fuel under pressure is momentarily trapped in the line. The pressure of this fuel immediately opens the relief valve, and the line pressure then decreases until the relief valve is closed by its spring. Unfortunately, the delivery valve is a very small part, and a relief valve inside of it would have to be of almost microscopic dimensions.

Elimination of Pump Delivery Valve—Line-pressure relief could be effected in a very simple way by merely doing away with the delivery valve of the pump and relying on the nozzle valve to perform the function of a delivery valve. This would result in a very rapid drop in line pressure down to that maintained in the pump-inlet line, and thus would prevent after-dripping completely. It would decrease the delivery of the pump for any given position of the control rod, by an amount depending upon the capacity of the line and the closing pressure of the injector valve. However, the principal objection to the system is that if any air should get into the line, it would be extremely difficult to remove it, because operating the pump by means of a priming lever would merely compress and expand the air. Owing to the fact that with this system the delivery depends on the capacity of the line, it would also be necessary to see to it that the lines to all of the cylinders had substantially the same capacity.

Pressure Waves in Fuel Lines—If the fuel were incompressible and the material of the fuel tube inexpansible, then the velocity of fuel flow and the pressure in the fuel at any instant would be the same at both ends of the line. Any change in velocity at the pump end would be instantly transmitted to the nozzle end, and since the fuel in the line would form a rigid column, the pressure imparted to it by the pump would be transmitted without change to the nozzle end. However, owing to the elastic nature of the liquid column and the property of inertia possessed by it, any increase of the pressure of the fuel at the pump end due to motion of the pump plunger results in a pressure wave which is propagated along the tube at a velocity which depends on the coefficients of elasticity (or bulk moduli) and the densities of the materials. Since the tube material plays a very minor part in the phenomenon as compared with the fuel, the former will be neglected in the following discussion. The blow of the pump plunger sets up a compression wave in the fuel, and the velocity at which this pressure wave is propagated through the fuel line is the same as that at which sound travels in the same medium.

Velocity of Pressure Waves—The velocity with which an elastic disturbance is transmitted through a fluid is given by the equation

$$V = \sqrt{\frac{Kg}{\rho}} \text{ in. per sec,}$$

where K is the bulk modulus of the fluid in psi; g, the acceleration of gravity, in ipsps, and ρ, the density of the fluid in lb per cu in. An average fuel oil has a bulk modulus at normal atmospheric temperature of about 280,000 psi; that is to say, when the fuel is subjected to a pressure of 1 psi on its surface, it contracts one part in 280,000 of volume. The value of g is 386 ipsps, and the value of ρ, the weight of one cu in. of fuel, for an average Diesel fuel of 0.86 specific gravity amounts to 0.0311 lb. Hence the velocity of pressure waves in such a fuel is

$$\sqrt{\frac{280{,}000 \times 386}{0.0311}} = 59{,}000 \text{ ips},$$

or, roughly, 4900 fps.

Actual measurements of the velocity of the pressure wave in gas oil in small-bore tubing were made by E. Blaum at Dresden Technical College. The velocity increases with the pressure and with the inside diameter of the tube. For gas oil of a specific gravity of 0.86 and a 3-mm (⅛-in.) tube the velocity is given by the equation

$$V = 4450 + 0.072P \text{ fps},$$

and for a 1.5-mm ($\frac{1}{16}$-in.) tube by the equation

$$V = 4340 + 0.076P \text{ fps},$$

P being the line pressure in psi.

Reflection of Pressure Waves—When a pressure wave from the pump end arrives at the nozzle and its amplitude is insufficient to open the nozzle valve, it is completely reflected and travels back through the line, and when it reaches the advancing pump plunger it is again reflected and adds further to the line pressure. The effect is somewhat as illustrated in Fig. 15. In addition to the pressure wave there is also a velocity wave. That this should be so is obvious from the fact that pressure variations at any point along the line are accompanied by a change in bulk of the fuel, and such a change in itself produces a flow of fuel along the tube; in other words, a velocity. And since pressure waves are propagated along the line, the velocity variations also travel along it. There is one difference between the pressure waves and the velocity waves, however, and that is that while the pressure waves when reflected add to the pressure already in the line, the velocity wave when reflected subtracts from the velocity of the wave as it arrives at the end of the line, so that at the nozzle end, for instance, if the valve remains closed, the velocity wave has a zero amplitude.

In any analysis of the effects of pressure waves, four different cases must be considered: If the nozzle is closed when the wave arrives there, the latter is totally reflected. If the nozzle opens but fuel leaves the line at a rate lower than that at which it enters at the pump end, the wave is partly reflected. If fuel leaves the line at the same rate at which it enters, the pressure wave comes to an end at the nozzle, and if fuel leaves the line faster than it enters, a negative pressure wave passes back through the line to the pump.

Amplitude of Pressure Waves—As would be expected, the amplitude of the initial pressure wave passing through the line depends upon the velocity of flow which is set up in the line by the motion of the pump plunger. This velocity v is

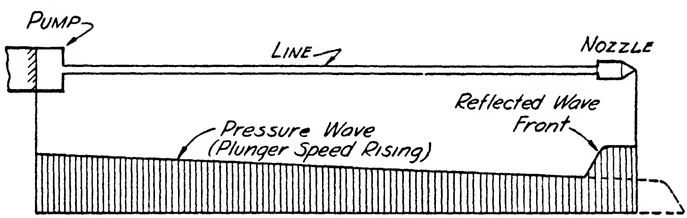


FIG. 15.—DIAGRAM OF PRESSURE WAVE IN LINE AND ITS REFLECTION.

equal to the product of the pump-plunger velocity at the moment the inlet port closes by the square of the ratio of plunger diameter to bore of fuel line. If the bulk modulus of the fuel is designated by K and the velocity of sound in the fuel by V, then the increase in line pressure by the pressure wave is

$$p = \frac{K}{V} v \text{ psi.}$$

For the same values of K and V as used before, viz., 280,000 and 4900, respectively, we find that

$$p = 57v;$$

that is to say, the amplitude of the pressure wave (the height of the wave front) is equal to 57 psi for every fps increase in the velocity of flow.

Since the amplitude of the pressure wave is proportional to the velocity which the plunger motion sets up in the line, it follows that it is inversely proportional to the square of the bore of the fuel line.

Form of Pressure-Wave Front—In theoretical investigations it is generally assumed that the pressure wave has a "vertical" front; that is to say, that there is a material rise in pressure between one section of the line and another section at an infinitely small distance therefrom. Such a "vertical" wave front is a physical impossibility, however. What gives a semblance of justification to the assumption of a "vertical" wave front is the fact that with a pump in which the inlet port is closed by the plunger after the latter has completed a moderate fraction of its stroke, at the moment the delivery valve opens, the pump plunger already has a definite velocity, and as the velocity of the fuel in the line is theoretically a definite multiple of the plunger velocity—the ratio between the two being the ratio of the cross sections of pump bore and line bore—the fuel in the line seems to start with a definite velocity. As a matter of fact, however, when the inlet port is very nearly closed by the plunger, the fuel cannot escape through it as rapidly as it is being displaced by the plunger, and a pressure is being built up in the pump barrel, so that the delivery valve may begin to lift before the inlet port is fully closed; besides, the fuel begins to compress in the pump barrel as soon as the pressure begins to rise therein, and in the line as soon as the delivery valve opens, hence the rate of pressure rise, though rapid, is always finite, and the wave front is "inclined."

Records of Pressure Waves—Actual records of pressure waves in fuel pipes of Diesel engines have been made at the Rugby Works of the English Electric Company by means of a Farnboro electric indicator fitted with a pressure-reduction device. One of these curves, reproduced from the I.A.E. paper of Davies and Giffen, is shown in Fig. 16. The solid line shows the variation of pressure at the pump end, and the dotted line that at the injection-valve end. The pressure waves set up after closing of the injection valve are very marked, but are not sufficient to cause the valve to re-open; that is, there is no secondary injection. It will also be noticed that the damping effect is quite strong and that the waves die out quickly.

Number of Pressure Waves per Injection—Since the velocity of the pressure wave is of the order of 5000 fps, for a line length of 2 ft, which is the approximate length on large truck engines, it takes

$$2/5000 = 0.0004 \text{ sec}$$

for the wave to travel from the pump end to the nozzle end of the line, and if the engine turns at 1600 rpm for instance, and injection extends over 20° of crank motion, the injection period is equal to

$$\frac{60 \times 20}{1600 \times 360} = 0.00208 \text{ sec},$$

so that the pressure wave has time to travel five times through the line during the injection period.

In Fig. 16 the pressure wave is seen to travel back and forth only once during the injection period, the crest of the

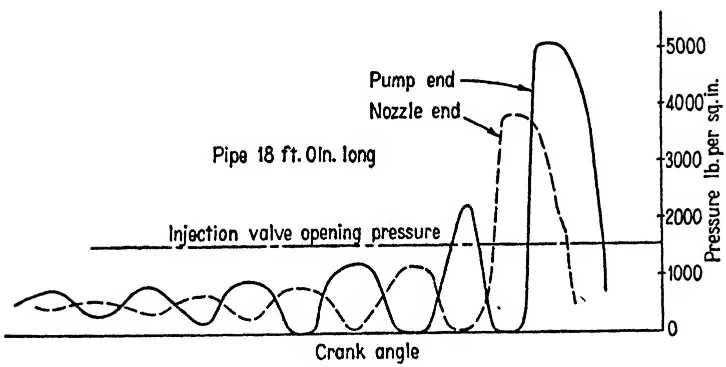


Fig. 16.—Pressure Waves in Fuel Line.

second wave as it arrives at the nozzle being below the opening pressure of the nozzle valve. This is due to the fact that the fuel pipe on which the record was taken was 18 ft long, or about ten times as long as in a conventional truck engine.

The effect of the pressure waves in the pipe is to increase the pressure at the nozzle stepwise. The time interval between the arrival of successive steps depends essentially on the length of the line, while the amplitude of the waves or steps depends upon the initial velocity of fuel in the line generated by the plunger motion; that is, it depends essentially upon the plunger velocity at the moment the inlet port closes and on the ratio of pump-bore section to line-bore section. Damping, of course, rapidly decreases the amplitude of the waves.

Davies' and Giffen's Experiments—Much experimental work on the effects of pressure waves in fuel lines has been done by Drs. Davies and Giffen of Kings College, London. They used a slotted disc stroboscope of a type first suggested by Dr. Schweitzer at the First National Oil and Gas Power Conference in 1928, which enabled them to collect the fuel delivered by the nozzle during one degree of cam travel, and by varying the phasing of the apparatus relative to the injection period, to obtain data which made it possible to trace curves of rate of fuel injection for the whole injection period. The form of these rate-of-injection curves (quantity of fuel injected per degree of crank angle as ordinate against crank angle as abscissa) depends, of course, on the variation of plunger speed with crank angle. During the early part of the plunger stroke the plunger velocity must necessarily increase, and during the latter part it must decrease, while in between it may be constant or nearly so.

Drs. Davies' and Giffen's experiments covered injection under all three conditions, with rising plunger velocity, falling velocity, and uniform velocity. In all cases there was a rapid initial rise in the rate of injection, to a value which was lowest for the rising plunger velocity, highest for the falling plunger velocity, and intermediate for the constant plunger velocity. During the following major part of the injection period, with rising plunger velocity the rate of injection rose slightly less rapidly; with constant plunger velocity the rate of injection rose slightly during the first half and remained constant during the second half of this period, while with falling plunger velocity the rate of injection, after continuing to rise more slowly for some time after the original rapid rise, decreased during the major part of this period. Any difference between rate of plunger displacement and rate of injection must, of course, be due to compression or expansion of fuel in the system between the pump plunger and the injection nozzle.

Injection-Rate Diagrams for Open-Nozzle System—The first series of experiments of Drs. Davies and Giffen were carried out with a Junkers pump, in which inlet and cut-off are controlled by plunger-closed ports, and with open nozzles, so the pressure waves were only partly reflected at the nozzle. One set of injection-rate diagrams is reproduced in Fig. 17. These show mainly the effect of pump speed on the rate and duration of injection. Since the nozzle is open, the line pressure relieves itself almost completely through the nozzle after the cut-off, and there is, consequently, considerable afterdripping, except at very low speeds, when the pressure in the line does not reach a very high value. The diagram shows that the higher the speed, the longer the duration of injection in terms of crank angle. The effects of the arrival of successive pressure waves at the nozzle are also quite evident in these curves. It is interesting in this connection that while the injection per degree of crank angle is greatest at low engine speeds, the injection per unit of time is greatest at high speeds.

Injection lag and after-dripping increase slightly with the viscosity of the fuel, but it was found that, except for very long lines, the effects are negligible from a practical standpoint. After-dripping, of course, increases with the length of the fuel line, as well as with the bore of the line. Using

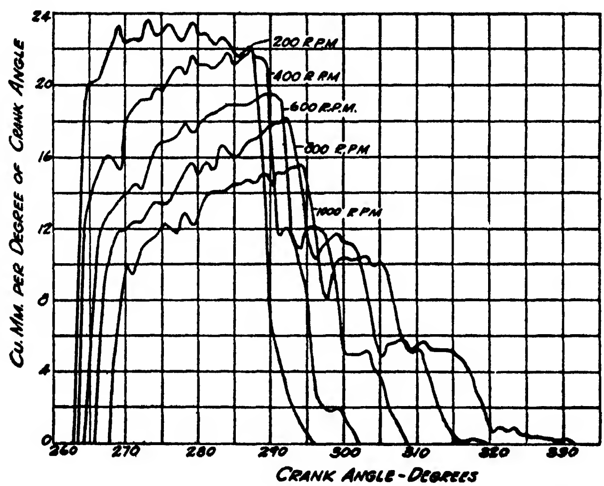


FIG. 17.—INJECTION RATE-CRANK ANGLE CURVES FOR DIFFERENT PUMP SPEEDS (OPEN NOZZLE).

orifices of three different diameters, 0.0075, 0.0125 and 0.0195 in. respectively, all of the same length/diameter ratio, it was found that the injection lag was least with the smallest and greatest with the largest orifice; the maximum rate of delivery was not very different for the two largest orifices, while with the smallest orifice it was much lower; the cut-off was sharpest with the intermediate-sized orifice, a little slower with the largest, and much slower with the smallest. Calculation showed that the maximum velocities of the sprays from the three orifices were substantially in the proportion of 6:3:1.5.

Fig. 18 shows delivery curves obtained by H. F. Bryan from an International Harvester Company closed nozzle at 650 pump rpm. One of the curves shows the volume delivered, in per cent of the total injection volume, plotted against elapsed time, in per cent of the total injection time, while the other shows the momentary rate of injection, in per cent of the mean rate, plotted against elapsed time. Aside from the fact that it is much smoother, the rate curve bears considerable resemblance to the open-nozzle curves in Fig. 17. As the valve opens, there is first a rapid increase in the rate of delivery, due to the fact that the high pressure in the line at the moment of valve opening rapidly accelerates the fuel. When about 20 per cent of the total injection time has

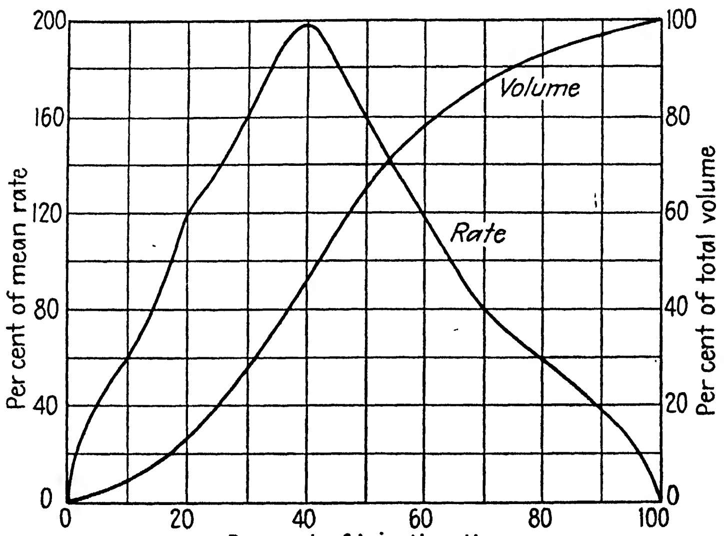


FIG. 18.—CURVES OF INJECTION RATE AND VOLUME INJECTED FOR A CLOSED NOZZLE.

elapsed, a state of equilibrium seems to be reached in the system. Thereafter the injection rate continues to increase only because the plunger velocity, and hence the line pressure, is still increasing. This continues for another 20 per cent of the total time. At this point the relief port of the pump is opened and the line pressure immediately begins to drop; the nozzle valve, however, remains open and injection continues until the line pressure drops below the valve closing pressure.

Eliminating Pressure Waves or Mitigating Their Effects—From the foregoing it is apparent that pressure waves in the fuel lines affect the injection process adversely in a number of ways. Pressure waves and their effects can be eliminated completely by combining an injection pump with the nozzle and mounting such an injection unit on each cylinder. The only objection to this practice is that the pump drive and the low-pressure fuel lines are somewhat more complicated. Where high-pressure fuel lines are used, all of them should be of the same length, so their effects on the injection will be the same in all cylinders. All lines will be of the same length naturally if a separate injection pump is used on each cylinder and all of the pumps are similarly located with respect to their cylinders. This practice has gained a certain vogue in connection with relatively large engines.

Preferred Form of Plunger Motion—Of the three possible forms of plunger motion during injection—with rising, constant, and falling velocity respectively—the first has definite advantages over the others. In the first place, the initial rate of injection is somewhat lower with it, hence less fuel is injected during the ignition-delay period, and combustion, consequently, should be smoother. Then, since the plunger velocity is a maximum at the end of the effective delivery stroke, the port opens rapidly, and the line pressure is relieved quickly, which should reduce the after-dripping.

Davies and Giffen have shown that at low speed the nozzle valve may vibrate violently during the injection period, so that there are extreme fluctuations in the delivery rate. At 100 rpm pump speed, for instance, the delivery rate rose rapidly almost to the maximum value, then dropped as rapidly to zero. Thereafter it passed through three more peak values, between which it dropped to from 25 to 65 per cent the preceding peak value. Of course, injection pumps operate at such low speeds only during the cranking period.

Calculation of Injection Pumps—Most injection pumps are operated by means of cams which move the plungers positively during the delivery stroke. This stroke, of course, is produced by the rising slope of the cam. During the first portion of the up-stroke the cam accelerates the cam follower and pump plunger upwardly, and during the last portion of the stroke it allows them to decelerate. If a design of cam is used which produces plunger acceleration according to a simple law, then the acceleration, speed and position of the plunger for any angular position of the cam can be readily calculated.

Constant-Acceleration Cam—One of the simplest forms of cam is that which imparts to the pump plunger a uniform acceleration during a certain portion of its stroke, and allows it to decelerate uniformly during the remainder of the stroke. The two portions of the stroke need not be equal, and it is advantageous to make the accelerating portion about twice as long as the other. The stroke, moreover, is made considerably longer than is required to deliver enough fuel to the cylinder under full load, so that it is possible to utilize only a fraction of the stroke near the middle thereof, when the plunger speed is at or near the maximum. This assures the high pressures required for thorough atomization. Moreover, by utilizing only the middle portion of the stroke, the pump plunger can be brought up to speed and decelerated more gradually, which makes for smoother operation and reduced strain on the mechanism. To ensure injection during a fraction of the delivery stroke near its middle, the inlet valve or suction valve or port is kept open until a certain portion of the delivery stroke has been completed, and a return-flow valve or spill valve or port is opened during the latter portion of the same stroke.

Sample Calculation—The calculations necessary to determine the required dimensions of a pump may be illustrated by a practical example. We will assume that the full stroke of the plunger is to be 0.42 in., and that the cam imparts this stroke to the plunger directly without intermediate levers, so that the throw of the cam must be 0.42 in. We will further assume that during the first two-thirds of the stroke (0.28 in.) the plunger is uniformly accelerated, and that during the last one-third (0.14 in.) it is allowed to decelerate uniformly. Acceleration may continue during a period correspondingly to 55° of cam motion, and deceleration during 27.5° of cam motion. The period of fuel delivery seldom is greater than 20° of crankshaft rotation, which is equal to 10° of camshaft rotation. The total angle of cam circumference over which the rising slope extends is chosen somewhat arbitrarily, but since it is desired to utilize only the central portion of the lift for injection purposes, it must be several times the cam-motion angle corresponding to the injection period. The greater the angular extent of the lift period, the smoother the operation of the pump cam be made; on the other hand, for a given lift, the less will be the speed of the plunger and the less the plunger travel during the fixed injection period; hence, it will be necessary to use a larger pump-barrel bore, which involves the risk of greater leakage.

We will assume that the engine for which this pump is intended is a four-cycle type and is to run at 1800 rpm, so that the pump shaft must turn at 900 rpm. The equation for distance covered by a uniformly-accelerated body is

$$d = \frac{a}{2} t^2,$$

where a is the acceleration, which in this case may be expressed in inches per second per second; and t the time in seconds, the distance d then being obtained in inches. As in our case the acceleration is to extend over 55° of cam motion and the pump shaft carrying the cam turns at 900 rpm. (= 15 rps), the time occupied by the acceleration of the pump plunger is

$$\frac{55}{360} \times \frac{1}{15} = 0.01018 \text{ sec.}$$

Since the plunger travels a total distance of 0.28 in. during the period of acceleration, by substituting in the foregoing equation we get

$$0.28 = \frac{a}{2} 0.01018^2,$$

from which it follows that

$$a = \frac{0.56}{0.01018^2} = 5400 \text{ ipsps.}$$

We next calculate the times occupied by cam motions of 5°, 10°, 15°, etc., which we can do by simple proportion, since we have already found that 55° of cam motion corresponds to 0.01018 sec. Next we determine the plunger travels corresponding to 5°, 10°, 15°, etc., of cam motion, by substituting the times corresponding to these cam motions in the equation for d. These calculations were made, and from the results obtained was drawn that portion of the plunger-travel curve in Fig. 19 which corresponds to acceleration.

The remaining portion of the plunger lift curve is of the same general form, but inverted and of different proportions. Since there is deceleration during 27.5° of cam motion, equivalent to 0.00509 sec, and the total plunger travel during this period is 0.14 in., we have

$$0.14 = \frac{a}{2} 0.00509^2,$$

from which it follows that a = 10,800 ipsps.

During the last 10° (0.00185 sec) of cam incline the plunger travel is

$$\frac{10{,}800}{2} \times 0.00185^2 = 0.0185 \text{ in.}$$

during the last 20° it is

$$\frac{10{,}800}{2} \times 0.0037^2 = 0.074 \text{ in.}$$

and during the last 30° it is

$$\frac{10{,}800}{2} \times 0.00509^2 = 0.14 \text{ in.}$$

or the full travel of the plunger during deceleration. With the aid of these figures the lift diagram Fig. 19 is completed.

Locating Injection Period on Delivery Stroke—We next have to choose that portion of the delivery stroke which we want to use for the actual injection. The injection should not extend over more than 20° of crankshaft rotation (10° of cam rotation) when the engine is running under over-load. (This relates to the theoretical or assumed duration of injection; the actual duration is somewhat greater, because injection is continued after the pump delivery valve closes, due to expansion of fuel in the line.) Under normal conditions the injection period will extend over about 16° of crankshaft rotation or 8° of cam rotation. Injection is preferably effected during the latter part of the period of accelerated motion, so that there will be a gradual increase in the pressure behind the jet. This is thought to be beneficial, at least with "quiescent" combustion chambers, because it enables the latter part of the spray to break through the zone of burnt gases resulting from combustion of the first part. If we start injection at 45° of cam motion (counted from the point at which the plunger begins to lift), it will be completed, under normal full load, at 53°.

Pump Bore Required—The next dimension to be determined is the required pump-barrel bore. Let us assume that the cylinder to be served requires—under normal full load—0.006 cu in. of fuel per cycle, which would be about right for a cylinder of 105 cu in. displacement. When the pump begins to deliver, at 45° of cam motion, its plunger is 0.1875 in. from the beginning of the stroke, and when delivery ceases, at 53°, it is 0.260 in. from the same point. Hence, the effective length of stroke is

$$0.260 - 0.1875 = 0.0725 \text{ in.}$$

From this it follows that the plunger diameter must be

$$\sqrt{\frac{0.006}{0.0725 \times 0.7854}} = 0.325 \text{ in.}$$

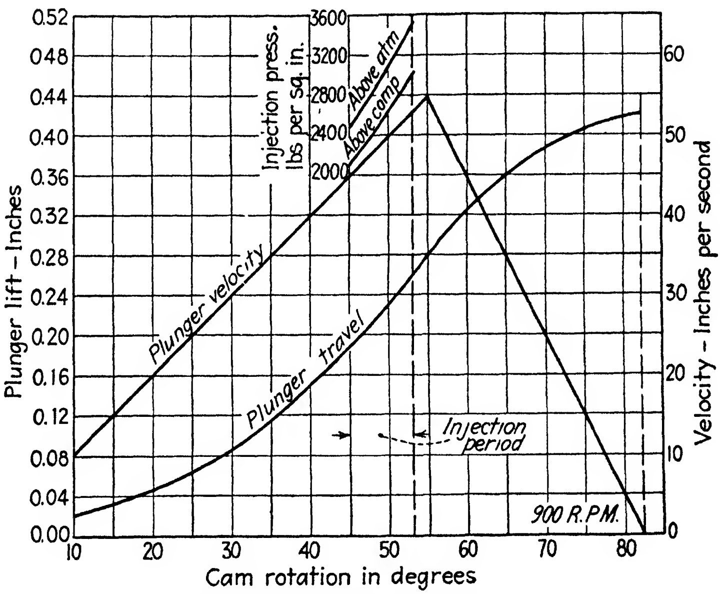


FIG. 19.—TRAVEL, VELOCITY AND PRESSURE CURVES OF INJECTION PUMP.

Nozzle Pressure—In order that the fuel may be properly atomized, the pressure on the nozzle must be adequate. This pressure, of course, depends on the velocity of flow through the nozzle orifice or orifices, which is directly proportional to the velocity of the pump plunger. Hence in this case the pressure at the nozzle will be a maximum at the moment delivery by the pump ceases. The maximum velocity of the pump plunger is

$$5400 \times 0.01018 = 55 \text{ ips.}$$

This velocity is reached at 55° camshaft angle. At 53°, when delivery is shut off under full load, the plunger speed is

$$\frac{55 \times 53}{55} = 53 \text{ ips.}$$

Now let us suppose that from a study of the shape of the combustion chamber and the characteristics of fuel sprays we have come to the conclusion that at the end of injection the effective pressure behind the spray should be 3000 psi. If in the equation for the jet velocity v on page 114 we substitute for the pressure head h its equivalent in terms of P_1, P_2, and s, and insert the numerical value of g, we get

$$v = 146.5f \sqrt{\frac{P_1 - P_2}{s}} \text{ ips,}$$

where f is the coefficient of discharge of the orifice; P_1, the pressure on the fuel; P_2, the pressure in the cylinder, and s, the specific gravity of the fuel. Let us assume that f is equal to 0.94 and s, 0.85. Then we have

$$v = 146.5 \times 0.94 \sqrt{\frac{3000}{0.85}} \doteq 8180 \text{ ips.}$$

The velocity of the pump plunger at this moment we found to be 53 ips, hence the velocity of the jet must be

$$\frac{8180}{53} = 154 \text{ times as great.}$$

This, of course, means that the area of the nozzle orifice or orifices must be 1/154 that of the pump-plunger section, and in the case of a single orifice its diameter must be $1/\sqrt{154} = 1/12.4$ that of the plunger. Since the plunger has a diameter of 0.325 in., that of the orifice must be 0.0262 in. If four orifices are used instead of a single one, then each orifice will have to have a diameter of 0.0131 in. This is more likely to give the right penetration.

Empirical Equation for Orifice Size—Values for the orifice diameter required arrived at by the above method of calculation may be checked by means of the following empirical equation, which is based on data from numerous production engines and may be said to represent average modern practice. The proper diameter of the individual orifice depends to quite an extent on the spray penetration required, that is, the distance from the nozzle orifice to the most remote point of the combustion chamber. With both the orifice diameter and the penetration given in inches, it is found that with open-combustion chambers the average practice is to make

$$\text{Orifice diameter} = \frac{\text{Log of penetration} + 0.55}{70} \text{ in.}$$

There is a simple relation also between the piston-displacement rate of the engine and the total nozzle-orifice area. The piston-displacement rate is found by multiplying the area of the piston head in sq in. by the mean piston speed in ips. For engines with an open combustion chamber the total nozzle-orifice area A in square mills (1 mill = 0.001 in.) is given by the equation

$$A = \frac{\text{Piston-displacement rate} + 1600}{18} \text{ sq mills.}$$

With pintle-type nozzles the maximum opening area of the orifice is made considerably greater, and its proper value may be found from the equation

$$A = \frac{\text{Piston-displacement rate} + 2000}{12} \text{ sq mills.}$$

Selection of Pump Dimensions—It was shown in the first part of this chapter that the volume of fuel consumed at full load by a cylinder of V cu in. displacement is approximately $V/20{,}000$ cu in. The maximum delivery capacity of the injection pump is made somewhat larger and the delivery is limited by a stop on the fuel control rack. Usually, the amount of fuel injected at full load is only from one-fourth to one-sixth the plunger displacement.

Let d be the plunger diameter; l, the length of stroke, and r, the ratio of plunger displacement to full-load delivery. We then have

$$\frac{V}{20{,}000} = \frac{\pi d^2 l}{4r}.$$

Now let us suppose that the stroke of the plunger is equal to 1.3 its diameter. Then, evidently,

$$\frac{V}{20{,}000} = \frac{1.3\pi d^3}{4r}.$$

and

$$d = \sqrt[3]{\frac{Vr}{20{,}400}} \text{ in.}$$

The value of r is determined by the injection angle. It is obvious that for a given quantity of fuel injected, the greater the injection angle, the smaller will be the diameter of the pump plunger, and vice versa. The present tendency seems to be to reduce the injection angle. This reduces the specific fuel consumption (provided the timing is right), but it also increases the injection pressure. The nominal injection angle is given by the equation

$$\alpha = \frac{\text{Full-load injection quantity in cu mills}}{\text{Cam rate (mills per crankshaft deg)} \times \text{plunger area (sq mills)}}.$$

Applying the above equations to the injection system calculated in the foregoing, we obtain the following results: Penetration of jet, $2\frac{5}{16}$ in.; nozzle-orifice area required (assuming the engine to have cylinders of 4.80-in. bore by 7.00-in. stroke = 126.6 cu in. displacement), 511 sq mills (instead of 539, the actual area of four orifices of 0.0131 in. diameter each). With a stroke/bore ratio of 1.3 for the injection pump and a full-load fuel charge equal to one-sixth the plunger displacement ($r = 6$), the plunger diameter required figures out to 0.328 in. Finally, the equation for the theoretical duration of injection gives 16 crankshaft degrees.

If an injection nozzle with orifices of the size calculated is used, the excess of pressure on the nozzle over the compression pressure into which injection takes place, at the end of the injection period, will be 3000 psi, and with a compression pressure of 500 psi the actual pressure on the fuel will be 3500 psi. The pressure excess on the nozzle varies as the square of the plunger velocity, and since the graph of plunger velocity has been already determined, it is an easy matter to find the injection pressure for any point of the injection period. Values of the pressure throughout the injection period were calculated and plotted in the upper part of Fig. 19.

Variation of Injection Pressure with Orifice Size—The need for a high degree of accuracy in the orifice diameter may be easily shown. With fixed plunger speed the velocity of flow through the nozzle orifice varies as the square of the orifice diameter, and the excess pressure at the nozzle varies as the square of the velocity of flow through the orifice; consequently, the nozzle pressure varies as the fourth power of the orifice diameter. An error of only 5 per cent in the diameter of the orifice causes a variation of about 23 per cent in the nozzle pressure.

According to Dr. Schweitzer, the most practical method of accurately determining the diameter of the orifice is by means of a low-power microscope. Every shop which has anything to do with Diesel nozzles, he says, should have a microscope of 20-30 magnification. This serves not only to accurately determine the diameter of the hole, but also for verification of its shape.

Graphical Method—If the pump cam is of such shape that it does not produce plunger acceleration according to some simple law, then the plunger speed for different points of the effective stroke must be determined graphically. First of all a diagram of plunger travel is drawn, angular motions of the cam being plotted as abscissas and corresponding plunger travels from the beginning of the stroke as ordinates. The data for such a diagram can be obtained graphically by laying off the cam mechanism to an enlarged scale, with the cam successively in different angular positions. From the plunger-travel curve thus obtained a plunger-velocity curve is derived.

Let us take any point P on the plunger-travel curve (Fig. 20) of ordinate y (in. of plunger travel) and abscissa x (degrees of cam motion). Let unit distance along the vertical axis represent b in. plunger travel, and unit distance along the horizontal axis a degrees of cam motion. Then the actual distance of the point on the plunger-travel curve from the horizontal axis is y/b units, and its distance from the vertical axis, x/a units. We then have

$$\tan \phi = \frac{d(y/b)}{d(x/a)},$$

where ϕ is the angle of the tangent to the travel curve at point P with the horizontal. Simplifying, we have

$$\tan \phi = \frac{a}{b}\frac{dy}{dx}.$$

Now, the velocity of plunger travel is dy/dt, where t is the time. Since the velocity of the cam is constant,

$$\frac{dt}{dx} = \frac{t}{x} = \frac{1}{360N} \text{ minute} \quad \text{or} \quad \frac{60}{360N} = \frac{1}{6N} \text{ second},$$

where N is the number of cam revolutions per minute. Hence,

$$dx = 6N\,dt.$$

Substituting this value of dx in the equation for $\tan \phi$ we get

$$\tan \phi = \frac{a}{6Nb}\frac{dy}{dt},$$

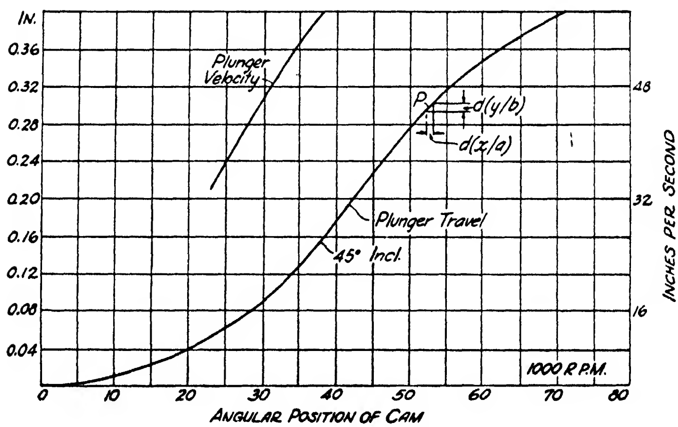


FIG. 20.—DIAGRAM FOR DETERMINING PLUNGER SPEEDS GRAPHICALLY.

from which it follows that the plunger velocity

$$v = \frac{dy}{dt} = \frac{6Nb}{a} \tan \phi \text{ ips.}$$

In Fig. 20 the divisions of the horizontal and vertical axes are equal; $b = 0.04$ in. and $a = 5°$. With a pump-shaft speed of 1000 rpm we therefore have for the point of the plunger-travel curve for which $\phi = 45°$ and $\tan \phi = 1.00$,

$$v = \frac{6 \times 1000 \times 0.040}{5} \times 1.00 = 48 \text{ ips.}$$

For other points of the plunger-travel curve the velocities are found in the same way.

Points in Pump Design—All parts of injection pumps must be designed so as to be comparatively rigid, so they will not deflect unduly under the fluid pressure on the plunger during the delivery stroke, which would affect the delivery. The pumps must be so designed, moreover, that no air pockets can form in them. This means that the delivery valve must be at the highest point and that any liquid or gaseous contents of the pump chamber can reach the delivery valve by a continuous upward movement. It is necessary to guard against air traps particularly in the case of pumps for automotive engines, owing to the fact that the fuel in the tanks of road vehicles always absorbs a certain amount of air, due to surging and splashing in the tanks.

Injection pumps are being manufactured in single- or two-element units to be flange-mounted on engines where the cams and tappets necessary for operating the pump plungers are incorporated in the engine; and in units of three or more pump elements for bracket-mounting or for flange-mounting to the housing of the front-end drive. Some multi-unit pumps have provision for mounting the transfer pump directly on them, while in the case of others that pump must be separately mounted. In pumps for multi-cylinder engines the cams should be arranged in such order that injection strokes follow in the same order as the firing strokes of the engine, so the fuel lines to the injectors need not be crossed.

Pump-Barrel Leakage—An investigation of the relative importance of leakage in injection pumps was made by A. M. Rothrock and E. T. Marsh (N.A.C.A. Report No. 477). Since the leakage is inversely proportional to the viscosity of the fuel, the subject is a particularly vital one when injection pumps are used to inject gasoline into spark-ignition engines (airplane engines). One reason why leakage past the pump plunger is objectionable is that it may lead to unequal distribution of fuel charges to the various cylinders of an engine.

It was found that the rate of leakage (expressed in weight/time units) between a lapped plunger and a sleeve varies directly as the density of the fuel, the pressure producing the leakage, the diameter of the plunger, and as the cube of the mean clearance between plunger and sleeve. The rate of leakage varies inversely as the length of the lapped fit and inversely as the viscosity of the fuel in absolute units. Because of these relationships, the length of the lapped fit is of minor importance as compared with the clearance. Provided the clearance is not greater than 0.0001 in., the leakage past the plunger will be limited to between 0.01 and 0.2 per cent of the fuel injected, depending on the viscosity. A length/diameter ratio of 3-5 is generally sufficient for the lapped fit.

It must be remembered in this connection that in operation the effective clearance will be affected by contraction of the plunger and expansion of the sleeve under the fuel pressure. This effect can be minimized by keeping the plunger diameter down and increasing the wall thickness of the sleeve, or by making the plunger hollow so the fuel pressure on the inside will tend to expand it. By maintaining fuel pressure also on the outside of the sleeve, it is possible to cause the clearance to decrease in operation.

CHAPTER VII

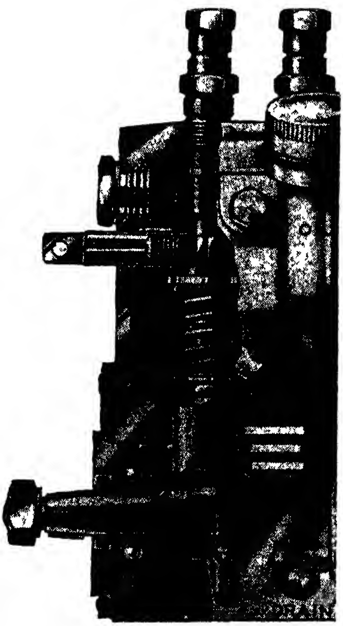
Modern Injection Pumps and Governors

Previous to the advent of the high-speed type of Diesel engine, practically every manufacturer built his own injection equipment. In the automobile industry such components as carburetors, ignition units, brakes, and friction clutches are produced almost exclusively by parts makers, and a similar parts and accessories industry has now grown up in the Diesel field. It is apparent that the production of such parts as injection pumps and spray nozzles calls for different skills and production equipment than the machining of engine blocks, crankshafts, etc. Among the difficult problems that arise may be mentioned the lapping of plungers and sleeves with a clearance not to exceed 0.0001 in., and the accurate drilling of holes of such small diameters as 0.006 and 0.008 in., in the spray nozzles. On the latter problem manufacturers of injection equipment have had some help from firms experienced in making dies for the manufacture of rayon.

A pioneer in the Diesel parts industry was the Robert Bosch Company of Stuttgart, Germany, which entered the field in 1926, and early in 1951 turned out its one-millionth injection pump. The Bosch company established branches and subsidiaries in different countries, which have since been succeeded by independent firms, including the American Bosch Corporation in the United States, C.A.V. Ltd. in Great Britain, and Lavalette & Co. in France. While most makes of high-speed Diesels on the market now carry injection pumps and spray nozzles manufactured by specialists, a number of large engine manufacturers produce their own injection equipment. One argument sometimes advanced in favor of the latter practice is that it obviates divided responsibility toward the final consumer, the engine operator.

In this chapter will be described various items of injection equipment other than spray nozzles (which have been dealt with already), and different types of engine-speed and injection-timing governors.

American Bosch Pumps—American Bosch fuel-injection pumps (American Bosch Corporation, Springfield, Mass.) are of the type having plunger-controlled ports. A side elevation of the driven end, partly in section, and a cross section of the APE-A and APE-B pumps are shown in Figs. 1 and 2 respectively. These pumps come with either aluminum-alloy or cast-iron housings. The housings comprise a camshaft compartment, a plunger-and-spring compartment, and a header section. The pumps are available with either a machined base for mounting on an engine bracket, or with an integral mounting flange at the driving end. A mechanical or a pneumatic governor can be mounted directly on the pump;



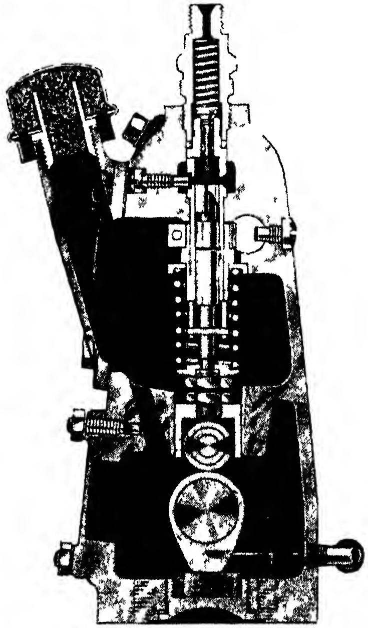


FIG. 1 (*Left*).—SECTIONED SIDE VIEW OF AMERICAN BOSCH PUMP.
FIG. 2 (*Right*).—CROSS SECTION OF AMERICAN BOSCH APE PUMP.

where a mechanical governor is so fitted it is advisable to use the flange mounting for the pump. These pumps are of the in-line type and are made with from one to eight cylinders.

The lower compartment houses the camshaft and forms an oil reservoir. Plugs in the base contain oil-saturated felt pads with which the cam lobes come in contact during every revolution of the camshaft. The camshaft compartment has end plates into which the ball bearings and oil seals for the camshaft are fitted. A fuel transfer pump, with or without hand primer, can be mounted on the outer side of the housing and is operated either by one of the fuel-pump cams or by a separate eccentric on the camshaft. (See Fig. 3.)

Referring to the sectional view, the plunger-and-spring compartment is separated from the camshaft compartment by a heavy bridge containing the bores for the tappet-and-roller assemblies. Adjustable tappet screws make direct contact with the plungers. The plunger-and-barrel assemblies extend into this compartment, and each plunger is provided with a control sleeve and a cadmium-plated return spring. Into slots

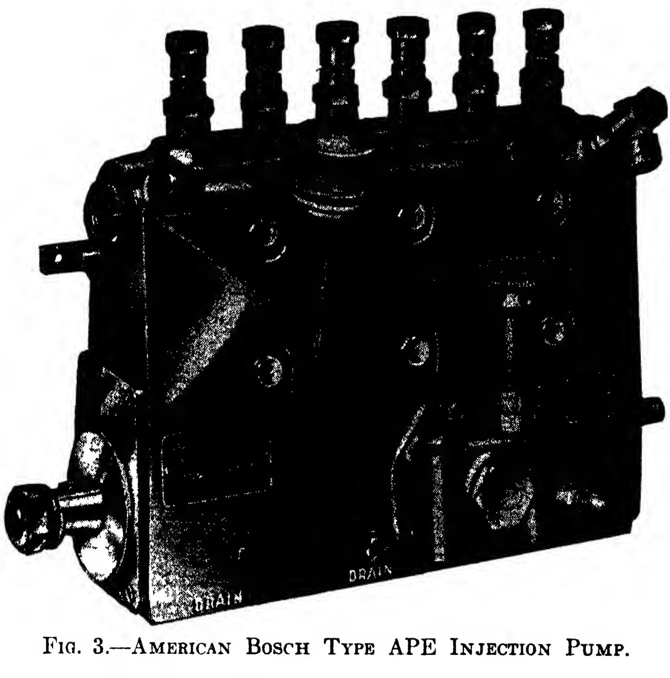


FIG. 3.—AMERICAN BOSCH TYPE APE INJECTION PUMP.

at the lower end of the control sleeve engages a crosshead carried by the plunger. The adjustable gear segment on the control sleeve is in mesh with a control rack. A large opening in the wall of the compartment provides ready access for calibration and timing.

The header section contains the fuel sump and is tapped for the delivery-valve holders. A shoulder on the pump barrel rests on a seat just below the sump cavity. The barrel head, with inlet and bypass ports, projects through the fuel sump and is located angularly by a locating screw. Lapped surfaces of the barrel head and the delivery-valve assembly are held together and sealed against leakage by a copper-asbestos gasket which the delivery-valve holder holds in place on top of the valve-body flange. The outlet end of the holder is threaded to receive a union nut securing the fuel tube.

Plungers can be provided with various helix arrangements. They may have either an upper, a lower, or both an upper and a lower helix, and the helices may be either right-hand or left-hand. A plunger with a lower helix gives a constant beginning and a variable end of injection. During the up-stroke the barrel ports are closed by the top edge of the horizontal plunger head, except when the plunger is in the "no-delivery" position. A plunger with an upper helix gives a variable beginning and a constant end of injection. With this arrangement, during the up-stroke, the barrel ports are closed by the helix and opened by the horizontal upper edge of the annular groove. A plunger with both an upper and a lower helix gives a variable beginning and also a variable end of injection. The preferred method of timing the pumps is by port closing, though under special conditions timing can be based on port opening. The tolerances on the timing are plus and minus ½ camshaft degree.

The cover over the opening in the plunger-and-spring compartment incorporates a combination lubricating oil filler and breather cap. The camshaft compartment should always be kept filled with S.A.E. 30 engine oil. It is recommended that the oil be changed whenever the engine oil is changed, or at least after 200 hours of operation. Plungers and barrels are adequately lubricated by the fuel oil. The oil level in the camshaft compartment can be checked by means of a cock. In operation the proper level is maintained automatically by an overflow fitting. A drain plug is located near the lower front edge of the compartment.

Delivery valves of these pumps are provided with retraction pistons which enter a bore in the valve body when the valve seats and give line-pressure relief on the principle explained in the preceding chapter. The manufacturer states that under full-load conditions the residual pressure in the lines should not be more than 60 per cent of the nozzle-opening pressure, and not less than 200 psi. At part load and idling the residual pressure will be correspondingly less, and in some cases it will be *nil*.

The quantity of fuel delivered per stroke is controlled by means of a rack which extends lengthwise through the pump and meshes with the gear segments of the individual pump plungers. The end of the control rack projects from the pump housing and is linked to the governor. A locating screw entering a slot in the control rack limits the travel of the latter. When the rack is at the end of its travel toward the governor, it is in the "no-delivery" or "stop" position. As it moves away from this position, fuel delivery begins after a few millimeters of "dead travel," and thereafter the delivery increases in proportion to the travel.

Base-mounted pumps should be driven from the engine through an adjustable, non-flexible Oldham coupling. Such a coupling requires the pump shaft and driving shaft to be parallel, but takes care of misalignment between them up to 0.010 in. There are timing reference lines on the protruding part of the camshaft and on the end plate, by means of which the pump can be properly timed.

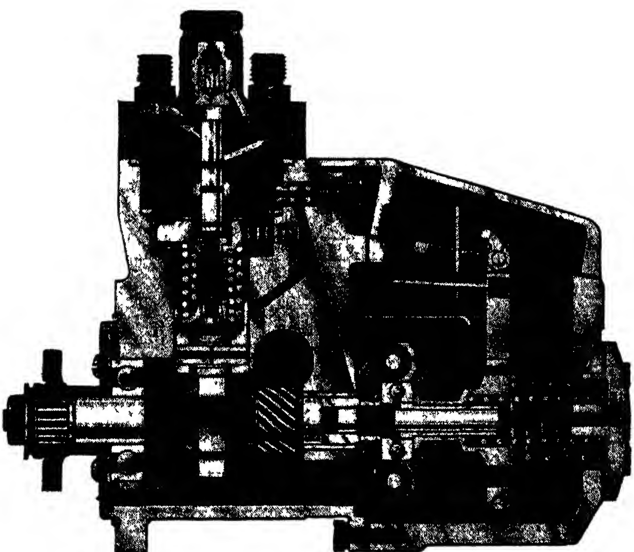


Fig. 4.—American Bosch PSB Single-Plunger Injection Pump.

Single-Plunger Pump for Multi-Cylinder Engine—In 1948 the American Bosch Corporation brought out a single-plunger injection pump for use on four- and six-cylinder, four-stroke engines, Model PSA. It was designed with a view to economical production and maintenance, and was of the cam-actuated, constant-stroke, throttled-intake type. It was to be driven at crankshaft speed. A continuous rotary motion was imparted to the plunger by a pair of bevel gears, of which the driving member was integral with the camshaft, and the plunger

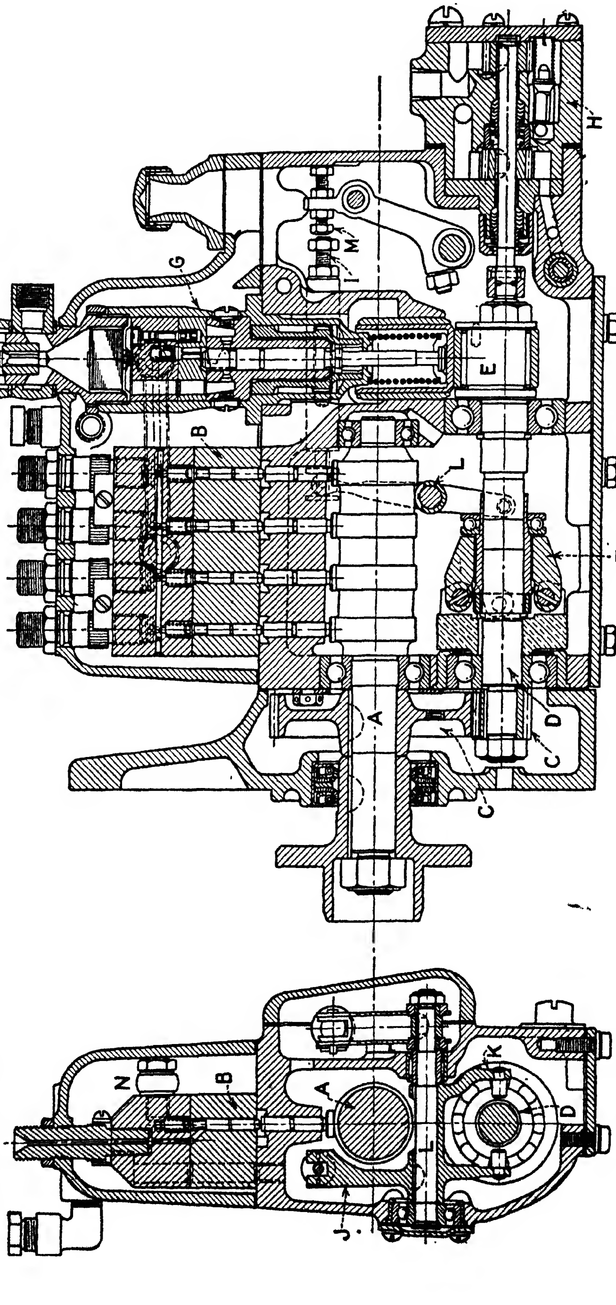


FIG. 5.—I.H.C. SINGLE-PLUNGER INJECTOR PUMP FOR FOUR-CYLINDER ENGINE.

A, distributor-valve shaft; B, distributor block; C, C, governor-shaft drive gears; D, governor shaft; E, eccentric of single-plunger unit; F, governor; G, single-plunger unit; H, primary pump; I, quantity-control rack; J, upwardly-extending arm of governor fork; K, governor fork; L, control shaft; M, adjustable fuel stop; N, high-pressure fuel line.

served also as a distributing valve. The following year Model PSA was succeeded by Model PSB, of which a sectional view is shown in Fig. 4. This, also, is of the constant-stroke type, with one plunger and one delivery valve. It differs from the PSA in that a different type of gearing is used to rotate the plunger (see illustration), in that the operating mechanism incorporates a roller follower, and in that the delivery is controlled by a high-pressure sleeve on the plunger instead of by a throttled intake. This pump, moreover, can be equipped with a governor which not only limits the maximum speed, but also permits control of the injection quantity at any speed. While primarily designed to be driven at crankshaft speed, it can be arranged to be driven at camshaft speed by doubling the number of cam lobes and suitably changing the internal gearing.

I.H.C. Pump—The I.H.C. four-cylinder, single-plunger injection pump of the International Harvester Company, of which two sectional views are shown in Fig. 5, is flange-mounted on the crankcase front plate. It is a distributor-valve type of pump and uses a single eccentric to operate a lapped plunger at constant stroke. The unit is completely enclosed in a housing of alloy cast iron, provided with cover plates. A camshaft in the upper half of the housing operates four distributor valves in the distributor-valve block. The eccentric which operates the single plunger is formed on the rear of the governor shaft, and the removable single-plunger unit is mounted on the top face of the pump housing, to the rear of the distributor block.

A combination primary and scavenging pump (helical-gear type) is mounted on the rear end of the pump housing, and is driven by the governor shaft, which latter is driven from the distributor camshaft through a pair of spur gears with a 4 to 1 ratio, located outside the pump housing at the front. These gears are enclosed by the pump mounting flange. The hub of the pump drive gear is secured to the distributor camshaft. Distributor unit, plunger unit, and primary pump are separate assemblies which, if necessary, can be installed on the injection pump while it is in place on the engine, without interfering with other units already assembled thereon.

Operation of Pump—After passing through the auxiliary fuel filter, the fuel is forced by the primary pump (transfer pump) through the final fuel filter into the plunger unit and through a 0.002-in. strainer into a reservoir surrounding the upper part of the plunger bushing. The primary pump is provided with a by-pass valve which maintains a pactically constant pressure on the final fuel filter. A fuel-return check valve, which maintains a practically constant pressure on the injection pump inlet, is located on top of the plunger unit.

The single injection pump operates on the same principle as other pumps with plunger-controlled ports, and its mode of operation, therefore, does not need to be described. It delivers fuel through the high-pressure pipe to the distributor block. There the fuel passes through one of the four distributor valves (which has been lifted off its seat by the camshaft) into the injection pipe, and it is finally discharged through the injection nozzle into the combustion chamber of the engine.

Line-pressure relief between injections is a feature of this pump. A delivery valve and reverse-check-valve unit is located above the plunger bushing. As soon as the port is opened and the delivery valve has seated, the reverse check valve opens, because it has a lower opening pressure than the nozzle valve. Fuel from the high-pressure line then flows back through the reverse check valve and through a port to the pump chamber, which is now open to the reservoir. This pressure relief allows the nozzle valve to snap to its seat, thus ensuring a sharp cutoff and preventing dribbling. The cam then allows the distributor valve to seat. The injection-pump unit goes through a complete cycle for each distributor-valve lift or opening. Distributor valves open in the order 1-3-4-2 during one revolution of the injection-pump camshaft.

Fuel injection is under the control of the centrifugal governor, mounted on the governor shaft below the camshaft in the pump housing. The centrifugal weights of the governor act through a governor fork on the control rack of the pump. A motion of $\frac{1}{16}$ in. of the control rack from the shut-off position is required before fuel injection begins. From that point on the injection quantity increases in direct proportion to the motion of the rack.

Fuel by-passed through the fuel-return check valve passes through a fuel-return pipe on the back of the pump and down through drilled passages in the pump housing to the scavenging pump. Any leakage fuel from the plunger and distributor units returns to the scavenging pump through a filter and scavenging valve located at the lower rear in the pump housing, and is pumped back to the fuel tank together with the by-passed fuel. The governor mechanism will be described in detail farther along in this chapter.

Single-Unit Pumps—Some engines are fitted with a separate injection pump for each cylinder, which is mounted over an opening in the engine crankcase directly over the camshaft, and operated by a cam on that shaft. A pump of this type, manufactured by Scintilla Magneto Division of Bendix Aviation Corp., Sidney, N. Y., is shown in Fig. 6. A tappet in a guide on the crankcase contacts the follower of the pump. Adjustments are made by the adjusting screw of the tappet, or in the case of non-adjustable tappets, by means of shims under the mounting flange of the pump. In multi-cylinder engines equipped with such pumps, the maximum combustion pressures of individual cylinders can be equalized by varying the beginning of injection. This can be accomplished by adjusting either the angular position of the injection cam,

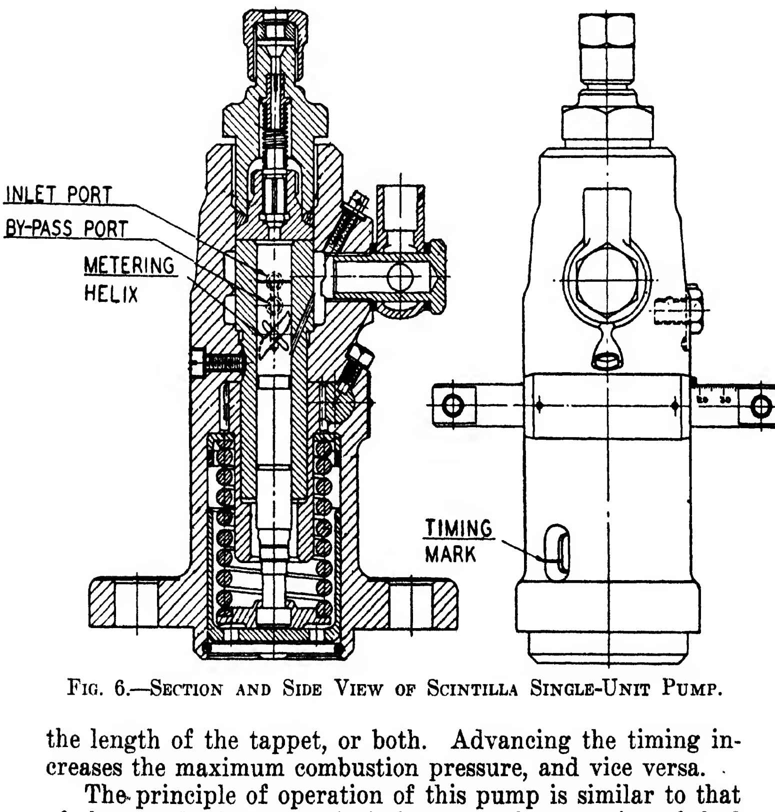


FIG. 6.—SECTION AND SIDE VIEW OF SCINTILLA SINGLE-UNIT PUMP.

the length of the tappet, or both. Advancing the timing increases the maximum combustion pressure, and vice versa.

The principle of operation of this pump is similar to that of the standard multi-unit jerk pump, the quantity of fuel injected being controlled by turning the pump plunger around its axis by means of a rack. This control rack is connected to the governor and the engine controls.

A single-unit injection pump with injection control by means of a spill valve is being manufactured by Aircraft &

Diesel Equipment Corporation of Chicago, and a sectional view thereof is shown in Fig. 7. Plunger *6* and barrel *7* form a separate assembly which is secured in the pump body by cap screws. The discharge valve *18* comprises the discharge valve, its guide, a spring, a valve stop, and a nipple. It is said to be of such design as to permit accurate control of line

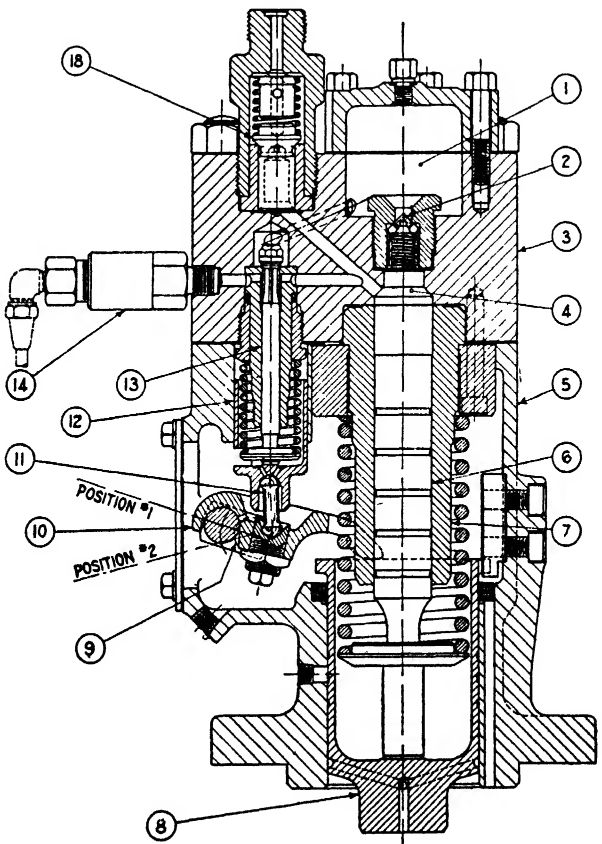


FIG. 7.—DECO SINGLE-UNIT PUMP WITH SPILL VALVE.

relief with sturdy parts. The spill-valve assembly *13* consists of the spill valve proper and its guide. This valve is hydraulically balanced, to minimize its reaction on the control shaft. Motion is transmitted from pump plunger *6* to spill valve *13* through rocker *10* and ball link *11*. The fulcrum of the rocker is on control shaft *9*, and is provided with a screw adjustment for the clearance between the tappet and the valve. Pump housing *5* contains the spring cup *8*, which serves to relieve plunger *6* of side thrust. The control shaft, a single-throw crankshaft, is carried in needle bearings in the housing, and is connected to the governor and engine control by a linkage. With the control shaft in the "stop" position (position *1*), the spill valve begins to open at the same time the pump plunger begins its delivery stroke, and no fuel is delivered. With the control shaft in the "full-power" position (position *2*), the clearance between the spill valve and its tappet is a maximum, and the valve lifts only after a full charge has been delivered. All working parts of this pump are hardened and ground or lapped.

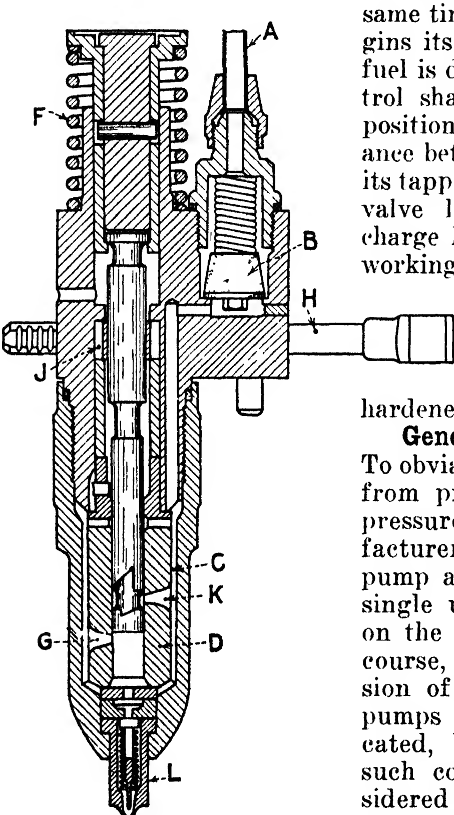


FIG. 8.—GENERAL MOTORS INJECTOR.

General Motors Injector—To obviate the difficulties arising from pressure waves in high-pressure fuel lines, some manufacturers combine the injection pump and injector nozzle in a single unit, which is mounted on the cylinder head. This, of course, renders the transmission of power to the injector pumps somewhat more complicated, but the advantages of such combined units are considered to outweigh their disadvantages.

In the General Motors injector, shown in section in Fig. 8, fuel is circulated through all of the units continuously by the transfer pump, which not only helps to cool the injectors but makes it impossible for air locks to develop, even if the tank should run dry. Cooling of the injectors is further facilitated by the method of installation, the unit being inserted into a drawn shell that extends through the cylinder head and is completely surrounded by cooling water.

Referring to Fig. 8, fuel enters at connection *A* and passes through filter *B* into annular chamber *C* surrounding pump barrel *D*. The pump plunger is forced down by means of a mechanism comprising a cam, tappet and rocker lever, and is returned by spring *F*. During the return stroke or suction stroke, fuel enters the pump chamber through inlet port *G*. The pump plunger is undercut on its cylindrical surface near its lower end, and the chamber formed by the undercut communicates with the pump chamber through a longitudinal slot in the plunger. The undercut has inclined or helical upper and lower edges, and the plunger can be turned around its axis by means of a rack *H* and pinion *J*.

Shortly after the beginning of the down-stroke, inlet port *G* is closed by the plunger. Injection begins when spill port *K* is closed by the upper edge of the undercut, and ceases when inlet port *G* is uncovered by the lower edge of the undercut. The fuel displaced by the plunger is injected into the combustion chamber through injector *L*, which has six orifices of 0.006 in. diameter. As the pressure for injection is generated close to the orifices, there are no pressure waves in the system, and a pressure-cam angle diagram from this injection equipment is comparatively smooth. Measurements have shown that at 780 rpm the peak pressure during injection is about 28,000 psi, and that this pressure varies substantially as $n^{1.25}$, where n is the pump speed.

Oil-Fog Injection—In all of the early successful Diesel engines the fuel was injected into the combustion chamber by means of a blast of compressed air. This ensured good distribution throughout the air charge, but the need for a compressor capable of compressing injection air to between 1000 and 1200 psi was a disadvantage, and engineers naturally tried to get away from the system. About 1912 the so-called solid-injection system was introduced, which is in use today on the great majority of high-speed engines. At first the efficiency of combustion was not nearly as good with solid as with air injection, and the author remembers hearing a sea captain remark at an engineering meeting that at sea he always could pick out the motorships with solid injection, as they invariably left a trail of smoke behind. Inventors therefore tried to provide means for injecting the fuel in a manner similar to that with compressed air, but eliminating the need for a compressor.

One scheme tried consisted in storing some of the gases of combustion in a chamber within the engine, and admitting it in lieu of compressed air to the fuel passage at the moment of injection. Little success was achieved along this line. In another type of engine a steel fuel cup was arranged in the cylinder below the head, with a number of orifices in its wall through which it communicated with the combustion chamber. A mixture of fuel and air was drawn into this cup during the suction stroke. Some of the fuel passed through the orifices into the combustion chamber, and ignition undoubtedly took place there toward the end of the compression stroke, but the flame was communicated to the fuel cup, and because of the much greater richness of the mixture in it, the pressure rose more rapidly there, and the excess pressure in the cup blew the remaining fuel into the combustion chamber. Engines making use of this principle (Brons and Hvid) were in regular production in Europe and this country for a number of years during the period of World War I, but their manufacture was later discontinued. One of their weaknesses was that the orifices in the cup frequently became choked with carbon.

Another system, which is in a sense a development of the one described in the foregoing, consists in forming an air-oil mixture (or an oil fog) in a small auxiliary cylinder communicating with the combustion chamber, and forcing it into the combustion chamber at the proper time by means of a piston in the auxiliary cylinder. This system evidently is intermediate between air injection and solid injection, and may be called oil-fog injection. It is made use of in the Cummins Diesel engine, which was the pioneer automotive-type Diesel on the American market.

Cummins Injection System—The Cummins injection system comprises a transfer pump of the gear type which delivers fuel under a pressure of 50-60 psi to a single plunger pump serving all cylinders of the engine. This pump is combined with a rotary distributing valve which places its barrel in communication with the transfer pump during the suction stroke, and with the fuel injector of a cylinder during its delivery stroke, the various cylinders being served in rotation. Fig. 9 is a diagram of the Cummins fuel-pump system. The single plunger pump is operated by means of a cam having as many lobes as there are cylinders to be supplied. Control of the injection quantity is effected by varying the stroke of the pump, this being accomplished by varying the point of contact between a strut swiveled to the pump plunger and a rocker lever serving as cam follower, the strut having a link connection to the governor. The pump requires no valves, as the distributor places its barrel in communication successively with the fuel transfer pump and the injector valve of the cylinder to be supplied. Thus the distributor serves the valve functions of the pump. Fuel is supplied to the metering pump under a sufficiently high pressure to ensure that its pump chamber is completely filled during each suction stroke. Owing to the fact that a single pump supplies all of the engine cylinders, uniform distribution is practically assured.

Injector Mechanism—The Cummins injector (Fig. 10) consists of a malleable iron casting forming a cylinder, with a properly-fitted plunger actuated by the camshaft in the crankcase through a pushrod extending up the side of the

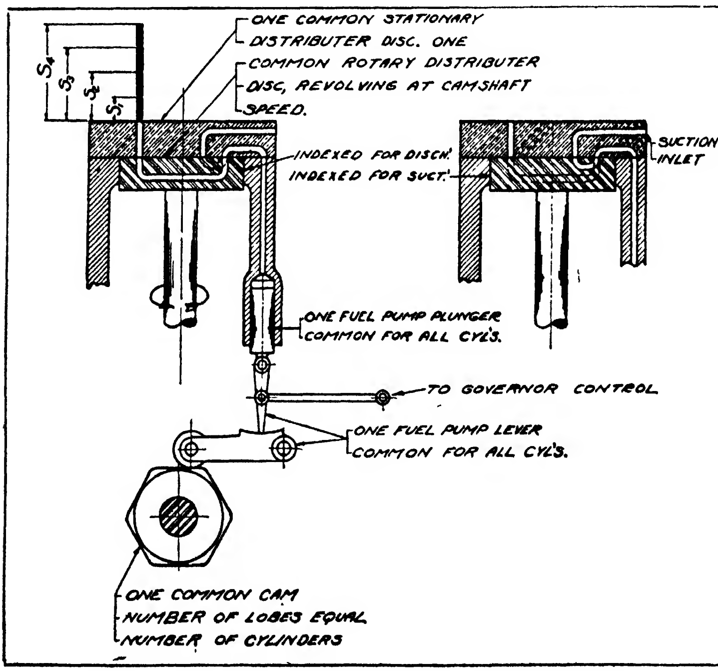


Fig. 9.—Diagram of Cummins Fuel Pump System.

engine, and a rocker lever. Fitted over the lower end of the injector is a nozzle cap with a conical bore into which the point of the plunger can descend. The plunger is held in its topmost position by a spring, and there is then a space in the nozzle cap below the plunger which is much larger than the volume of a single fuel charge. From this space there is a fuel passage leading to the fuel connection of the injector. There is one check valve in the fuel passage in the injector and another in the connector fitting.

Cycle of Operations—During the compression stroke of the engine the metering pump forces a fuel charge of the correct amount for the momentary load on the engine to the injector. Since the fuel line and the passage in the injector were already completely filled, any fuel added at the pump end of the line will force an equal amount into the space below the plunger of the injector. During its passage through the injector, and especially while in the cylinder of the injector, the fuel absorbs considerable heat. Although the injector is open to the combustion chamber, none of the fuel can leak into it, partly because the orifices of the injector are very small, but mainly because during the time the fuel is being forced into the chamber by the metering pump, the plunger of the injector is moving upward in it, and is therefore producing a suction effect which draws air from the combustion chamber through the orifices.

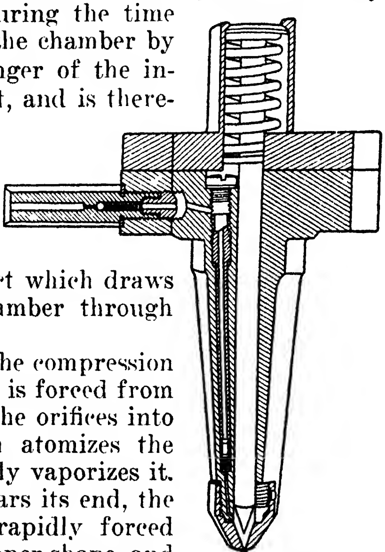


FIG. 10.—CUMMINS FUEL INJECTOR.

During the latter part of the compression stroke, heated, compressed air is forced from the engine cylinder through the orifices into the injector chamber, which atomizes the fuel charge, and at least partly vaporizes it. As the compression stroke nears its end, the plunger of the injector is rapidly forced down in it by a cam of the proper shape, and practically the whole of the air-fuel mixture is forced from the injector chamber through the orifices into the combustion chamber, where it ignites spontaneously.

Cummins Double-Disc Metering and Distributing Unit—In addition to the metering and distributing unit described in the foregoing, Cummins now also has in production a "DD" pump which, while operating on the same general principles, embodies a number of refinements. It weighs only about one-third as much as the earlier design, aluminum alloys being used wherever possible, and the whole design is worked out with an eye to compactness and low weight. The principal difference compared with the earlier model is that the metering pump now has two disc valves, one for suction and one for delivery, where a single valve formerly served both functions. A flow diagram of the system is shown in Fig. 11. The complete unit comprises five subassemblies—a camshaft,

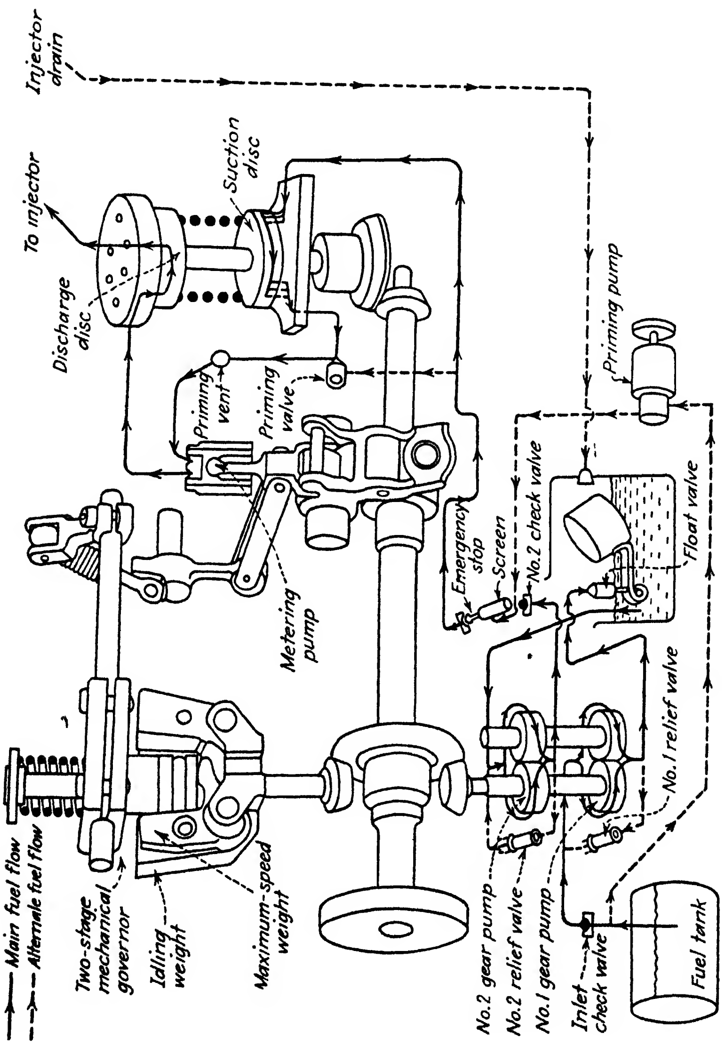


FIG. 11.—FLOW DIAGRAM OF CUMMINS DOUBLE-DISC METERING AND DISTRIBUTING UNIT.

a distributor, a metering pump, a governor, and a pair of feed pumps.

In the new unit the camshaft turns at crankshaft speed and carries a three-lobed cam (for a six-cylinder, four-stroke engine). The metering is controlled by the governor or the hand control in the same way as in the earlier unit already described. The governor is of the two-stage type and connects to the strut of the metering pump through a linkage. With this pump fuel is distributed under a pressure of 120 psi.

Referring to Fig. 11, two gear-type feed pumps, each provided with a pressure-relief valve, are driven from the camshaft through the same bevel gear as the governor. During shutdown periods the fuel lines are kept filled with fuel by check valves in the feed-pump inlet line and in the cover of the feed pumps. An emergency stop valve, enclosing a fine-mesh screen, is inserted in the line from the feed pump to the distributor.

In the fuel-flow diagram, solid lines indicate the main path of flow, dashed lines the alternate or bypass path. Fuel from the supply tank is drawn into gear pump No. 1 through the check valve in the line, and is delivered by the pump into the float chamber through a float valve. When the float chamber is full and its valve closed, the fuel moved by pump No. 1 escapes through its relief valve. No. 2 pump draws fuel from the float chamber and delivers it under pressure through a fine-mesh screen and through the emergency stop valve to the suction disc and plate. The suction disc registers with the plate at the proper time to allow the fuel to pass on to the metering pump during the suction stroke of the latter. In between suction strokes the fuel delivered by pump No. 2 escapes through its relief valve. When the engine is not running, No. 2 check valve keeps the system primed.

When the priming valve is closed and the priming vent open, oil delivered by the priming pump bypasses the gear pump and primes the suction disc. When the priming valve is open, the priming fuel bypasses the suction disc and passes through the metering pump to the distributing disc, and through the distributor successively into the fuel lines to the different cylinders. A small amount of fuel drains back from the injectors to the float chamber by way of the injector drain manifold and connecting line.

Atlas Pilot-Injection System—The pilot-injection system, to which reference was made in Chapter III, had its first extensive application in 1949, when it was adopted for use on the bus engines of the London Passenger-Transport Board. The engines of these buses were of the open-chamber or direct-injection type, which have a tendency to be rough in operation because of their high combustion pressures. The system adopted was one developed by the Atlas Diesel Company of

Stockholm, Sweden. As shown in Fig. 12, the cam of the fuel pump has a variable slope. At first the pump plunger is raised gradually, and a small, fixed quantity of fuel is injected at a low rate, so that when ignition occurs there is only little fuel in the combustion chamber, and detonation is prevented. The pilot charge, which is just sufficient to keep the engine idling, is constant at all loads. The two parts of the cam up-slope are joined by a smooth curve, so that there is no interruption in the injection.

With a conventional injector the nozzle opening pressure is greater than the closing pressure, because when the valve is open the fuel presses against an area equal to the full cross section of the valve, whereas when the valve is closed, it presses against a smaller, annular area, and in each case the total fluid pressure must equal the spring pressure before valve motion can take place. A fairly high closing pressure is needed to ensure a sharp cutoff and prevent after-dripping, and if in the Atlas system the opening pressure were higher than the required closing pressure, the valve might not open under the relatively low pressure generated by the gentle slope of the pilot section of the cam. The system therefore is so designed that the opening pressure of the injector is less than its closing pressure.

Referring to Fig. 13, it will be seen that the valve comprises three sections of different diameters, and that two annular surfaces on it are acted upon when the line pressure is increasing during the delivery stroke of the pump. There is a small check valve in the passage connecting the lower fuel chamber to the fuel-supply line, which prevents the pressure of combustion from forcing fuel from this chamber back into the supply line toward the end of the injection period. As the line pressure increases during the injection stroke, it acts on both of the annular surfaces, and the valve opens at about 1000 psi. At the end of the injection period the pressure in the supply line drops, and the check valve in the injector closes rapidly. The injector spring then works only against the fuel pressure on the larger one of the annular surfaces on the valve, which enables it to close the valve against a higher pressure, thereby preventing after-dripping.

As a further precaution against after-dripping, the pump delivery valve, shown in section in Fig. 14, is so designed that at the end of injection it meters fuel back into the pump and unloads the fuel line to the injector. This is claimed to have the additional effect of preventing disturbances in the injection of other cylinders.

Equipment for pilot injection similar to that just described is now being manufactured in Great Britain by C.A.V. Ltd.

Slowing Down Injection at Start—Effects similar to those from pilot injection are obtained from a number of other injection systems. For instance, the throttling nozzle, described and illustrated in Chapter V, which was introduced by the Bosch company about 1934, is so designed that the area of the annular opening between pintle and nozzle body increases gradually as the nozzle valve lifts. Consequently, the flow is

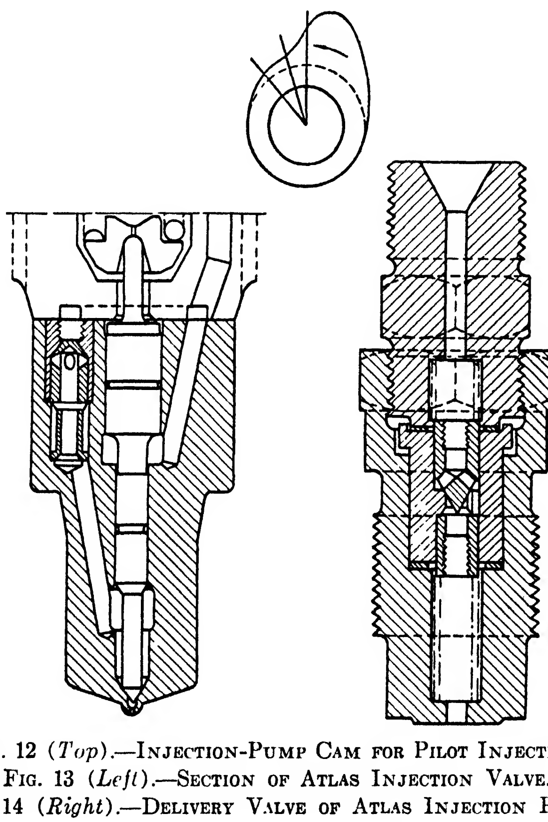


FIG. 12 (*Top*).—INJECTION-PUMP CAM FOR PILOT INJECTION.
FIG. 13 (*Left*).—SECTION OF ATLAS INJECTION VALVE.
FIG. 14 (*Right*).—DELIVERY VALVE OF ATLAS INJECTION PUMP.

throttled at the beginning, and less fuel enters the combustion chamber in a given time. Such throttling nozzles are used in some precombustion-chamber and auxiliary-chamber engines, mainly with the object of quieting the engine at idling. In the Cummins injector the space below the injector plunger is filled with a mixture of air and fuel (or a fuel fog) at the beginning of injection. This mixture is compressible, and as the plunger descends, pressure is built up gradually, hence fuel starts to enter the combustion chamber at a low rate.

Transfer Pumps—In stationàry installations, where the fuel tank can be placed a considerable distance above the injection pump, the latter can be connected to the tank directly, but this is not practical in automotive installations, as a rule. The injection pump, of course, exerts a suction on the fuel line, but if this were depended upon to fill the pump barrel with fuel during the inlet stroke, a vacuum would exist in the fuel line, which would have a tendency to draw

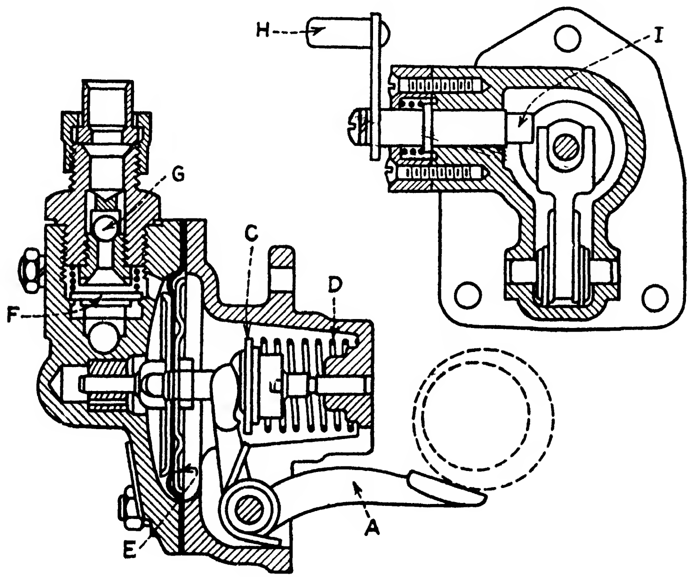


FIG. 15.—C.A.V. DIAPHRAGM-TYPE TRANSFER PUMP.

air into the line through any slight leaks at joints, etc. This is highly undesirable, and a fuel transfer pump should be used. The thought naturally occurs that the diaphragm-type fuel pumps which are used on most modern automobile engines might be suitable for the purpose, the thing to recommend them being that they are produced in very large numbers and therefore relatively cheap. The conventional automobile fuel pump is hardly sufficiently robust for the purpose, but a special diaphragm-type transfer pump (Fig. 15) is in production in England by C.A.V. Ltd. This pump is secured to the side of the injection pump and is operated by an eccentric on the camshaft of that pump. One arm *A* of the operating bellcrank extends through an opening in the side of the camshaft housing, and the forked arm *B* inside the transfer pump presses against the spring plate *C*. During half a turn of the eccentric the bellcrank compresses spring *D* and imparts a stroke of about 0.13 in. to the diaphragm *E*. This motion of the diaphragm creates a partial vacuum in the compartment to the left of it, which causes the plate valve *F* to lift against the force of its spring, and fuel to enter the diaphragm chamber through this valve. This suction stroke of the diaphragm is produced positively by the eccentric, while the return or delivery stroke is performed by spring *D*. During this stroke fuel is forced from the diaphragm chamber through the ball-type delivery valve *G*. The pressure against which the pump can deliver fuel is determined by spring *D*, and in this case is limited to between 3½ and 4½ psi. Once this pressure is reached, the stroke of the diaphragm is reduced, and less fuel is delivered. Under these conditions the operating lever *A* is held in contact with the eccentric by a spring, to prevent noise and undue wear.

Transfer pumps, as a rule, are provided with priming devices, which enable the operator to prime the system in the event the fuel tank should have been allowed to run dry and fuel should have drained out of the system. If no such priming device were provided, it would be necessary to "motor" the engine with the starter for priming, which would be hard on the battery. The C.A.V. diaphragm pump is provided with a priming lever *H*, the shaft of which at its inner end carries a cam *I* which engages the spring plate *C*. When the priming lever is depressed, it imparts a full stroke to the diaphragm, hence the system can be quickly primed. When not in use, the priming cam is held out of contact with the spring plate by a coil spring on its shaft.

American Bosch Transfer Pump—Fig. 16 shows two sectional views of the American Bosch transfer pump, which is of the plunger type. It has a single-piece cast-iron body with bores for the plunger, plunger spindle or pushrod, tappet guide, valve seats, and oil ducts. This pump also is mounted on the injection pump. It has a three-bolt flange and is located in the housing of the injection pump by a pilot which also serves as guide for the roller tappet. The roller is held in contact with an eccentric on the camshaft by a spring. From the tappet motion is transmitted to the pump plunger through a pushrod. The plunger is moved down (in the view at the left) positively by the eccentric, and is returned by its spring. During the "down" stroke fuel is drawn into the pump barrel through the inlet valve shown at the right in the right-hand view, and during the delivery stroke it is forced out of the barrel through the delivery valve, at the left in the same view. From the space above the delivery valve there is an internal duct to the delivery port directly below it. Inlet and delivery connections are made at the bottom by means of $\frac{1}{4}$-in. threaded fittings. One of these transfer pumps was shown in place on the injection pump in Fig. 3.

With this pump also the stroke decreases as the pressure in the delivery line increases, and the pressure in the sump of the injection pump is automatically maintained at about 15 psi. A priming pump is shown screwed into the transfer

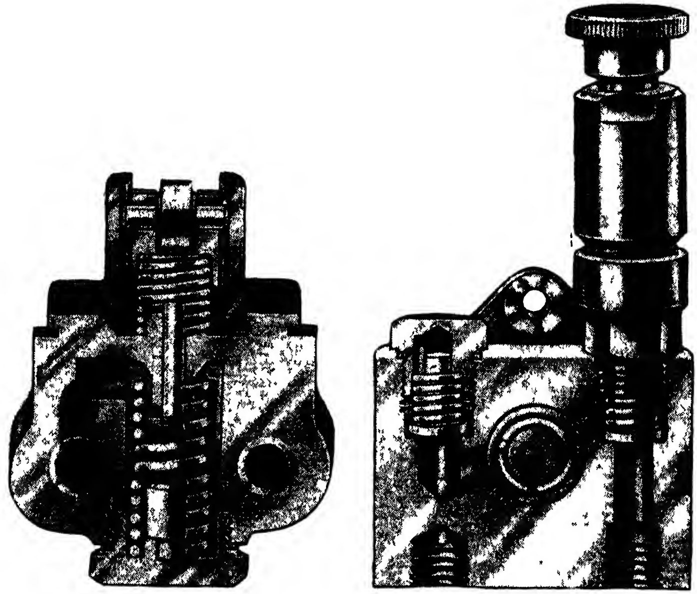


FIG. 16.—AMERICAN BOSCH TRANSFER PUMP WITH PRIMER.

pump over the suction valve. Without this priming pump the opening for the priming pump is closed by a screw plug.

Governors—In common with most other prime movers, the Diesel engine has the characteristic that it speeds up when the torque load on it is reduced, and to protect it from abuse by overspeeding or racing, it is provided with a governor. There are two principal types of governor, mechanical (centrifugal), and pneumatic. A simple centrifugal or pneumatic governor controls only the maximum speed of the engine. In automotive applications it has been found advisable to provide, in addition, a minimum-speed governor, which prevents the engine from stalling when the control device is moved to the idling position. The need for such a governor is due to a characteristic of certain fuel-injection pumps, of delivering less fuel per cycle at low than at high speed for a certain position of the control rack. The control rack must have sufficient range of movement so the pump can deliver the correct amount of fuel when the engine is running without load at high or intermediate speeds, and since with pumps of this type the delivery decreases with the speed, the tendency is for the engine to stop when the torque load increases. To prevent this is the function of the minimum-speed governor. Maximum- and minimum-speed governors can be combined in a single unit.

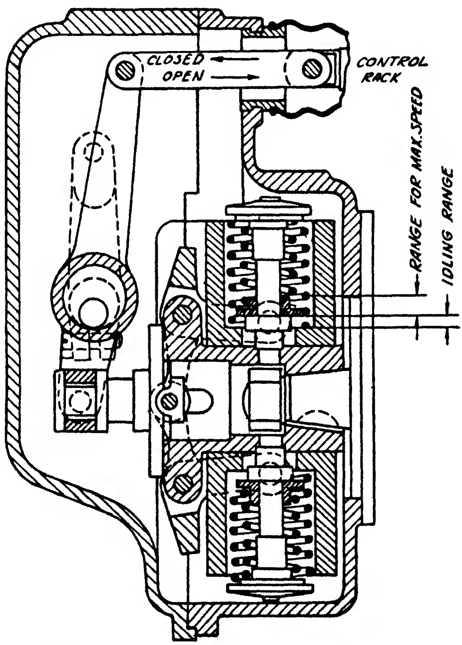
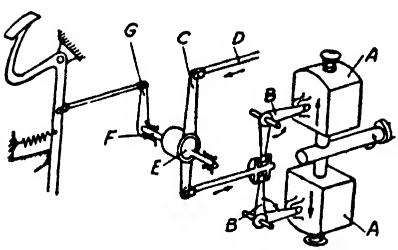


FIG. 17.—MAXIMUM- AND MINIMUM-SPEED GOVERNOR.

Full-Range Governors—Most automotive engines now are equipped with maximum- and minimum-speed or full-range governors. The basic principles of all of these governors are similar, and may be explained by reference to Figs. 17, 18, and 19, which represent an early Bosch design diagrammatically. The governor is combined with the injection pump, the two cup-shaped governor weights being mounted on pins extending radially from a tubular fitting keyed to the tapered end of the pump shaft. The governor springs are inside the cup-shaped governor weights. This arrangement has the advantage that the centrifugal force on the governor weights and the spring force of the governor springs act in direct opposition, whereby friction in the governor mechanism is reduced to a minimum, and the governor is made more sensitive and less prone to "hunt." However, the governor preferably should be operated at a higher speed than that of the pump shaft, so that the weights will produce a greater operating force. The governor weights act through bellcranks on a sliding member, which in turn acts on one arm of a double-armed floating lever whose other arm is connected to the pump control rack.

Operation of Maximum- and Minimum-Speed Governor—The operation of this governor may be readily understood from the diagrams shown in Figs. 18 and 19. The two governor weights *A*,*A* are mounted on an extension of the injection-pump camshaft, and from these weights connection is made through bellcranks *B*,*B* to floating lever *C*. When the engine is turning over at high speed, the centrifugal force



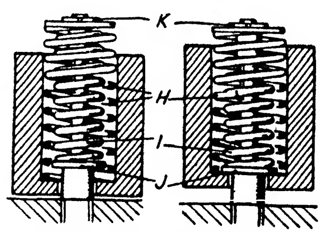


FIG. 18 (*Left*).—DIAGRAM OF MAXIMUM- AND MINIMUM-SPEED GOVERNOR.
FIG. 19 (*Right*).—SHOWING GOVERNOR WEIGHTS IN IDLING AND NORMAL OPERATING POSITIONS.

causes the weights *A*,*A* to move outward from the governor shaft, and thus to pull control rod *D* of the injection pump toward the "idling" position, thereby causing the engine to slow down. The weights are under the influence of the governor springs, and when the speed of the engine (and the centrifugal force on the governor weights) decreases, the weights are moved inward toward the governor shaft, which results in increased fuel delivery and an increase in engine speed. Floating lever *C* is connected to control rod *D* by means of adjustable links, adjustments being made by means of the central bolt in the link.

Floating lever *C* has a movable fulcrum, being mounted on the eccentric *E* on shaft *F*, which carries the lever arm *G*.

From the latter a link connects to the accelerator pedal shown at the left in the diagram. The spring on the pedal tends to draw the control rod in the direction toward the idling position. When the accelerator is depressed by the driver, the control rod is moved away from "idling" and the engine is speeded up regardless of the position of the governor weights. In this way automatic speed control by the governor is combined with independent control by pedal.

The governor is of such design that it governs both the maximum speed which the engine can attain when the driver presses down on the accelerator pedal, and the minimum speed of idling when the foot is removed from the pedal. How this is accomplished is made clear by the diagram Fig. 19. Each governor weight is provided with two springs, a lighter outer spring *H* and a stiffer inner spring *I*. When the engine is idling, only the outer spring *H* presses against the governor weight, the washer *J* of the inner spring resting against a shoulder on the transverse pin on which the governor weights are mounted. Spring *H* forces the governor weight inward, toward the shaft, and such motion of the weight results in an increase in the quantity of fuel injected. This increase in fuel as the idling speed is approached prevents stalling of the engine. A relatively small increase in speed above idling will move the governor weight into contact with washer *J*, because spring H has a low rate. With further increase in engine speed the governor weight remains in this position until the "governed speed" is reached, because the inner springs are set to exert a high initial pressure. However, if the pedal is pressed down so far that more fuel is delivered than is required to keep the engine running at the maximum desired speed under the load upon it, then springs *I* are compressed, the governor weights move farther out from the shaft, and the control rod of the pump is moved toward the "idling" position.

When the engine is being cranked over, the centrifugal force on the governor weights is very light, and springs *H* force them inward, thereby moving the control rod away from the "idling" position. As this would result in more fuel being delivered to the engine than can be burned properly, a stop is provided for the control rod which limits its motion away from the "idling" position. This stop is adjustable. In addition to this stop at the far end of the control rod, there is provided a stop on the outside of the governor housing which limits the angular motion of shaft *F*. The speed range of the governor can be adjusted by means of adjusting nuts *K*, which support the outer spring washers.

American Bosch Type GV Governor—Fig. 20 shows two views of a more recent type of governor produced by American Bosch Corporation. The shaft on which the centrifugal masses are carried is driven from the end of the pump shaft through spur gears giving an increase in speed, thereby making it possible to obtain the needed governing force with smaller centrifugal masses. The lever arms of the pivoted centrifugal masses act on a sliding sleeve *A*, and the axial force exerted on this sleeve as a result of centrifugal force on the rotating masses is opposed by a pair of coil springs *B*. A collar *C* is placed in a groove cut in a flange on the sliding sleeve, and a yoke *D* surrounds the collar and is connected to it by a pair of trunnion screws *E*. The upper end of yoke *D* connects to the pump control rack *F* by means of a link *G*, while its lower end is hinged to a lever *H* mounted on control shaft *I*. Lever *H* is free on the control shaft, but there is a fixed hub *J* on the shaft, and lever *H* and hub *J* are provided with lateral prongs to which are attached the ends of a double-coil torsion spring *K* which surrounds hub *J* and tends to turn lever *H* around shaft *I* in such a direction as to cause the two points of spring anchorage to approach each other. To control shaft *I* outside the governor housing is secured a control lever *L*, from which connection is made to the accelerator pedal.

Adjustably secured to the upper end of yoke *D* is a torque cam *M* which, by abutting against an adjustable stop plate *N* limits the maximum fuel quantity that can be injected. It follows that motion of quantity-control rod *F* to the left increases the injection quantity, and vice versa. Carried in a boss on the governor housing back of the upper end of yoke *D* is a bumper spring *O*. The yoke contacts this spring only when the engine is running at high speeds without load, and the object of the spring is to prevent surging or hunting of the engine when the load is suddenly removed and the yoke comes up against the stop.

How the governor increases the full-load injection quantity when the speed is reduced may be explained as follows: Since the engine is under full load, torque cam *M* is in contact with stop plate *N*, and if the speed is reduced, the centrifugal masses approach their axis of rotation, and sliding sleeve *A* is forced toward the left by governor springs *B*. The yoke, pivoting around the line of contact of the torque cam with the stop plate, turns in a clockwise direction around this pivot axis, such motion being permitted by the torsion-spring connection between lever *H* and fixed hub *J* on control shaft *I*. But since the connection between link *G* and yoke D is below the axis around which the yoke turns, the link—and the control rod with it—will move toward the left and cause a larger quantity of fuel to be injected. This effect is obtained entirely automatically, without change in the position of the accelerator pedal and of the control shaft I to which it is connected.

In a separate compartment at the side of the governor housing—and therefore quite accessible—are located the adjusting means for idling speed and maximum speed. These consist of set screws with the points of which a stop plate on the control shaft contacts when the latter is in its extreme positions.

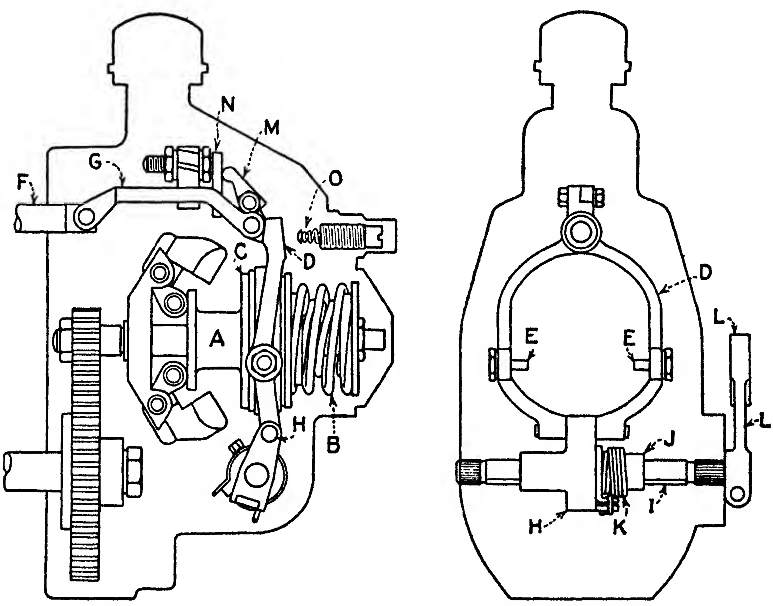


FIG. 20.—AMERICAN BOSCH TYPE GV CENTRIFUGAL GOVERNOR.

A still later model, the GVA, shows a number of detail improvements: The rollers on the contact fingers of the centrifugal masses are omitted, the fingers now pressing against an antifriction thrust bearing on the sleeve. The coil-type bumper spring O has been replaced by a flat spring.

I.H.C. Governor—A governor used on International Harvester Company's four-cylinder, single-plunger injection pump, which was illustrated in Fig. 5, controls the maximum speed, the idling speed, and the torque characteristics under full load at less than governed speed. The governor proper—the governor weights with the governor sleeve—was shown in the drawing of the pump. In Figs. 21 and 22 are shown three views of the mechanism by means of which the governor acts

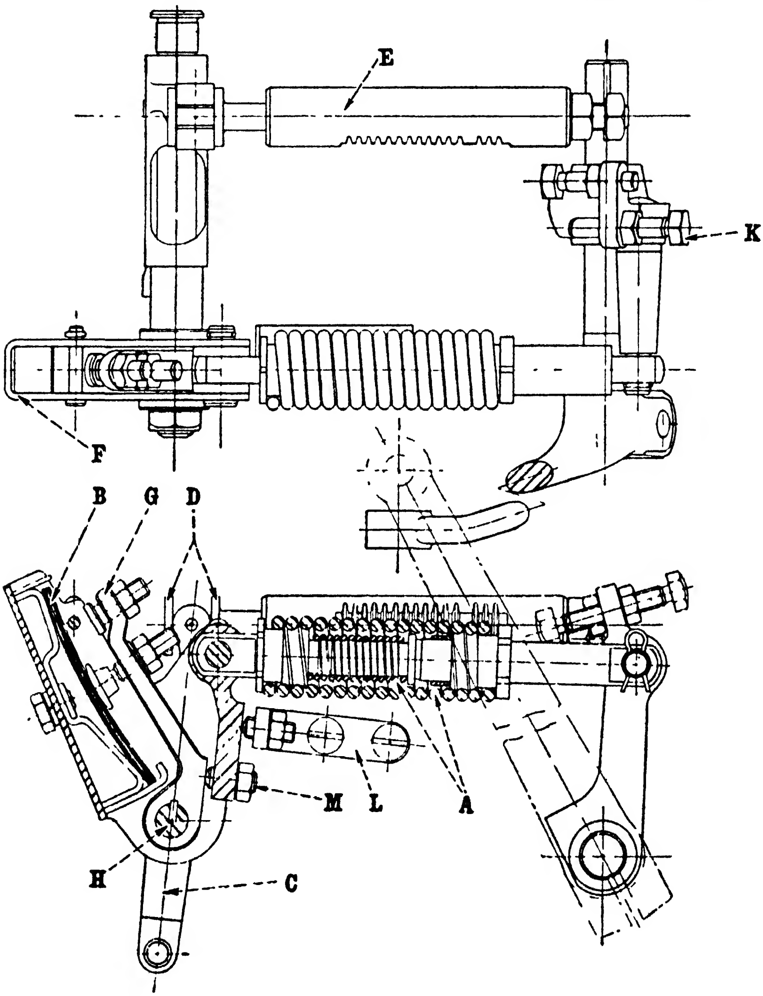


FIG. 21.—TWO VIEWS OF I.H.C. CENTRIFUGAL GOVERNOR.

on the control rack of the pump and is acted upon by the operator.

Referring to the drawings, *C* is the governor fork, the prongs of which at their ends carry horizontal pins against which the governor sleeve acts through a thrust bearing. An increase in engine speed causes the governor weights to move out from their axis of rotation and turns fork *C* and its shaft *H* in the counterclockwise direction. An upward extension of governor fork *C* engages a spool *D* on pump control rack *E*. Any increase in engine speed tends to move rack *E* toward the left, decreasing the injection quantity, and vice versa.

Centrifugal force on the governor weights is counteracted, under different conditions, by three different springs, the heavy and light coil tension springs *A* (referred to as the governor springs), and the leaf spring *B*, called the torque spring. The latter forms part of a torque-control unit which also includes the torque arm *F* and the torque-control lever *G*. Lever *G* is keyed to shaft *H*, while torque arm *F* is free to rotate on that shaft within limits.

When the governor weights move outward from their axis of rotation under the influence of centrifugal force, lever *G*

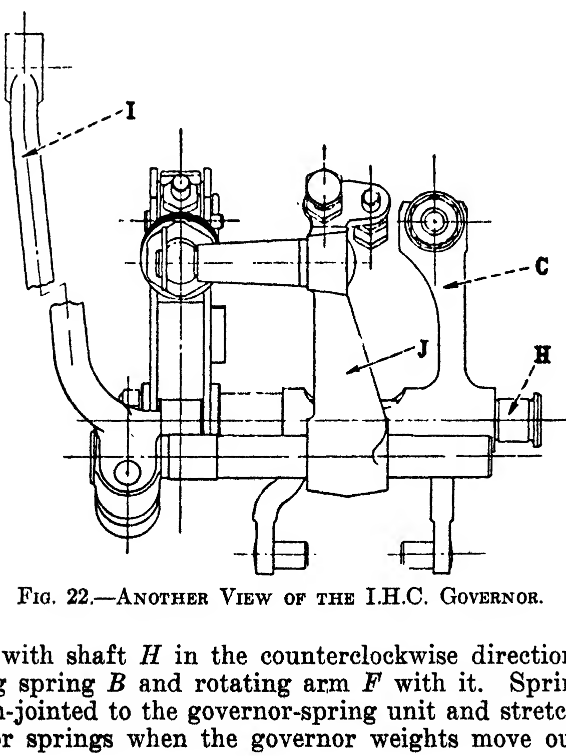


FIG. 22.—ANOTHER VIEW OF THE I.H.C. GOVERNOR.

rotates with shaft *H* in the counterclockwise direction, compressing spring *B* and rotating arm *F* with it. Spring arm *F* is pin-jointed to the governor-spring unit and stretches the governor springs when the governor weights move outward.

Control lever I on the outside of the pump is rigid with governor spring lever J inside the housing, both being secured to the same shaft. Lever J connects to the rear (right-hand) end of the spring unit. The position to which control lever I is moved by the operator determines the position of the anchorage point of the governor spring, and also the position of the torque-control unit, which in turn controls the engine performance. Movement of hand-control lever I is limited by stop screws carried by lever J inside the pump housing.

When control lever I is set in its extreme rear (right-hand) position, which is determined by stop screw K, governor spring A will be under light tension and will pull the torque-control unit against stop L on the outside of the pump housing. This is the maximum-power setting of the control lever. At governed or maximum speed, the tension in spring A will hold torque-control arm F against stop L, and torque spring B will be fully compressed by torque-control lever G keyed to shaft H.

If now the engine load decreases, the speed will increase, the governor weights will move outward, and fork C will turn in the counterclockwise direction. This puts additional tension on governor spring A and moves control rack E forward (to the left), decreasing the injection quantity and reducing the speed to substantially its original value. If all of the load is removed from the engine, the control rack is moved to a position corresponding to the "high idling speed," in which the increased centrifugal force on the governor weights is balanced by additional tension in the governor springs. Any increase in engine load, resulting in a decrease in speed, moves control rack E in the opposite direction (to the right), until at full load torque arm F ends up against stop L on the outside of the housing. At this point the governor springs A become ineffective, and if the load is increased still more, the speed of the engine will drop below "governed" speed, and the centrifugal force on the governor weights will be balanced solely by the force of torque spring B. The drop in speed resulting from the increase in load will reduce the centrifugal force on the governor weights, and torque spring B, acting on torque-control lever G, will move control rack E to the rear (right), causing the pump to deliver more fuel per stroke. Thus the effect of the torque-control unit is to increase the injection quantity as the load increases and the speed decreases (below governed speed).

If it is desired to idle at low speed, the hand-control lever is moved forward until a poppet in the governor friction control on the engine drops into its socket. With the lever in this position, the anchorage point of governor spring A is far enough forward (to the left) to remove all tension from the outer governor spring. Torque spring B has moved torque-control lever G against stop screw M in torque spring arm F, and control rack E has been moved forward (to the left) to a position where only enough fuel for slow idling is injected. The centrifugal force on the governor weights now is balanced solely by the tension of the light inner governor spring, which latter is so arranged that it can stretch a small distance before the outer spring is picked up.

Cummins Governor—Cummins Engine Company fits its engines with what is known as a two-stage governor. As shown in the upper left-hand corner of Fig. 11, the device comprises two pairs of centrifugal weights. The light spring shown at the top tends to increase the fuel quantity injected into the cylinders, preventing stalling of the engine at idling speeds. It is counteracted by the idling weights, which prevent the injection of an excess of fuel under these conditions. For normal operation the strut of the metering pump is moved away from the idling position by hand control. That compresses the light idling spring, and the idling weights are then held against a stop. When the engine reaches the "governed" speed, the maximum-speed weights compress the heavy spring, and move the strut of the metering pump toward the shutoff position, thereby preventing a further increase in speed.

Pneumatic Governor—A pneumatic-type of governor was developed by the Robert Bosch Company and in slightly modified form is being manufactured by the various Bosch successor firms. Fig. 23 shows a design due to C.A.V. Ltd. of Great Britain. This governor, which seems to be used mainly on tractor engines, is actuated by inlet-manifold vacuum. Ordinarily there is no throttle in the inlet pipe of a Diesel engine, but when this pneumatic governor is used, a venturi-shaped throat is provided, and a butterfly valve is placed in this throat. A small section is scalloped out of the butterfly disc near its edge, and a by-pass or tunnel is formed in the throat in such a position that it will occupy the recess in the throttle disc when the latter is closed.

The pneumatic governor consists essentially of two parts, the throttle valve C already mentioned, which is located in a flanged casting inserted between the air cleaner and the inlet pipe, and a diaphragm chamber mounted on the injection pump. At the side of the throttle housing (or venturi) there is a small fitting D which opens into the by-pass in the venturi and connects by a tube E to the vacuum chamber. The vacuum in the venturi is governed by the position of throttle C, which latter, by means of a linkage to the accelerator pedal, can be moved around its axis, movement in both directions being limited by adjustable stops. The diaphragm chamber

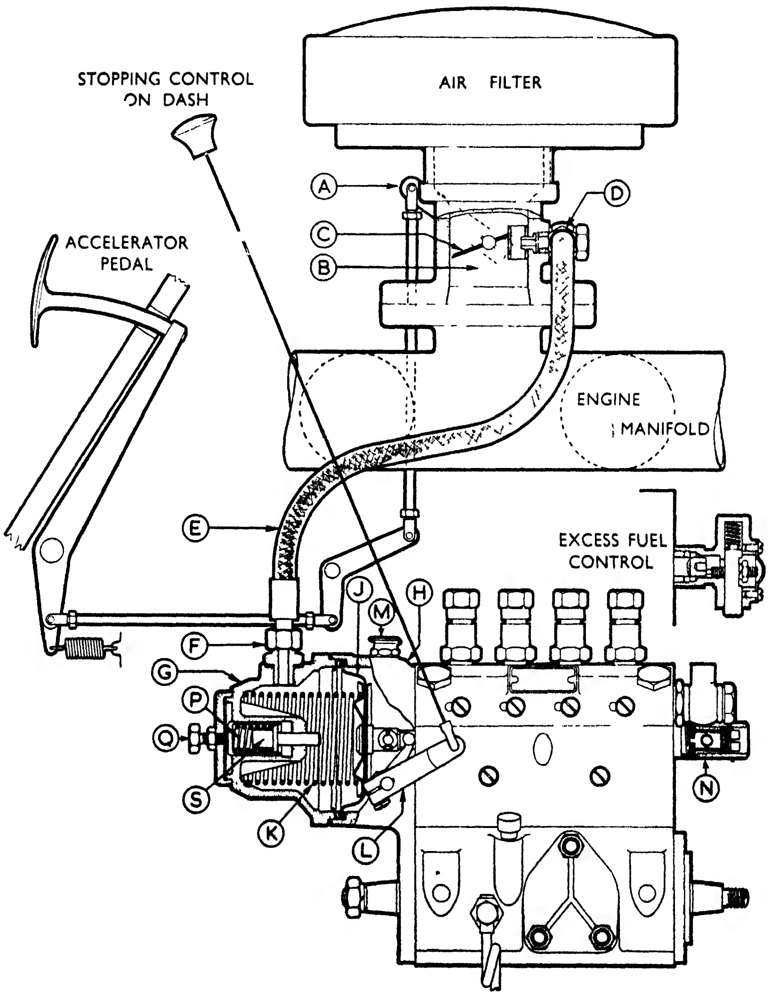


FIG. 23.—C.A.V. PNEUMATIC GOVERNOR.

consists of two castings, with a leather diaphragm clamped between them. A light spring *K*, pressing against the diaphragm, tends to keep the latter in the "full open" position and to damp out pressure fluctuations. A stop lever *L*, mounted on the diaphragm chamber, connects to a stopping control on the dash by a wire and acts directly on the pump control rack. To prevent difficulty in controlling idling speeds under conditions where the intake pressure may vary widely, an auxiliary spring *P* is provided which can be adjusted by set screw *Q* to suit the conditions of the particular installation.

To start the engine, control lever *L* is released and spring *K* then forces the fuel-control rack all the way to the right, which is the starting position. After the engine has been started, the accelerator pedal is partly released, which partly closes throttle valve *C* and increases the vacuum in compartment *G*. The overpressure in compartment *H*, which is open to the atmosphere, then forces the diaphragm and the fuel-control rack toward the left, thereby cutting down the injection quantity and reducing the engine speed in accordance with the position of the pedal. To increase engine speed, the accelerator pedal is depressed, which opens throttle *C*, decreases the vacuum, and allows spring *K* to move the fuel-control rack so as to increase the injection quantity.

General Motors Fuel Modulator—G.M. Series 71 engines installed in highway trucks are equipped with a special governor which, in addition to governing the maximum and minimum (idling) speeds, reduces the fuel charge when the engine is slowed down by a heavy load, regardless of the throttle position. The object of this additional governor element—known as the Fuel Modulator—is to improve the fuel economy and to prevent a smoky exhaust at high load factors. Two sectional views of the governor are shown in Fig. 24.

The governor on the Series 71 engine is driven through the Roots blower, the splined end of its shaft fitting a splined hole in the end of the blower upper rotor shaft. As in other maximum- and minimum-speed governors, there are two pairs of centrifugal weights, high-speed weights *A* and low-speed weights *B*, which act through a sleeve or riser and a ball thrust bearing on an operating fork at the lower end of an operating shaft extending up through a tube to the control housing. The latter is located at the level of the unit injectors, which are on top of the engine. In the lower view of Fig. 24 a part of the vertical tube is cut away, for reasons of space.

Within the control housing there are three coil springs, a high-speed spring *C*, a low-speed spring *D*, and a modulating spring *E*. The governor weights act on these springs through the vertical shaft, which at its upper end carries an operating lever (or bellcrank) *F*, one arm of which bears against low-

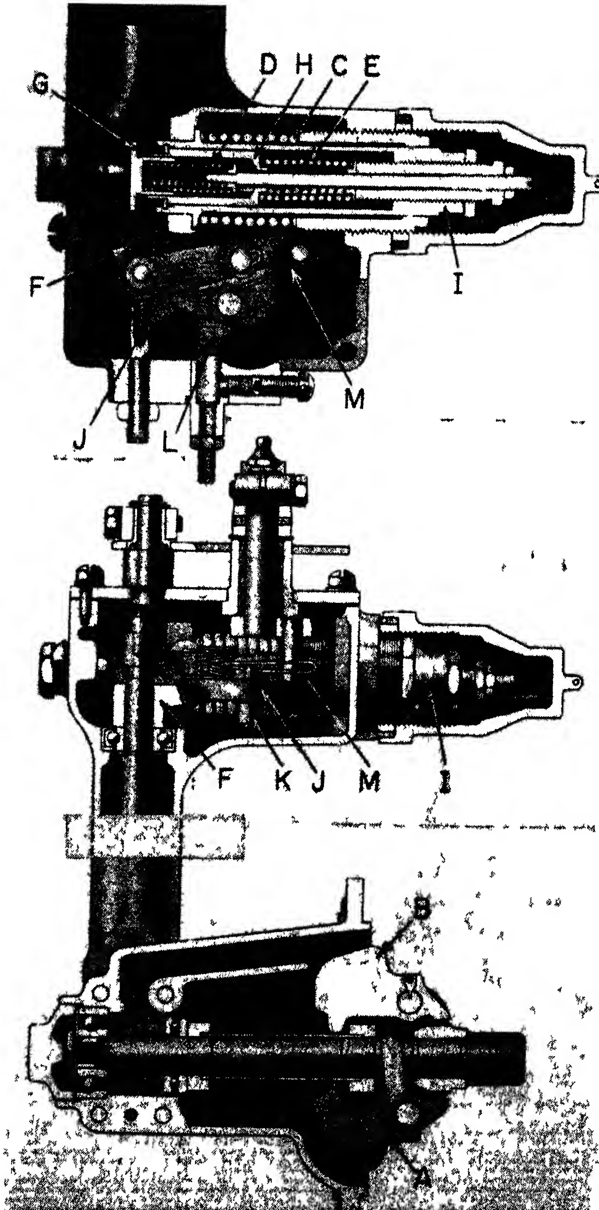


FIG. 24—GENERAL MOTORS FUEL MODULATOR.

speed spring cap G. When low-speed spring D is compressed, its cap first comes in contact with modulating-spring plunger H, and further movement of the cap brings it into contact with high-speed spring plunger I. It is not necessary to describe in detail the action of the so-called limiting-speed governor, as it is similar to that of other governors limiting both the maximum and the idling speed, and in the following only the fuel-modulating feature of the device will be explained.

The centrifugal force on the governor weights naturally decreases with engine speed, and as the speed drops below 1800 rpm, low-speed spring D and modulating spring E, acting together, overcome the centrifugal force and move low-spring cap G toward the right. This motion of the cap is transmitted to operating lever F, on which is pivoted the differential lever J. The latter is caused to rotate around its pivot, but its rotation is restricted by engagement of its roller K with modulator cam L. The cam causes the differential lever to rotate in such a direction that the governor link is pulled into the governor and the fuel-control racks of the unit injectors are withdrawn to reduce the injection quantity.

The modulating motion of lever J is opposed by torsion spring M, which holds roller K lightly in contact with cam L. It is due to the provision of this spring that the fuel modulator is able to operate regardless of the position to which the operator may move the "throttle."

The fuel modulator is used together with two different injectors, of 70 and 80 cu mm capacity, respectively. With the 80-cu mm injector the fuel quantity is decreased immediately as the speed drops below 1800 rpm, because the helix of its plunger starts at the "full-load" end. With the 70-cu mm pump, on the other hand, the first 0.075 in. movement of the fuel rack, corresponding to a decrease in engine speed from 1800 to 1600 rpm, does not reduce the injection quantity, because the helix of the injector plunger starts only at a certain distance from the "full-load" end.

Governor with Hydraulic Relay—On certain locomotive, rail car and other large engines, use is made of the Woodward governor with hydraulic relay. With this system of control, the injection-pump control rod is directly connected to a "power piston" in a cylinder. A spring pressing against one side of the piston always tends to force the control rod of the injection pump toward the idling position. The other side of the power piston is acted on by oil under pressure. The oil is maintained under constant pressure by a gear pump and a spring-controlled by-pass valve; this corresponds to the usual pressure-feed lubrication system, and the engine oiling system can be tapped for the purpose, in which case no special pump is required. Admission of oil under pressure to one end of the power-piston cylinder and evacuation of oil therefrom are controlled by a piston valve, which in turn is under the control of a centrifugal governor.

A centrifugal governor with hydraulic relay is used on some Cummins engines, and a section through the governor

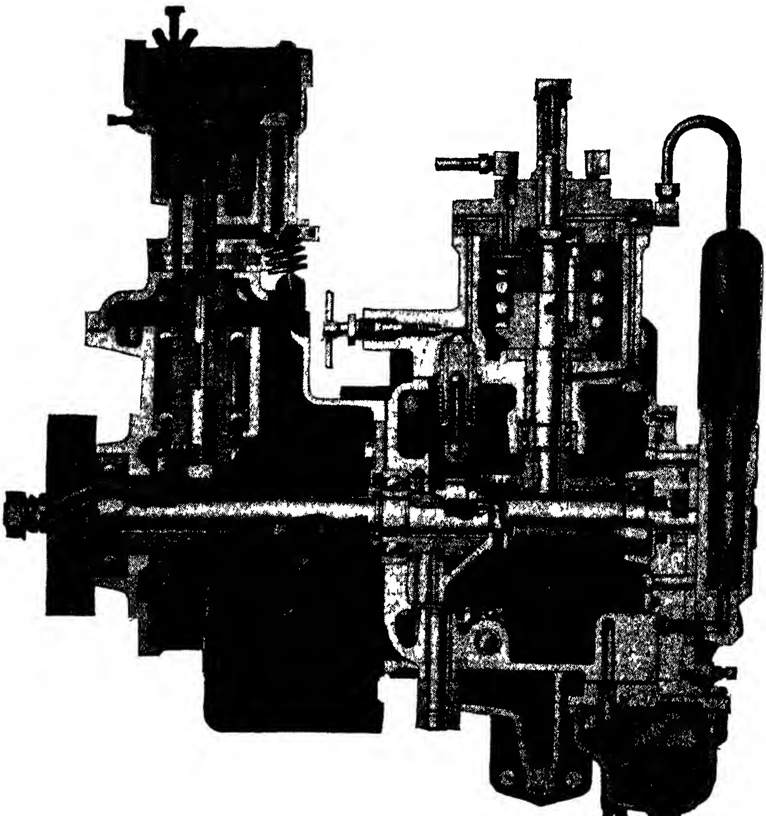


FIG. 25.—SECTION OF CUMMINS FUEL PUMP AND GOVERNOR WITH HYDRAULIC RELAY.

and fuel-injection unit of this engine is shown in Fig. 25. A horizontal shaft driven from the engine crankshaft through helical gears drives the governor (on the left) and the injection unit (on the right) through bevel gears. The L-shaped centrifugal masses are clearly shown at the top of the governor shaft; they act through a ball thrust bearing on a piston valve inside the hollow, ported governor shaft, which controls the admission of oil under pressure to the space below the plunger shown to the right of the governor shaft. This plunger acts on a lever which connects to the mechanism that varies the stroke of the single injection pump, as explained earlier in this chapter. In the illustration the piston valve is shown to close the port to the space below the hydraulic plunger. If the engine speed increases, the valve is raised, and oil is allowed to escape from below the plunger to the bottom of the governor housing. The coil tension spring, of which a part is shown below the plunger, then pulls the plunger down and cuts down the injection quantity. If, on the contrary, the engine speed should decrease, the piston valve will be forced down in the hollow governor shaft by the governor spring, and place the space below the plunger in

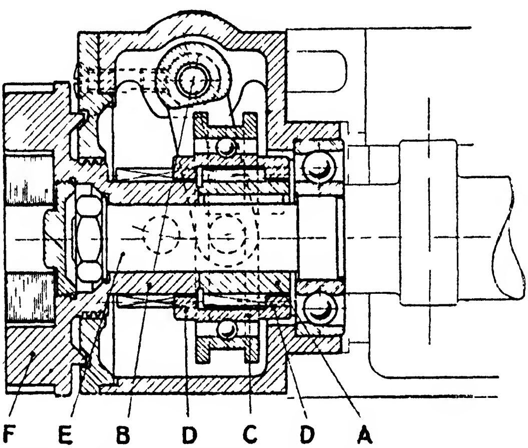


FIG 26—MECHANISM FOR MANUAL CONTROL OF INJECTION TIMING.

communication with the source of oil under pressure, the oil entering the hollow governor shaft through ports located above that controlled by the valve. The oil pressure then forces the plunger up and increases the injection quantity, which in turn restores the engine speed.

The right-hand part of the illustration shows many details of the injection unit, which will be recognized if reference is made to the diagrammatic drawing and description of the Cummins injection unit given earlier in this chapter.

Injection Timing—For best results with respect to both power and economy, the injection must be advanced as the speed increases. Fig. 26 is a sectional view of a manual timing device. At its driving end the camshaft of the injection pump is provided with an extension *E* which carries a splined cylinder *A* keyed to it, and another, *B*, which is free to turn on it. Both cylinders are cut with helical splines, but the two sets are of opposite hand, that is to say, they are inclined in opposite directions. The two splined members are surrounded by a sliding sleeve *C* which is chambered out at the middle and has internal splines cut in its end portions *D,D*. Sleeve *C* forms the inner race of a ball bearing, whose outer race is cut with a groove for the ends of the shifter fork. The splined cylinder *B* is formed integral with clutch member *F*. Since the splines on *A* and *B* are of opposite hand, when sliding sleeve *E* is moved axially, the angular relation between the driving and driven shafts is changed, and the injection either advanced or retarded.

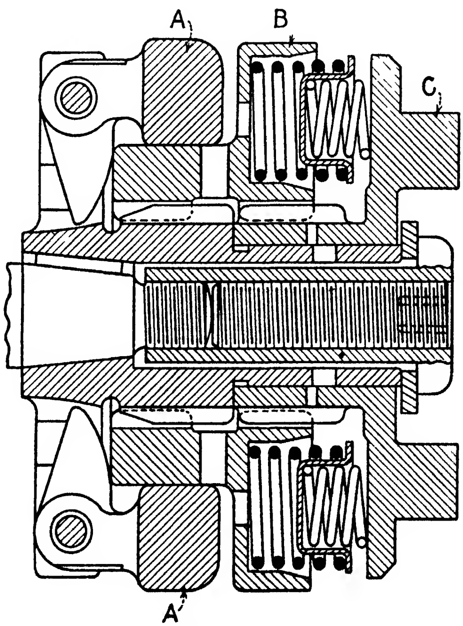
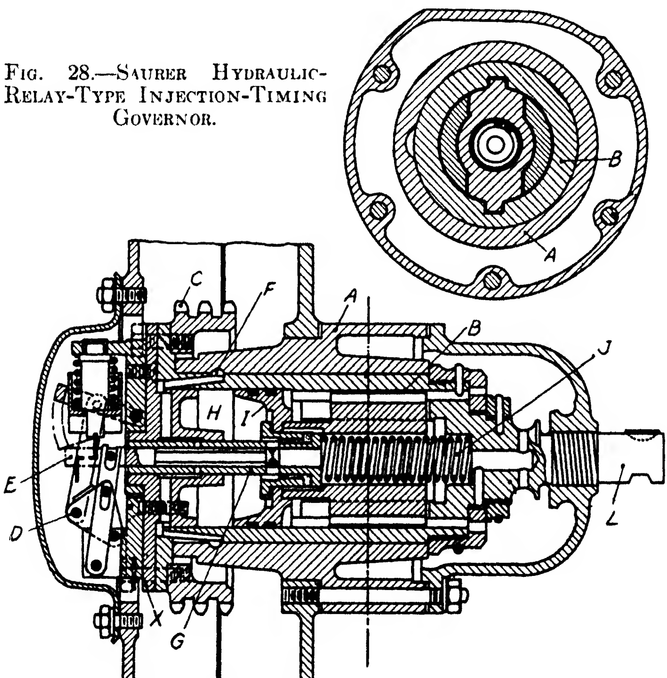


FIG. 27.—MACK INJECTION-TIMING GOVERNOR.

Timing Governor—Timing governors sensitive to speed changes are fitted by a number of manufacturers, and Fig. 27 is a sectional view of that used on Mack Lanova engines. It consists essentially of three elements, viz., a pair of governor weights *A,A*, mounted on the pump shaft; a sliding member *B* with two sets of internal helical splines cut in it, and a clutch member adapted to engage with a similar member on the driving shaft. The two pairs of internal splines engage similar external splines on a short tubular shaft keyed to the clutch member. Coil springs located in sockets formed on the sliding member force the latter away from the clutch member and change the angular relation between the driving and pump shaft in such a way as to cause injection to occur late. As the speed of the engine increases, centrifugal force on the governor weights causes them to move out from the axis of rotation. The sliding member is thereby moved axially against the spring pressure, and injection is advanced.

Hydraulic-Relay Type Timing Governor—An injection-timing governor incorporating a hydraulic relay is being used

FIG. 28.—SAURER HYDRAULIC-RELAY-TYPE INJECTION-TIMING GOVERNOR.



by Adolphe Saurer of Arbon, Switzerland, and sectional views of it are shown in Fig. 28. Mounted in bearing *A* in the housing of the "front-end drive" is a hollow shaft *B* which is driven through a triple roller chain running over the sprocket *C* secured to a flange on it. Fastened to this flange is a header *X* on which are carried the centrifugal masses *D* of the governor, which act through levers on the central slide valve *E*.

A lead from the pressure lubricating system of the engine connects to bearing *A*, and oil from this lead passes through groove *F* and radial passages in the header into tube *G*, in which valve *E* is located. There are ports in tube *G* which are covered or uncovered by valve *E* when the latter moves relative to the tube. When these ports are uncovered, oil from the engine lubricating system will enter cylinder *H* and exert a pressure against piston *I*, which pressure is resisted by spring *J*. Connected to piston *I* is a member *K* which has on it two helical splines that engage into helical grooves on the inside of shaft *B*. Member *K* does not completely fill the bore of shaft *B*, but leaves spaces on opposite sides into which the yoke-shaped extension of shaft *L* enters. Tube *G* also is connected to piston *I*, and the ports in the tube therefore are displaced axially whenever the piston moves.

Assume that the ports in tube *G* are closed by valve *E* and that the whole system is in equilibrium, as it will be whenever the engine is running at constant speed. If now the speed increases, the governor masses *D* will move out from the axis of rotation, with the result that valve *E* will uncover the ports in tube *G*, and oil will enter cylinder *H*, forcing piston *I* and the attached member *K* toward the right, against the pressure of spring *J*. Because of its helical splines, any axial motion of member *K* is accompanied by an angular motion. This angular motion (but not the axial motion) of member *K* is communicated to shaft *L*, from which the injection pump is driven, and thus the time of injection is advanced. Motion of tube *G* with piston *I* closes the ports in the tube, and thereby shuts off the flow of oil into the cylinder, so that equilibrium is restored. If thereafter the speed of the engine should decrease, valve *E* would be moved by the governor spring to open the ports in tube *G* again, and as at the lower engine speed the oil pressure in the lubricating system will be lower, spring *J* will move piston *I* toward the left, forcing oil from cylinder *H* and drawing member *K* toward the left, thereby retarding injection.

CHAPTER VIII

Precombustion-Chamber Engines

The type of high-speed engine first to be produced commercially was that having a precombustion chamber (also known as an ignition chamber, ante-chamber, or pre-chamber). An engine of this type for use on trucks and tractors was placed on the market in Germany in 1923 by the Mannheim Motor Works, formerly Benz & Co. Quite a number of high-speed Diesel engines had been built previous to that year, but none, apparently, had advanced beyond the experimental stage.

Definition—A precombustion-chamber engine is one in which a small, separated part of the combustion chamber communicates with the main part, in the cylinder, through one or more small passages or orifices, and the fuel is injected into the small (precombustion) chamber opposite its point of communication with the cylinder. In an engine of this type, all of the fuel injected must necessarily pass through the precombustion chamber.

In general arrangement the precombustion-chamber engine is similar to another type, known as the turbulence-chamber engine. In both a portion of the combustion chamber is divided off, and in both the fuel is injected into the separated portion of the combustion chamber. The main differences between the two types are that, whereas in the turbulence type the volume of the separated chamber comprises nearly the whole of the compression volume, the volume of the precombustion chamber is only one-third or less of the total compression volume; that the passage through which the turbulence chamber communicates with the cylinder is much larger in cross section than the orifice (or orifices) of the precombustion chamber, and, finally, that the orifices of the precombustion chamber are arranged with a view to effecting uniform dispersion of the mixture of gases and atomized liquid ejected from the precombustion chamber throughout the combustion chamber in the cylinder, whereas the passage through which the turbulence chamber communicates with the cylinder is so arranged that the air entering the chamber from the cylinder during the compression stroke is caused to swirl therein. The principal objects of the two designs are quite different. What is aimed at in the precombustion-chamber type is to generate pressure for the injection of the fuel into the cylinder, by the combustion of a small part of the fuel injected into the chamber, while in the case of the turbulence chamber the object is to produce a vigorous swirl of air in the chamber, so that when fuel is injected the combined motions of fuel and air may result in a better mingling of the two.

In addition to the two types just described, there is a third type of Diesel engine with divided combustion chamber, referred to as the auxiliary-chamber type. In this type, however, the fuel is not sprayed into the separate chamber from the side opposite the outlet to the cylinder; it is sprayed into the portion of the compression space outside the separate chamber, though generally into or toward the outlet from that chamber.

The precombustion-chamber principle has been widely used in Germany, and there was a great deal of patent litigation in that country in connection therewith. The courts recognized Prosper L'Orange, designer of the Benz engine,[1] as the real inventor of this type, but the patent expired in 1932, and its use therefore is now free to the public. It is characteristic of the way in which ideas regarding new inventions often change in the course of their development, that L'Orange in his patent specification referred to what we now call the precombustion chamber or ignition chamber, as an enlargement of the spray nozzle. As he considered the inlet to the precombustion chamber, the chamber itself, and the outlet therefrom all parts of the injection nozzle, he naturally expected to spray the fuel right through the precombustion chamber. He claimed, however, that the fuel would be ignited in the nozzle and that the fuel spray throughout the period of injection would be accompanied by gases and vapors formed in the enlargement, which would help to atomize the fuel.

Advantages of the Type—The precombustion-chamber type of engine has the advantage over the direct-injection type that a much lower injection pressure suffices for it. With direct injection the injection pressure is generally of the order of 3000-4000 psi and sometimes much higher, while in a precombustion-chamber engine the injection pressure ranges between 900 and 1500 psi. With these lower injection pressures the elasticity of the fuel and fuel lines is a factor of much less importance, and trouble from dripping is practically eliminated. To get the same amount of fuel into the engine in the same time under lower fuel pressure, the orifices must have a considerably larger diameter, and these are less likely to become clogged. Self-cleaning pintle nozzles are generally used in this type of engine.

A very important advantage of the precombustion-chamber principle is that it materially reduces peak pressures. With direct injection, peak pressures of 1200 psi and over are quite possible, while in precombustion-chamber engines they seldom exceed 750 psi. Consequently, the latter type of engine is considerably smoother in operation. The cause of the high peak pressures in direct-injection engines was fully explained in the chapter on Combustion Phenomena. That the peak pressures are much lower in precombustion-chamber engines would indicate that in these there is practically no ignition lag in the main combustion chamber, the fuel burning therein at the rate at which it enters. The relatively small rise in pressure above the compression pressure at the

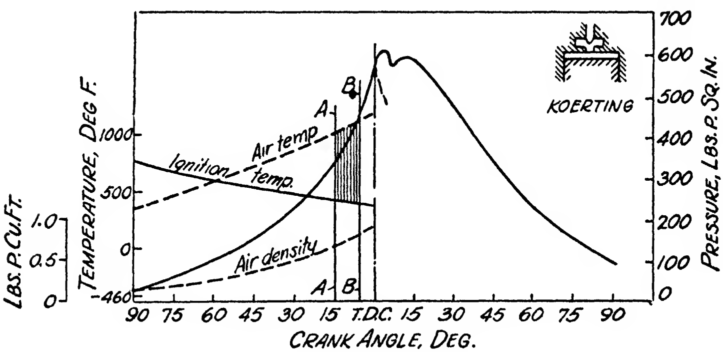


FIG. 1.—PRESSURE-TIME DIAGRAM OF PRECOMBUSTION-CHAMBER ENGINE (KOERTING).

beginning of combustion in a precombustion-chamber engine is well illustrated by Fig. 1, which shows a pressure-time diagram taken from the cylinder of a Koerting precombustion-chamber engine by Professor Neumann. The compression pressure shown is 580 psi, and the maximum combustion pressure, 625 psi, hence the pressure rise on combustion amounts to only 8 per cent. A direct-injection engine tested at the same time had a compression pressure of 565 and a maximum combustion pressure of 930 psi, and in its case the pressure rise due to combustion was practically 65 per cent.

Better Idling Qualities—It is claimed for the precombustion-chamber engine that it idles better and also pulls better at low speeds. In a direct-injection engine, if the accelerator pedal is pressed down smartly while the engine is still running slowly, there is likely to be insufficient pressure behind the fuel spray to carry it to the more remote parts of the combustion chamber, which results in incomplete combustion and a smoky exhaust, particularly in the case of a quiescent combustion chamber. In a precombustion-chamber engine, the quantity of fuel burned in the precombustion chamber and the pressure generated thereby are independent of both speed and load, hence the dispersion of the fuel charge in the main chamber is equally good at all engine speeds. Early engines of the direct-injection type did not idle well, and one maker of such engines solved the problem by injecting fuel into three cylinders only, during idling periods. The difficulty has since

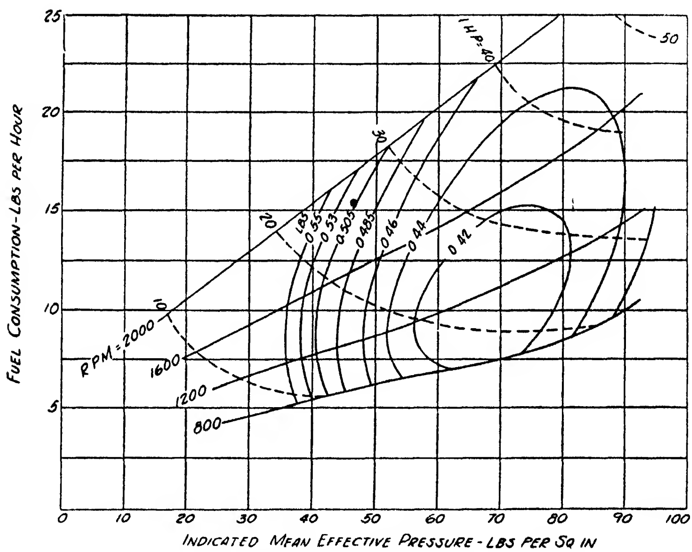


Fig. 2.—Consumption Characteristics of Precombustion-Chamber Type 50-HP Engine.

been overcome by subjecting fuel injection to governor control throughout the speed range. From the very fact that in the precombustion-chamber engine the combustion process is so modified that the peak pressures are materially reduced, it follows that this engine is much less sensitive to fuel characteristics. Moreover, since larger orifices (and lower injection pressures) are used, the requirements with respect to maximum viscosity of the fuel are less severe.

Higher Fuel Consumption—The chief disadvantage of the precombustion-chamber engine is that it consumes from 10 to 12 per cent more fuel than an equivalent direct-injection engine under similar conditions. There are several reasons for this. The principal one probably is that transfer of the fuel from the precombustion chamber to the main chamber takes place gradually, and generally in a number of successive discharges, so that some of the fuel burns rather late in the cycle, with consequent loss in thermal efficiency. Besides, the breaking up of the combustion chamber into two parts increases the wall area and therefore the heat loss through the walls.

Fuel-consumption characteristics of a small precombustion-chamber engine are plotted in Fig. 2. They apply to a four-cylinder light-truck engine of 230 cu in. displacement, rated 50 hp at 2000 rpm. The chart shows that the *indicated* thermal efficiency is at its maximum when about 25 hp is being generated at 1200 rpm. The specific fuel consumption is then about 0.41 lb per *indicated* hp-hr, and if the mechanical efficiency under these conditions is assumed to be 75 per cent, the consumption per brake hp-hr would be 0.55 lb. That seems rather high for a Diesel engine, but it must be considered that the figures apply to a small engine of an early design.

It is generally conceded that precombustion-chamber engines are more difficult to start than the direct-injection type, and glow plugs are frequently used as starting aids, particularly abroad. In these engines ignition takes place at a point close to metallic walls. When starting a cold engine these walls are at a low temperature and readily absorb heat generated by compression, hence it is more difficult to reach a temperature sufficiently high for ignition.

Arrangement of Precombustion Chamber—A sectional view of the upper end of the cylinder of the original Benz precombustion-chamber engine is shown in Fig. 3. It will be seen that the chamber is located in the cylinder head between the inlet and exhaust valves. The injection nozzle is fitted centrally into the top of the precombustion chamber, which latter communicates with the cylinder through a necked passage. Into this passage is screwed a device known as a "burner," a sort of nozzle with a number of orifices in its lower end. The "burner" is located between the two valve pockets, where there is no water jacket, and in addition its central section is relieved on the outside, to retard the flow of heat from it to the cylinder-head casting.

A central location of the chamber in the cylinder head is unsuitable for really high-speed engines, because of the restriction it imposes on valve dimensions. It has therefore become the custom to arrange the chamber to one side, symmetrically with relation to the valves, and some designers even place it at an angle of 45° with the cylinder axis. The question then arises as to the best arrangement of the outlet orifices, from the standpoint of optimum dispersion of the fuel through the air charge of the combustion chamber. In one design the chamber is arranged parallel with the cylinder axis and only sufficiently off center so as not to restrict the valve size. The chamber is provided with inclined orifices, those on the side nearest the cylinder wall being made somewhat smaller than those on the opposite side. Another scheme consists in placing the chamber parallel with the cylinder axis

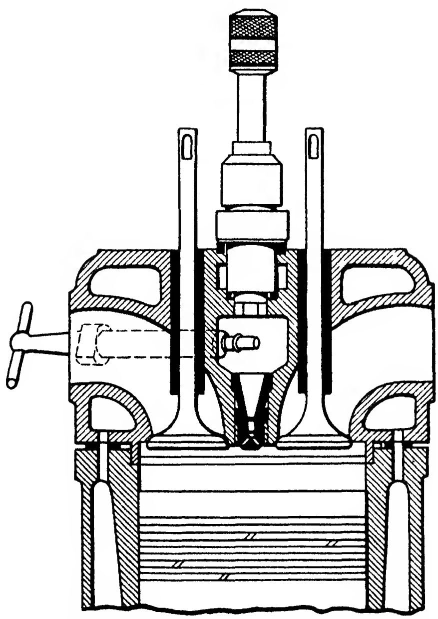


FIG. 3.—BENZ CYLINDER HEAD WITH PRECOMBUSTION CHAMBER.

and close to the cylinder wall, and to provide it with a single inclined outlet. The spray from the chamber will then strike the crown of the piston, where a hot spot will develop that will reflect the discharge and help to disperse and burn it. Care must be taken, of course, that the piston is not overheated at this point, and the section should be thickened or a shield of some special heat-resisting material should be provided there.

The principal manufacturers of precombustion-chamber engines in the United States are the International Harvester

Company and the Caterpillar Tractor Company. Their arrangements of the precombustion chamber are shown in Figs. 4 and 5 respectively. In each case the chamber is located over to one side of the cylinder and makes an angle with the cylinder axis. The chamber is completely water-jacketed and is provided with a steel insert that is held in place by the nozzle holder. In the Caterpillar engine the prechamber constitutes about 28 per cent of the total compression volume. In the American engines the precombustion chamber has a single outlet of relatively large size. No glow plugs are used as an aid to starting, elimination of this somewhat troublesome accessory being made possible by the methods of cranking employed, which will be described farther on.

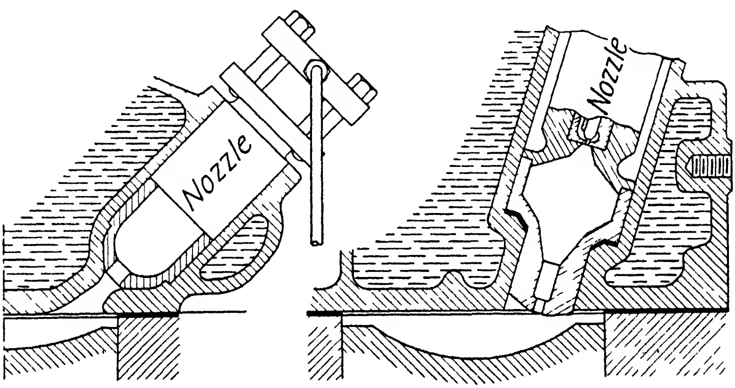


FIG. 4 (*Left*)—I H C PRECOMBUSTION-CHAMBER DESIGN
FIG. 5 (*Right*)—CATERPILLAR PRECOMBUSTION-CHAMBER DESIGN.

Cycle in Precombustion Chamber—Injection begins while the flow of air into the precombustion chamber is taking place, and when the fuel first enters the throat, it encounters a current of air flowing in the opposite direction. Ignition occurs in the throat before any fuel enters the cylinder. A partial combustion of the fuel in the precombustion chamber then takes place, which results first in an equalization of the pressures in the precombustion chamber and in the cylinder, respectively, thereby stopping the flow into the precombustion chamber; and then in an excess of pressure in the precombustion chamber over that in the cylinder, which results in the so-called "blow-off," that is, the forceful ejection of liquid fuel, gases of combustion, and air from the precombustion chamber.

Principles of Combustion—It is, of course, impossible for all of the fuel injected to burn in the precombustion chamber, because far too little air is present in the latter. However, what really limits combustion in the precombustion chamber is the time element.

In the combustion of a homogeneous mixture of hydrocarbon vapors and air, the hydrogen component of the hydrocarbons burns first, and if only enough oxygen were present to burn the hydrogen, the carbon would be precipitated as soot. If sufficient air is present, the carbon, of course, will burn too. Carbon and oxygen together form two different compounds, carbon monoxide (CO) and carbon dioxide (CO_2). However, carbon can combine with oxygen directly only into CO_2. If there is an excess of fuel, then some of the CO_2 will combine with additional carbon (provided the temperature and pressure conditions are favorable) to form CO.

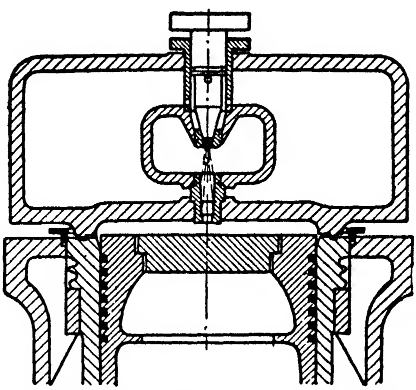


FIG. 6.—KOERTING CYLINDER HEAD WITH PRECOMBUSTION CHAMBER.

In the precombustion chamber, however, we do not have a uniform mixture of hydrocarbon vapors and air, but a very heterogeneous mixture. Therefore, although on the basis of the total amount of fuel injected there would be an excess of fuel in the precombustion chamber, on the basis of the amount of fuel actually available (at a sufficiently high temperature) for combustion, there is generally an excess of air in the precombustion chamber, and no carbon monoxide is formed. In some designs of precombustion-chamber engine the fuel is injected at such a slow rate that even counting all of the fuel injected at any time, there is never an excess of fuel in the precombustion chamber up to the time that the pressure in that chamber exceeds that in the cylinder and fuel is forced from the precombustion chamber.

Analysis of Combustion Phenomena—The following study of the process of combustion in a precombustion-chamber engine is based on research work carried out by Prof. Kurt Neumann of the Hannover Technical College on a single-cylinder Koerting engine (Fig. 6) of 7.5-in. bore and 12.5-in. stroke running at 350 rpm. This, of course, is not a high-speed engine, but the results arrived at give a good idea of what goes on inside an engine of this type. The piston displacement was 547 cu in.; the compression volume in the cylinder, 27.1 cu in., and the volume of the precombustion chamber, 11.9 cu in. Consequently, the compression ratio was exactly 14 to 1 and the volume of the precombustion chamber was a little more than 30 per cent of the total compression volume. Owing to the restricted communication between the two parts of the compression space, during the compression stroke the pressure rose more rapidly in the cylinder than in the precombustion chamber, as may be seen from Fig. 7, in

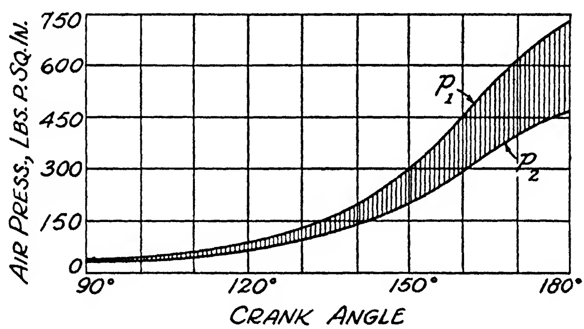


FIG. 7.—PRESSURE IN CYLINDER (P_1) AND PRECOMBUSTION CHAMBER (P_2) DURING LAST HALF OF COMPRESSION STROKE.

which the cylinder and precombustion-chamber pressures are plotted against the crank angle.

It was found that the air flow from the cylinder into the precombustion chamber during the latter part of the compression stroke played an important part in the atomization of the fuel, which in this type of engine is injected under comparatively low pressure. The rate of air flow from the cylinder into the precombustion chamber reached a maximum value about 14° ahead of top dead center.

The delivery stroke of the injection pump began 17° of crank motion ahead of top center, and injection into the precombustion chamber, 14° ahead of top center, or just about when the velocity of air flow into the chamber was at its maximum. Ignition occurred in the precombustion chamber 4.5° ahead of top center. Almost immediately after ignition, the pressure in the precombustion chamber rose above that in the cylinder, and the chamber began to "blow off" into the cylinder, this process continuing for about 90° of crank motion. During this period the composition of the gaseous portion of the "blow" changed continuously, the oxygen content decreasing and the contents of water vapor and carbon dioxide increasing.

Of the fuel injected during each cycle, about 20 per cent burned in the precombustion chamber, the remainder in the cylinder. Up to the time of ignition in the chamber, there

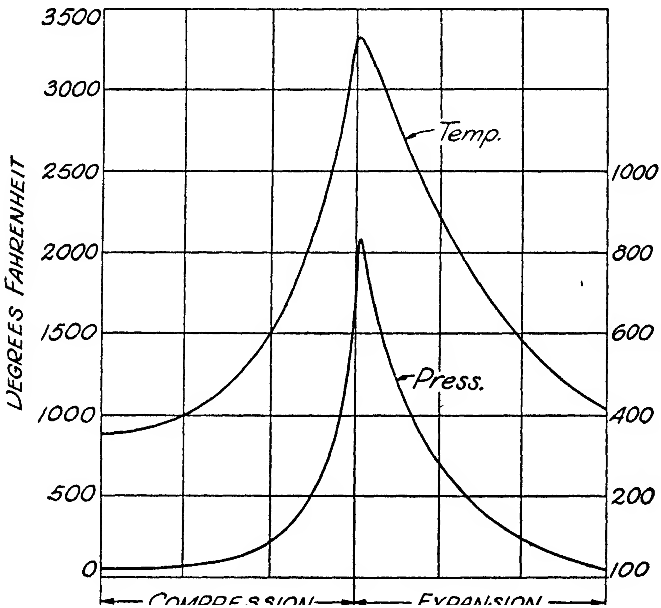


FIG. 8.—CURVES OF PRESSURE AND TEMPERATURE IN PRECOMBUSTION CHAMBER.

was not enough fuel present to require all of the air for its combustion. In fact, at this time, when the amount of air in the chamber was a maximum, there was an excess of air of 46 per cent in the chamber. This excess of air tends to prevent carbon deposits on the walls of the precombustion chamber.

A characteristic of the engine on which the tests were made is that both the precombustion chamber and the passage from it to the cylinder are completely water-cooled. The mean temperature of the chamber wall ranged from 170 F on the outside to 234 F on the inside, and the mean temperature of the passage wall from 162 F on the outside to 362 F on the inside.

In Fig. 8 are shown curves of pressure and temperature in the precombustion chamber during the compression and power strokes. During the other two strokes both the pressure and the temperature in the precombustion chamber remain substantially constant, the temperature at about 875 F. The maximum pressure reached was 840 psi, and the maximum temperature, 3375 F.

A point that needs consideration is the evacuation of the precombustion chamber. There is no outlet from this chamber except that into the cylinder, and at the end of the exhaust stroke the chamber is filled (mainly with products of combustion) at the same pressure as the clearance space in the cylinder. A small amount of the contents of the chamber will be drawn from it during the suction stroke, when the pressure in the cylinder drops further, so that at the beginning of the compression stroke the chamber is filled to substantially atmospheric pressure with gases consisting mainly of products of combustion. During the compression stroke, however, the entire charge of the cylinder and precombustion chamber is compressed to one-fourteenth its original volume, with the result that at the end of the compression stroke the proportion of pure air in the precombustion chamber is very large.

Injection of fuel into and combustion in the precombustion chamber and cylinder respectively, together with related factors, are plotted in Figs. 9 and 10. In Fig. 9 are shown curves representing the amount of fuel burned in the precombustion chamber, the pressure in the chamber, and the blow-off velocity or velocity of flow from chamber to cylinder, in relation to the crank angle. The chart also indicates the point at which ignition occurs, viz., $-5°$. In Fig. 10 are shown the total amount of fuel injected (0.000915 lb), the amount of fuel blown into the cylinder (0.000726 lb), the amount of fuel burned, the coefficient γ of excess air, the rate of combustion, the point at which the fuel valve opens ($-13°$), and the point of ignition ($-5°$).

Interruption in the Blow-Off—Recent tests on precombustion-chamber engines, in which pressure-time curves of both the precombustion chamber and the main chamber were superposed on the same card, have shown that the blowing-off process is not a continuous one. Throughout the compression stroke the pressure is higher in the main chamber than in the precombustion chamber, because of the smallness of the orifice through which the two communicate. When ignition takes place in the precombustion chamber, the pressure therein rises above that in the main chamber, and the blow-off starts.

It continues until the pressures in both chambers are equalized. Then, since the pressure in the main chamber is reduced rapidly by the expansion (downward movement of the piston), the pressure in the precombustion chamber once more rises above that in the main chamber, and the blow-off starts again. This renews combustion in the main chamber, and the resulting liberation of heat in the main chamber, together with the loss of charge by the precombustion chamber, equalizes the pressures once more and definitely stops the blow-off. It sometimes happens that the precombustion chamber is so poorly cleared of fuel that preignition occurs during the following cycle; that is, ignition takes place before any fuel is injected.

Brake Mean Effective Pressure—The average bmep at rated horse power of twenty-four precombustion-chamber engines of which the author has the data is very close to 80 psi. The average bmep of a somewhat smaller number of direct-injection engines is practically the same, and from this point of view the two types therefore are practically on a par.

International Tractor Engine—Figs. 11A and 11B are a longitudinal section of an engine manufactured for installation in farm tractors by the International Harvester Company, and Fig. 12 is a cross section of that engine. The latter view shows the arrangement of the precombustion chamber and the location of the injection equipment. The engine is equipped with an injection pump of special design, of which an illustrated description was given in Chapter VII. These engines are being manufactured in six models, three four-cylinder and three six-cylinder. The one represented by the drawings is the four-cylinder with a bore of 4¾ and a stroke of 6½ in., which is rated 76 hp at 1400 rpm. This rating corresponds to a bmep of 93 psi.

The cylinder block and crankcase are in a single casting which is provided with "wet" cylinder liners. The valve arrangement is rather unusual, due to the fact that the engine is started as a spark-ignition engine on gasoline, as described in detail in Chapter XVI. It will be observed that the pistons have a great deal of metal in the crown, which helps to keep down the temperature around the upper piston rings. There are six rings on each piston, four narrow compression rings and two oil control rings, of which one is below the piston pin.

All bearings, including those of the piston pins, are pressure-oiled, a gear-type oil pump being located in the deepest part of the sump, supplying oil to the main bearings through a main header drilled in the crankcase wall. A compression ratio of 15.5 is used. The weight of the engine is 1775 lb,

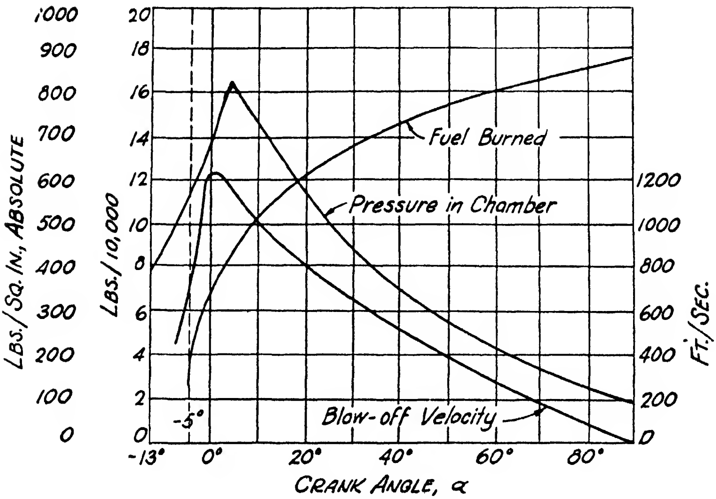


Fig. 9.—Precombustion-Chamber Phenomena.

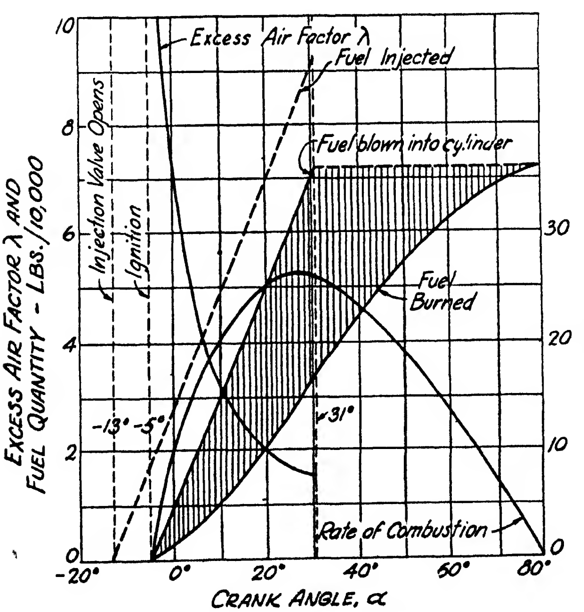


Fig. 10.—Cylinder Phenomena.

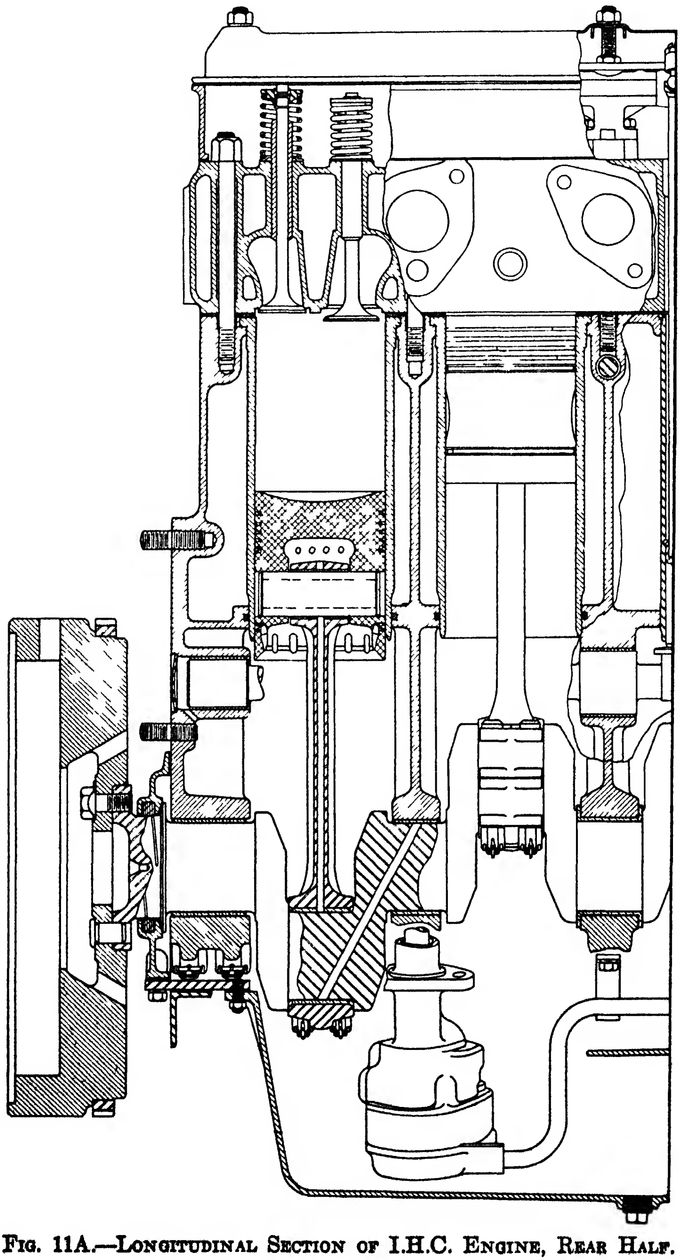


FIG. 11A.—LONGITUDINAL SECTION OF I.H.C. ENGINE, REAR HALF.

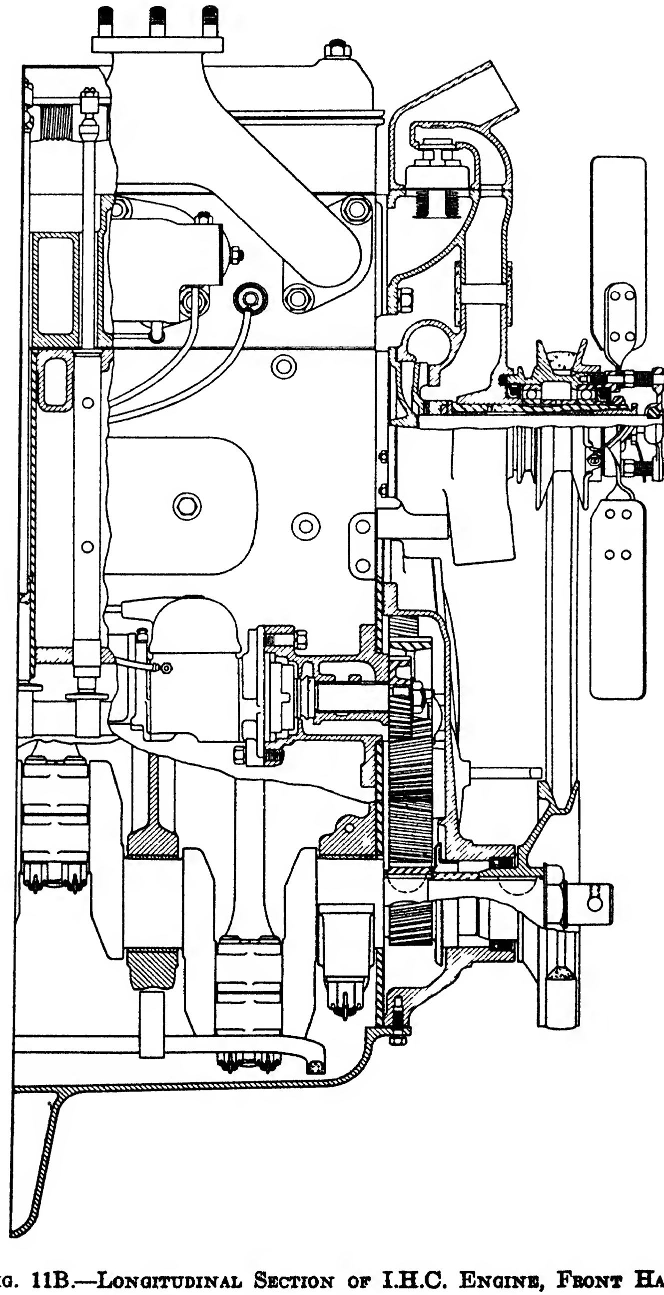


FIG. 11B.—LONGITUDINAL SECTION OF I.H.C. ENGINE, FRONT HALF.

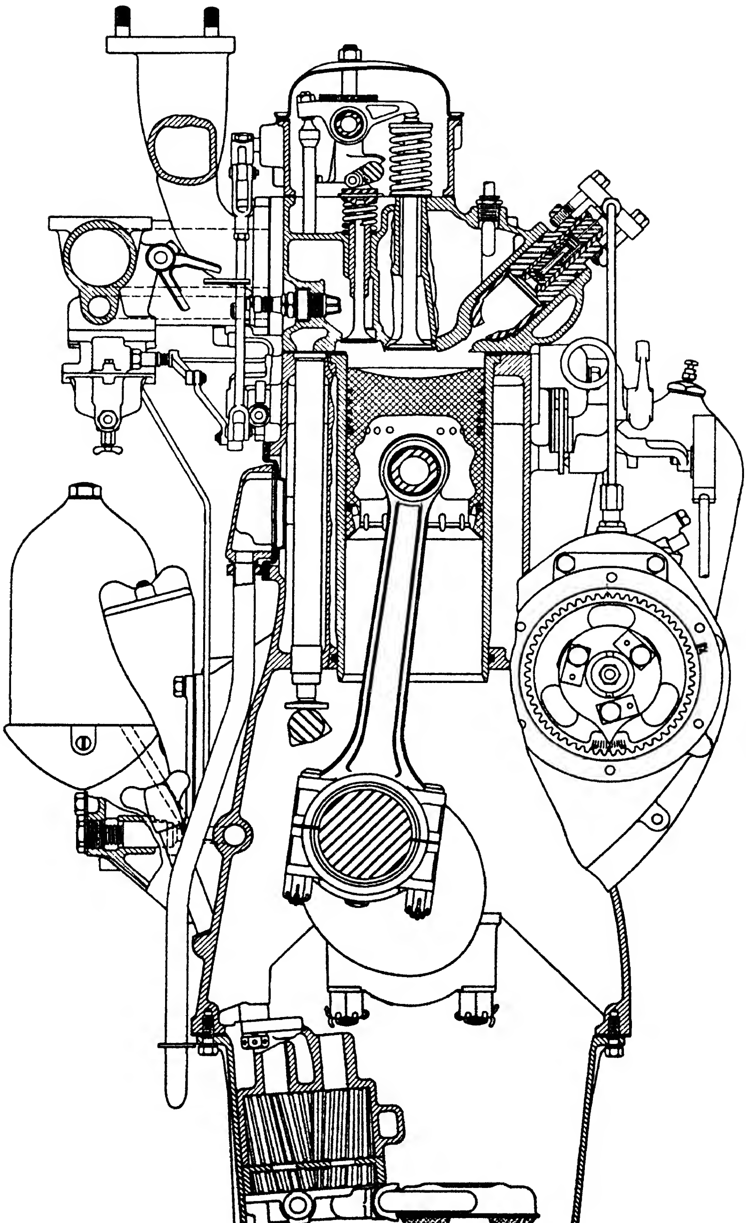


Fig. 12.—Cross Section of I.H.C. Diesel Engine.

which makes the specific weight a little more than 23 lb per horse power.

Caterpillar Tractor Engines—Diesel engines specially designed for use on farm and industrial tractors have been developed by the Caterpillar Tractor Company, Peoria, Ill. One of these has six 5¾ by 8-in. cylinders and develops 145 hp at 1000 rpm. Since the displacement is 1246 cu in., this output corresponds to a bmep of 92 psi. A longitudinal section of a Caterpillar engine is shown in Fig. 13. The following description applies to the 145 hp engine.

The construction is very rigid, cylinder shell and crankcase down to several inches below the crankshaft axis being in a single iron casting. Cylinder liners of alloy iron, heat treated, are inserted in the block, leak-proof joints being effected at the upper end by means of a soft copper gasket and at the bottom by two packing rings of oil-proof soft rubber. Gas-pressure stresses are taken up on "through" studs which hold both the cylinder head and the crankshaft bearing caps in place. Inspection holes are provided in the cast-iron sump.

The crankshaft has seven main bearings, 3¾ in. in diameter. End thrust is taken up on a bronze and cast-iron thrust bearing at the forward main bearing. Cylinder heads are in two alloy iron castings, each covering three cylinders. The seat of the injection valve, the precombustion chamber, and the burner tube are a single unit which is pressed into the cylinder head. Its axis makes an angle with the cylinder axis and it is sufficiently offset from the cylinder axis so as not to reduce the possible valve diameter. Means are provided for holding the exhaust valves off their seats during the starting period. There are cylindrical shields around the exhaust valve stems inside the valve pockets, protecting the stems from the hot exhaust gases. These are intended to prevent burning of the lubricating oil on the stems and consequent sticking of the valves in their guides.

Injection pumps and injection valves of the company's own design are being used. Fuel is delivered to the injection pump through a five-stage filter at a pressure of 15 psi by a gear-type transfer pump. The filters are of the absorbent type and are said to exclude even very fine particles of dirt. There is a separate pump for each cylinder, but all of the pumps are mounted in a common housing at the side of the cylinder block, from which they can be removed after loosening the connection of the high-pressure fuel tube and taking out a number of cap screws. All fuel pumps are interchangeable, and pumps are not serviced in the field or locally, but replaced by new units when necessary. The same applies to

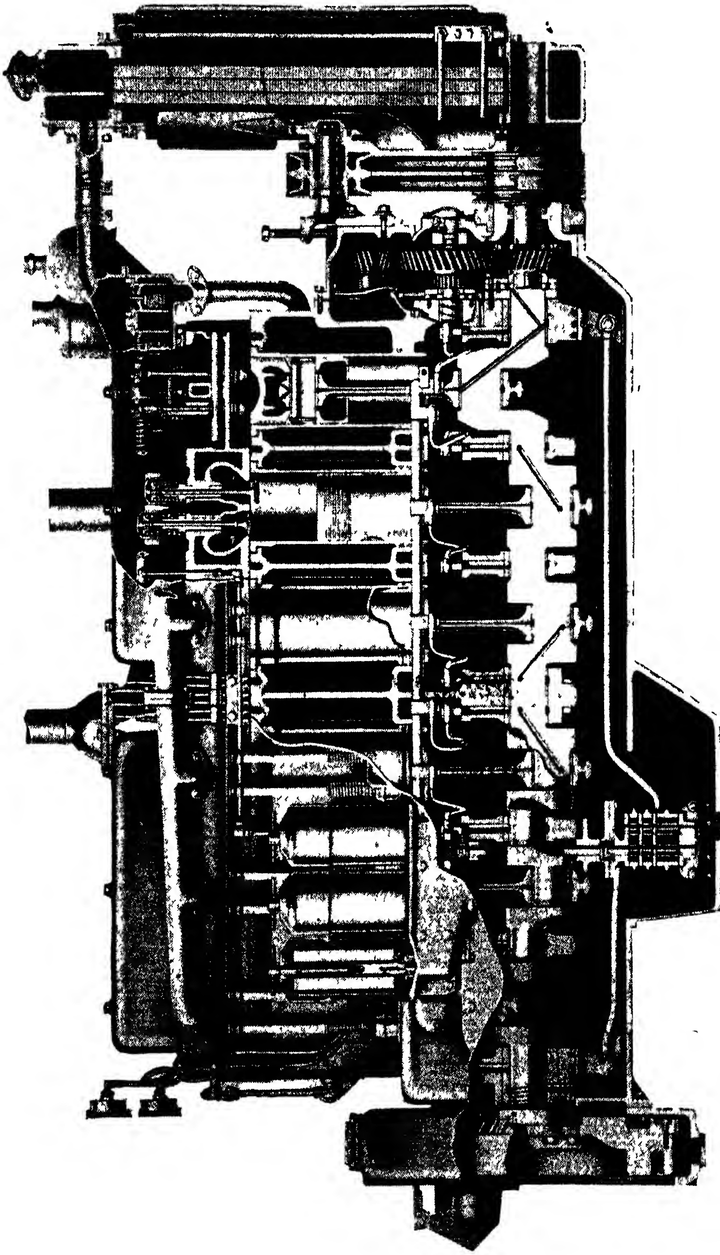


FIG. 13.—CUTAWAY VIEW OF CATERPILLAR 145-HP DIESEL ENGINE.

the injection nozzles. Quantity control is effected by rotating the pump plungers in the barrels by means of a rack which is connected to the centrifugal governor. The injection nozzles are located outside the valve cover, so they are quite accessible.

A dry-sump system of lubrication is employed. In addition to the pressure unit of the pump, there are two scavenging units which draw oil from both ends of the crankcase and deliver it into the sump. The oil pump, which is located in the sump, draws in oil through a screen with large area, and on its way from the pump to the bearings the oil is forced through three large oil filters outside the engine.

Daimler-Benz Rail Car Engine—It was stated in the foregoing that the precompression type of Diesel engine originated with Benz & Co. of Mannheim, Germany. The succession firm, Daimler-Benz Company, has produced this type of engine in great variety, from a four-cylinder 24 hp for small passenger cars to a 20-cylinder V of 3000 hp for fast naval vessels. Fig. 14 shows a cross section of a 12-cylinder supercharged, 800-hp engine for rail car installation. Cylinder dimensions are 6.78-in. bore by 8.08-in. stroke, which makes the displacement equal to 3490 cu in. The cylinder banks and the upper half of the crankcase are in a single casting of aluminum alloy. Set into this block are wet cylinder liners of alloy iron having a Brinell hardness of 240-250. There are two inlet and two exhaust valves in each cylinder head, and the precombustion chamber is located centrally between them, in line with the cylinder axis. It will be seen that the so-called ''burner'' at the outlet from the precombustion chamber is continued. Its design is said to have been refined to improve the fuel economy, the refinement apparently consisting in an enlargement of the orifice or throat and rounding of its edges.

The crankshaft is forged of low-carbon steel and case-hardened to 60-64 Rockwell, to increase its wear resistance. It has six counterweights and is supported in lead-bronze bearings. It is of such robust design that there are no critical periods within the speed range. Connecting rods are mounted side by side on the crankpins, hence all twelve are alike and interchangeable.

This engine can be equipped with either a mechanically-driven blower or a Buchi turbocharger. It delivers its rated output of 800 hp at 1400 rpm (128 psi bmep), with a specific fuel consumption of 0.38 lb per bhp-hr. As the engine is supercharged, the performance figures cannot be directly compared with those of other engine types with natural aspiration.

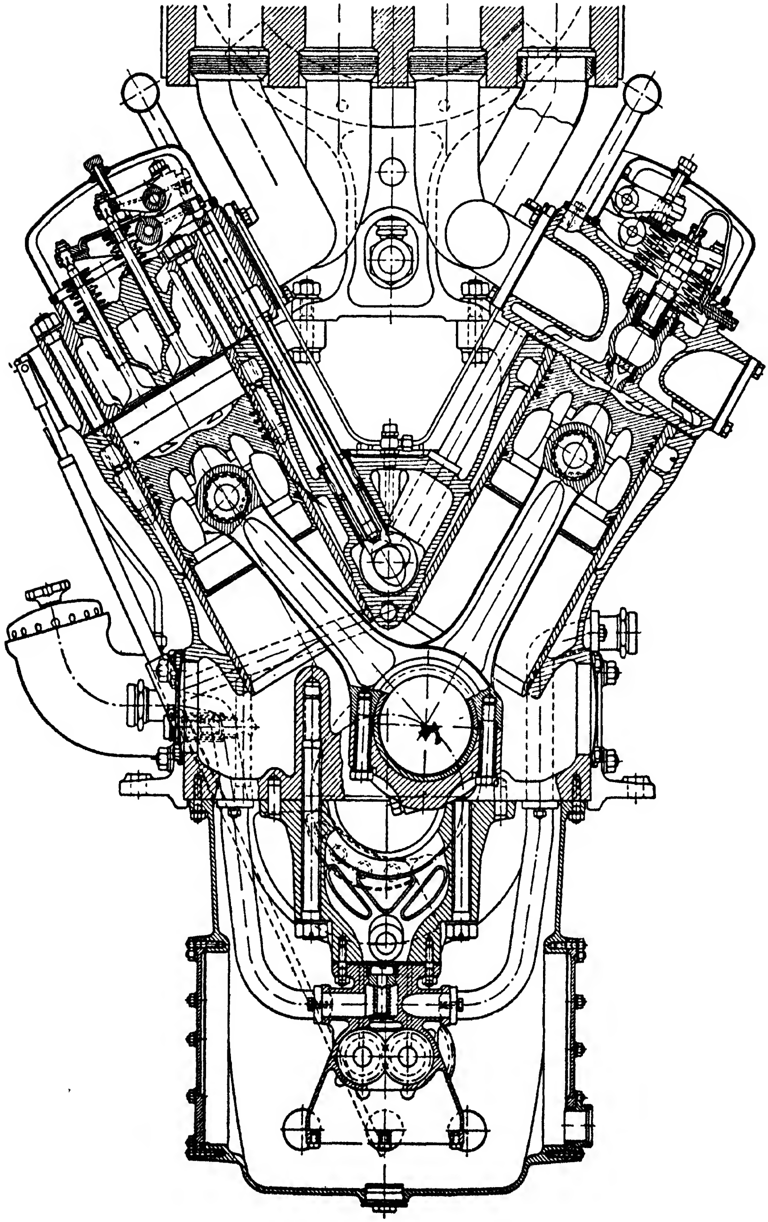


FIG. 14.—CROSS SECTION OF DAIMLER-BENZ 800-HP RAILWAY ENGINE. (FROM *Motortechnische Zeitschrift*)

CHAPTER IX

Direct-Injection Engines

Engines in which the fuel is injected directly into the main combustion chamber may be divided into two classes, viz., those in which the air is comparatively quiescent during the period of fuel injection, and those in which there is controlled air flow in the combustion chamber, calculated to assist in the distribution of the fuel throughout the air charge. Engines with air flow may be divided again into four sub-classes, according to the nature of the flow, which may take the following forms:

1. Swirl around the cylinder axis.
2. Radial flow toward the cylinder axis.
3. A combination of these two.
4. Flow across the combustion chamber.

There are also engines in which each cylinder is provided with an air-storage chamber, communicating with it through a narrow passage or throat, into which air is forced during the compression stroke and from which it escapes during the power stroke, but these "auxiliary-chamber engines" form a class by themselves and will be discussed in a separate chapter. Here again the boundary between the two classes is not clearly defined. In addition to engines in which the combustion chamber is in the form of a flat cylinder between the piston and cylinder head, there are others in which the greater part of the combustion space is constituted by a chamber either in the cylinder head or in the piston. These will be considered as of the auxiliary-chamber type only if the communicating passage between chamber and cylinder has a definite restricting effect.

In all automotive-type two-stroke engines the fuel is injected directly into the combustion chamber, but in this chapter only four-stroke engines will be dealt with.

Quiescent Combustion Chambers—If the air in the combustion chamber is substantially at rest while the fuel is being injected, atomization of the fuel and penetration of the spray must be depended upon to distribute the fuel particles. It is then necessary to so coordinate the form of the sprays and the form of the combustion chamber that fuel will be sprayed into practically every part of the chamber. If this is neglected the result will be a very low mean effective pressure and a high ratio of excess air at maximum output with smokeless exhaust.

The natural form of the combustion chamber of an engine having its valves in the head is a very flat cylinder. For instance, if the stroke is, say, 25 per cent greater than the bore, and a 15 to 1 compression ratio is used, the compression space, if of flat cylindrical form, has a depth only about one-eleventh its diameter. Such a form is undesirable for two reasons. In the first place, without the aid of air flow it is difficult to properly disperse the fuel in it; secondly, at the time of maximum compression, all of the air within the combustion chamber is relatively close to a wall through which the heat of compression is carried off, and none of the air therefore gets as hot as if the compression space were of more compact form

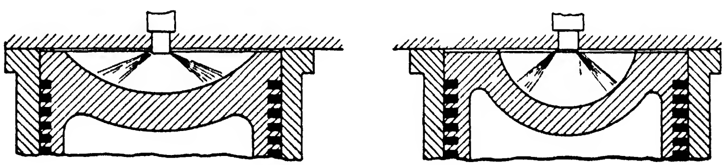


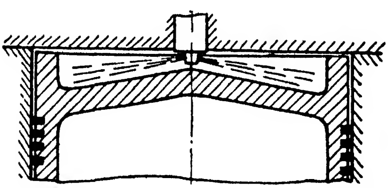
FIG. 1 (*Left*)—DISHED TYPE OF PISTON CROWN.
FIG. 2 (*Right*).—HEMISPHERICAL PISTON CROWN.

and the distance from its center to the cooling wall surface were greater.

Some of the early submarine engines had a slightly dished piston crown, with the injector located in the center of the cylinder head and directing a number of sprays at an angle of about 30° from the horizontal, as shown in Fig. 1. Naturally, if the top of the piston and the bottom of the cylinder head are not parallel surfaces, there will be a certain air flow as the piston approaches the top of the stroke, and as it starts on the down-stroke. In this case the flow is radially inward and outward.

Hemispherical Combustion Chamber—By providing the piston with an upwardly extending rim which comes close to the cylinder head at the end of the up-stroke, a hemispherical depression may be formed in the center of the piston head, of such a diameter as to give the desired clearance (Fig. 2). If the nozzle is then provided with five or six orifices making an angle of 45° with the cylinder axis, the fuel will be fairly well distributed. As the piston approaches the cylinder head, the inward rush of air tends to break through the sprays, and the following outward flow will tend to carry fuel particles toward the cylinder wall. One difficulty with this arrangement for small engines is that the diameters of the orifices become exceedingly small, and such orifices not only are difficult to produce accurately, but they also become choked easily. In considering the best form of the combustion chamber from the standpoint of fuel distribution, not only its form at the end of the up-stroke, but the range in form from the beginning to the end of the injection period must be taken into account.

A further development in combustion chambers is represented by Fig. 3, this type having been used by Hesselman, M.A.N., and others. The piston crown has a flange running around it and is cone-shaped, sloping down from the center



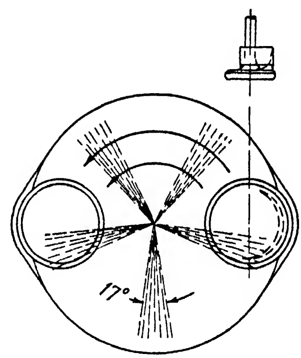


Fig. 3 (*Left*)—Piston with Circumferential Flange on Top.
Fig. 4 (*Right*).—Air Swirl Created by Masked Valve.

toward the flange. The angle between the sloping surface and the bottom surface of the cylinder head is made to correspond to the angle of the spray cone. An injector with four or five orifices is located at the center of the cylinder head. An advantage of this type of combustion chamber is that the spray does not come in contact with the water-cooled cylinder wall; besides, there is no danger of the fuel of any missed charges running down the cylinder walls and diluting the crankcase oil.

Location of Spray Nozzles—In large, low-speed engines it is the common practice to place the spray nozzle at the center of the cylinder head, as that is about the only point in which it can be symmetrically located with respect to the combustion chamber, so the spray may reach all parts of the chamber equally. But having the nozzle centrally in the cylinder head necessarily reduces the space available for the valves—assuming that there are two valves per cylinder—and in a high-speed engine it is desirable to have these as large as possible. The requirements affecting the location of the spray nozzle are therefore rather conflicting. One solution of the problem consists in using four valves per cylinder, or what is usually referred to as dual valves. This has been done in engines with comparatively large cylinders, and the practice is now being extended to smaller cylinders, in spite of the fact that it involves complications, especially in connection with the valve-operating mechanism.

Swirling Air Flow—With the inlet valve located in the cylinder head, a swirling flow of the air in the cylinder can be produced by means of a so-called masked valve. As illustrated in Fig. 4, this is a poppet valve having a flange extending part-way around the stem side of the head, the height of the flange being slightly greater than the valve lift. Provisions are made so the valve cannot turn in its guide, and the flange is located unsymmetrically with respect to the plane through the valve axis and the cylinder axis. The air then enters the cylinder tangentially and swirls around in it, as indicated by the arrows, the swirling motion being maintained throughout the compression stroke. As a help in distributing the fuel through the air charge, this method of inducing air flow has merit, no doubt, but the capacity of the inlet valve is reduced by the mask. Other disadvantages are that the valve assembly is more expensive to produce and that the valve itself weighs more and therefore calls for the use of a heavier spring. Masked valves have never been used in production engines in the United States, so far as is known to the author, but they are in extensive use in European countries, where the direct-injection engine has come to the front on account of its higher fuel economy.

Bowl-Shaped Combustion Chamber—In England a type of combustion chamber which may be regarded as a development of that shown in Fig. 3 has come into general use. As shown in Fig. 5, the chamber is in the crown of the piston, but its diameter is little more than one-half that of the piston and its depth about one-half its diameter. The rim of the piston is made to approach the cylinder head at the end of the up-stroke as closely as working conditions permit; hence, toward the end of the up-stroke a strong inward flow of air (referred to as the "squish" by British writers) is produced, and this air motion is superimposed on the swirling motion due to the masked valve. As the diameter of the swirling body of air is reduced from that of the cylinder to that of the chamber in the piston, the angular velocity of the swirl is materially increased. At the same time swirls in vertical planes are set up in the chamber, which assist further in mingling of the fuel with the air charge. It has been found that a central "mound" at the base of the air chamber improves the combustion, apparently because it facilitates vertical movement of the air.

In the A.E.C. engine, which employs this type of combustion chamber together with masked inlet valves, the spray nozzle, located centrally with respect to the chamber in the piston, has four orifices which produce sprays making angles of 75 deg with the cylinder axis. These sprays cut across the air swirl produced by the masked inlet valves, and the vertical

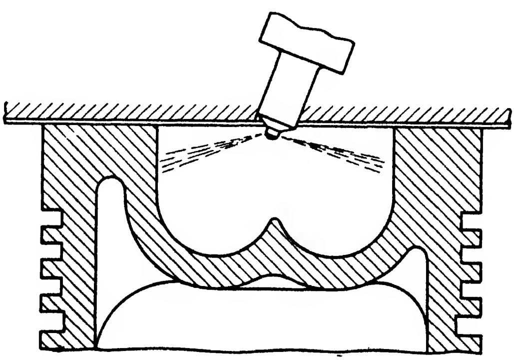


FIG. 5.—BOWL-SHAPED COMBUSTION CHAMBER IN PISTON.

movement of the air, produced by the "squish" and facilitated by the "mound," serves to bring the whole of the air charge into proximity with the sprays.

Effect of Angular Position of Valve Mask—The effect of a masked inlet valve in increasing engine output is shown by Fig. 6, which is based on an illustration in an S.A.E. paper by C. B. Dicksee, research engineer of the Associated Equipment Co., Ltd. To determine the optimum form of the mask, tests were begun with a mask extending nearly halfway around the valve head, and the angular width was gradually reduced. With a mask of an angular width of 80° the bmeps at different engine speeds were decidedly higher than with a 140° mask. In the experimental setup provisions were made to rotate the valves through a whole circle, and the effects of the angular position of the mask on the bmep at both 1000 rpm and 1500 rpm are shown in Fig. 6. It can be seen that while there is a difference of about 20 psi between the bmeps for the optimum and the worst positions of the valve, the peaks of the curves are quite flat, indicating that positioning of the mask is not critical.

Tangential Valve Passage—An effect similar to that of the masked valve can be produced by making the valve passage outside the inlet-valve port of a particular form. Ordinarily this passage has approximately the form of a pipe "ell." By making it substantially straight and at an angle of, say, 45° to the plane of the valve seat, the air can be made to enter the cylinder tangentially instead of in a direction parallel to the cylinder axis. If the axes of the inlet and exhaust valves are in a vertical plane through the crankshaft

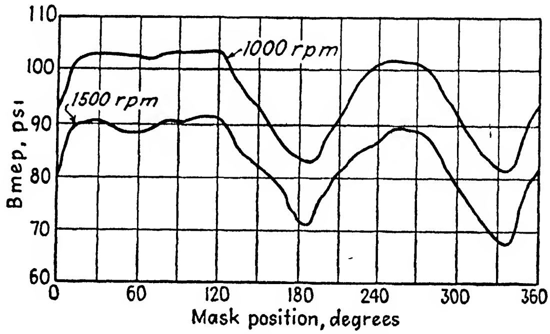


Fig. 6.—Variation of BMEP with Angular Position of Masked Valve.

axis and the inlet passage extends down to the valve seat from the side of the cylinder head, the air will have a swirling motion imparted to it as it enters the cylinder.

Displacer Piston—Another arrangement designed to produce energetic air flow in the combustion chamber during the period of injection is what has been referred to as the displacer piston. Where this is used, the two valves are arranged coaxially in the cylinder head, at right angles to the cylinder axis, the greater part of the compression space being between the valve heads. The compression space is connected to the cylinder by a passage of limited cross section, and as the piston approaches the top end of the stroke, there is a very violent surge of air through the passage, which naturally means great turbulence in the combustion chamber. One such design, which has been used by the Deutz Motor Works in Germany, is illustrated in Fig. 7. In this case the piston is provided with a central extension (or a pilot) which enters the "throat" leading to the space between the valves, leaving only a very restricted passage between it and the wall of the "throat."

A similar design but without the pilot on the piston head, which has been used by Ruston-Hornsby, in England, is illustrated in Fig. 8.

N.A.C.A. Experiments with Displacer Pistons—Very satisfactory experimental results from an engine with a displacer piston were obtained by the National Advisory Committee for Aeronautics. The combustion chamber of the 5 by 7-in. single-cylinder experimental engine was of flat-disc shape and had

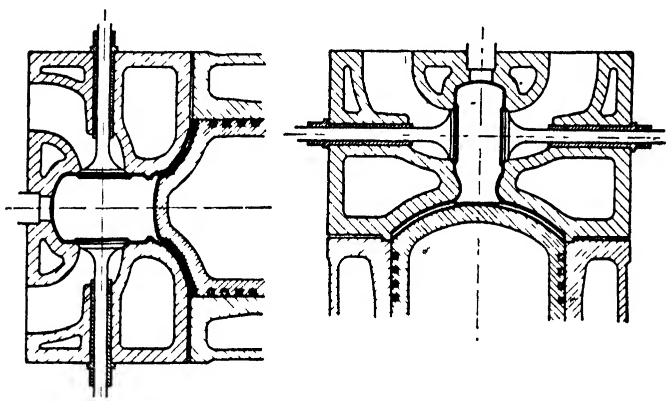


FIG. 7 (*Left*).—DEUTZ CYLINDER HEAD AND DISPLACER PISTON.
FIG. 8 (*Right*).—RUSTON-HORNSBY CYLINDER HEAD.

the form shown diagrammatically in Fig. 9. The displacer, which was of rectangular section, fitted the throat fairly closely at the sides, but left passages for air flow at both ends of the rectangle. A compression ratio of 15.3 was used. It was found that the optimum width of each of the two flow areas was $^{19}/_{64}$ in., which gave a flow velocity equal to eight times the instantaneous piston speed. Performance was not greatly affected by the height of the displacer, and that which allowed the displacer to enter the throat at 42° ahead of top center was found to be best.

The performance of this engine at 1500 rpm was compared with that of another similar one with a quiescent combustion chamber, the compression ratio being the same in both. At the limit of clear exhaust, the bmep was 90 psi in the engine with quiescent combustion chamber and 115 in the one with air flow. At these loads the maximum cylinder pressure was slightly greater in the quiescent combustion chamber, while the specific fuel consumptions were substantially the same. The main difference between the performances of the two engines was that as the fuel quantity was increased beyond that giving approximately 70 psi bmep, the bmep curve of the engine with air flow continued to rise fairly uniformly, while that of the quiescent engine dropped away rapidly. With the displacer piston the specific fuel consumption dropped as low as 0.41 lb per hp-hr.

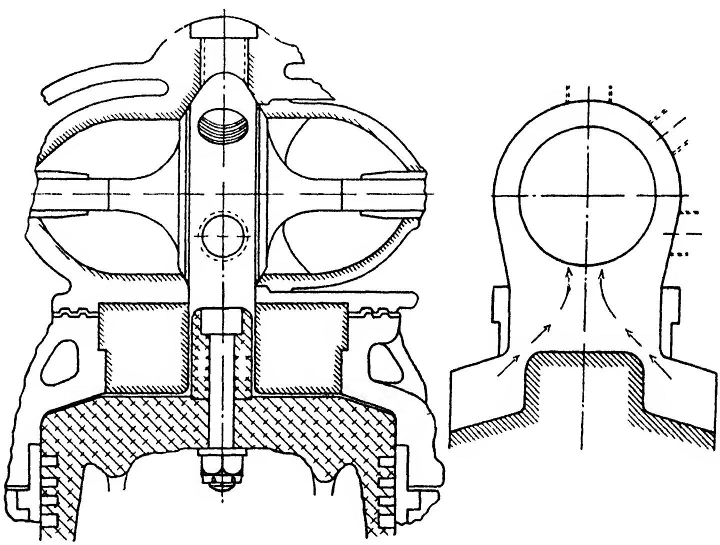


FIG. 9.—COMBUSTION-CHAMBER FORM OF N.A.C.A. TEST ENGINE.

Performance Data—As with other types of high-speed Diesel, the specific output and the fuel economy of the direct-injection type have been gradually improved, chiefly by refinements designed to produce better utilization of the air charge and better timing of the combustion. Some pertinent data are contained in C. B. Dicksee's S.A.E. paper already referred to. When his firm first started the development of direct-injection engines, the best they could obtain was a bmep of 85 psi and a fuel consumption of 0.52 lb per bhp-hr, and these results generally were accompanied by an evil-smelling exhaust that was by no means clean. With their latest engine, which has a combustion chamber of the type shown in Fig. 5, the bmep varies between 110 and 120 psi within the speed

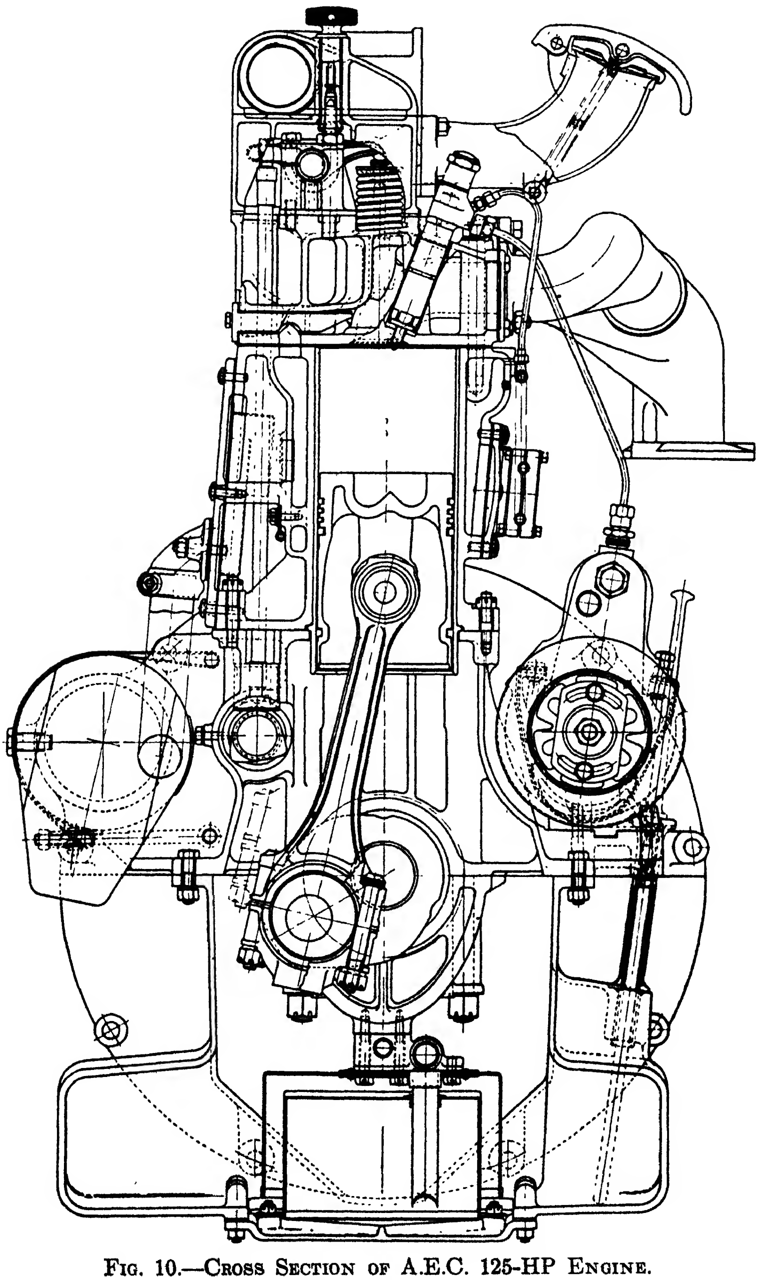


FIG. 10.—CROSS SECTION OF A.E.C. 125-HP ENGINE.

range of 1000 to 1400 rpm, and the exhaust is said to be no more offensive than that of a good gasoline engine. The fuel consumption is between 0.37 and 0.38 lb per bhp-hr, and drops to 0.36 lb if the load is reduced to between 70 and 95 psi bmep.

A cross section of an A.E.C. engine is shown in Fig. 10. It has six 120 by 142-mm cylinders (approximately 4¾ by 5⅝ in.), and is rated 125 hp. The maximum bmep is 111 psi at 1000 rpm. For many years the firm governed its engines at 2000 rpm, but the governed speed was later reduced to 1700 rpm, because the fuel economy drops off rather rapidly in the higher speed range, owing to a rapid increase in the friction horse power. Owing to the high cost of fuel, British operators of commercial vehicles naturally place great weight on high fuel economy. The firm also manufactures a larger engine of very similar design, with 130 by 142 mm cylinders, and both engines are also produced in the horizontal form for under-floor mounting in buses. The 125-hp engine weighs 1680 lb.

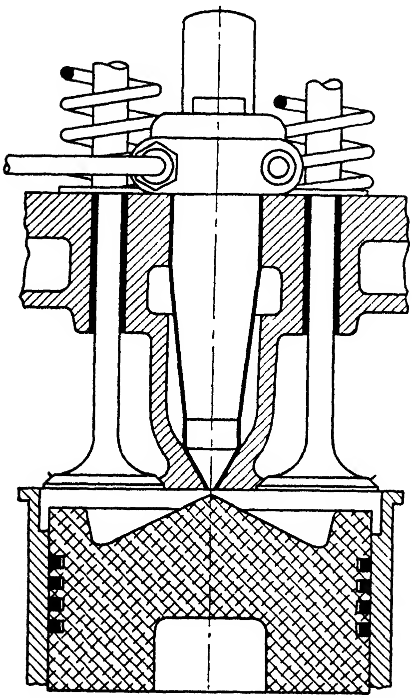


FIG. 11.—COMBUSTION CHAMBER OF CUMMINS ENGINE.

Cummins Engine—Comparatively few makes of American high-speed, four-stroke Diesel engines work with direct injection. Among these is the Cummins, which, however, does not have solid injection, but injection of an oil fog by means of a cam-actuated plunger in the injector. Fig. 10 is a section through the upper part of the cylinder and the cylinder head of a Cummins engine, and shows the form of the combustion chamber and the location of the injector between the two valves in the head. In general form the combustion chamber is not unlike that shown in Fig. 3.

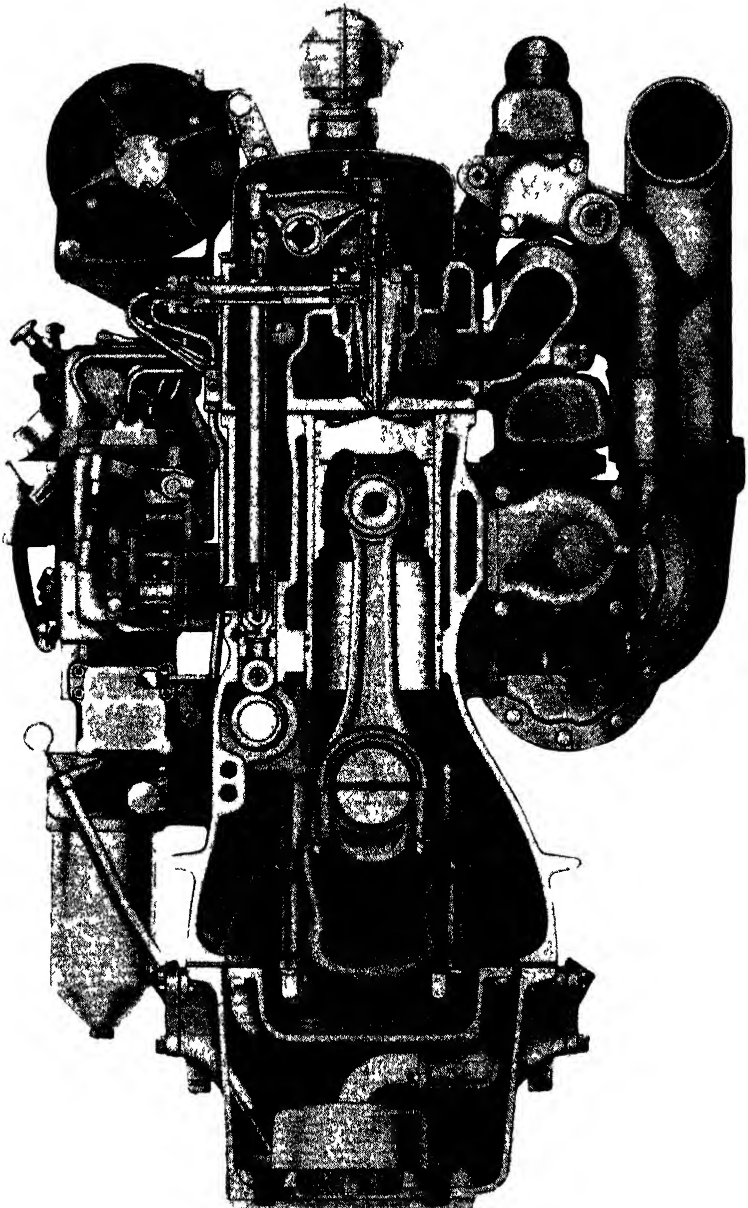


FIG. 12—CROSS SECTION OF CUMMINS ENGINE.

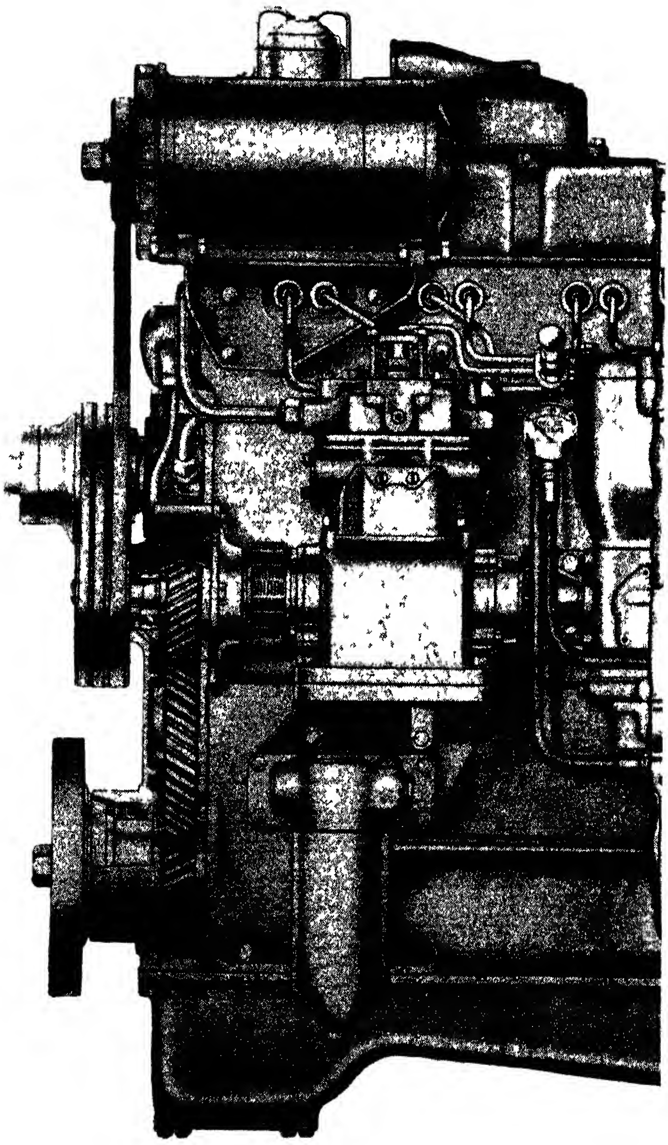


FIG. 13A.—LONGITUDINAL SECTION OF CUMMINS ENGINE, FORWARD HALF.

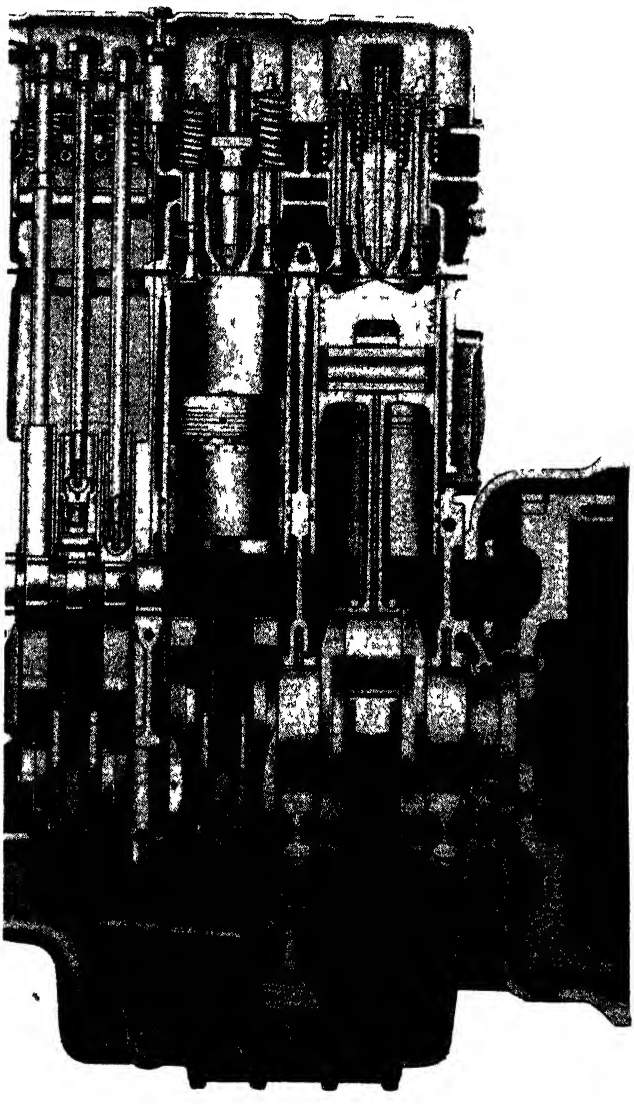


FIG. 13B.—LONGITUDINAL SECTION OF CUMMINS ENGINE, REAR HALF.

The Cummins, which was the first American high-speed or automotive-type Diesel engine to be placed on the market, is now being manufactured in numerous models with from four to twelve cylinders, some of them supercharged. Outputs range from 100 to 550 hp. A six-cylinder model of $5\frac{1}{8}$-in. bore by 6-in. stroke is rated 200 hp at 2100 rpm maximum, and 130 hp at 1800 rpm for operation under continuous load. This same model when supercharged is rated 300 hp at 2100 rpm maximum and 188 hp at 1800 rpm under steady load. The supercharger, which is of the Roots-blower type, is mounted at the side of the engine and positively driven through the front-end gears. Maximum ratings of the two models correspond to 101 and 152 psi bmep, respectively. With atmospheric induction a compression ratio of 15.5 is used, while the supercharged model has a compression ratio of 12.

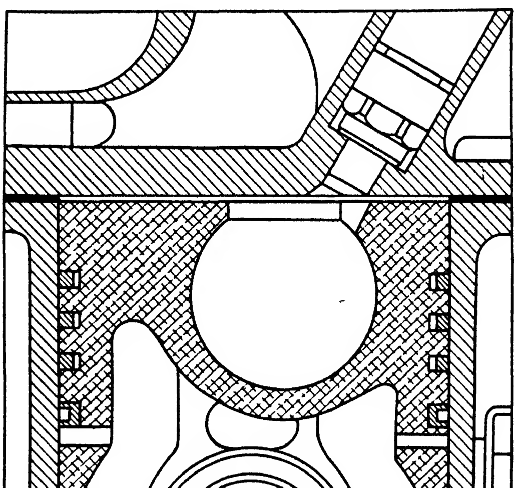


FIG. 14.—SPHERICAL COMBUSTION CHAMBER OF M.A.N. ENGINE.

Fig. 12 is a cross section and Fig. 13 (A and B) a cutaway side view of the Cummins six-cylinder $4\frac{1}{8}$ x 5-in. engine (406 cu in., 150 hp at 2500 rpm). Cylinders and crankcase are in a single casting, the cylinders being provided with "wet" liners. Pistons are of aluminum alloy and carry three compression rings and one oil ring each. Indicative of the rigidity of the design is the fact that the seven main bearings supporting the crankshaft have a diameter of 3.88 in. Piston pins are 1.5 in. in diameter. Among the notable features of the engine are that the crankshaft has counterweights on all arms, that all main bearings are interchangeable, and that the very rigid "big-end" caps are held to the rods by U-bolts and nuts. This permits of gradually merging the flanges of the shank into central reinforcing ribs on the upper half of the "big end," thus avoiding points of stress concentration which often result from milling seats for the connecting-rod bolts.

M.A.N. Engine—The Maschinenfabrik Augsburg-Nürnberg, which has been manufacturing automotive-type Diesel engines since 1923, started out with an engine having a flat cylindrical combustion chamber into which fuel was sprayed by two injectors from opposite sides. It later changed to the air-chamber type described in Chapter XI, for one reason because with injection nozzles on both sides the fuel connections were rather obtrusive. In 1939 the firm returned to the direct-injection type, because of its higher fuel economy, but the new engine had a combustion chamber of spherical form in the piston, as shown in Fig. 14. The small offset of the combustion-chamber center from the piston axis probably is due to a desire to so position the injection nozzle that it will permit the use of valves of maximum diameter in the cylinder head. A fuel consumption of 0.375 lb per bhp-hr is claimed for the truck engine.

Maybach railroad engines, described in Chapter XIII, have spherical combustion chambers in the cylinder head, while engines built by the Austrian Saurer Works in Vienna have combustion chambers with a vertical section of elliptic form in the piston. None of these engines have masked valves, all of them depending on the "squish" at the end of the upstroke to produce in the combustion chamber the turbulence necessary to carry fuel to all parts of the air charge.

CHAPTER X

Turbulence-Chamber Engines

It was pointed out in a preceding chapter that turbulence-chamber engines are similar in form and arrangement of parts to the precombustion-chamber type, the main difference between them being one of proportions, in that the separated portion of the combustion chamber and the cross-sectional area of the passage (or passages) between the latter and the main chamber are smaller in the precombustion-chamber engine. It would be difficult, of course, to fix limits for the proportions of each class; and it is therefore quite conceivable that designs will be evolved which it will be difficult to classify on the above basis. In such a case, one test that will help to decide to what class a particular engine belongs is based on the form of the separated chamber. In a turbulence-chamber engine, a section through the chamber which cuts the communicating passage in halves is at least approximately circular in form, whereas in a precombustion-chamber engine it is most likely to be rectangular.

A diagram of an engine cylinder with turbulence chamber is shown in Fig. 1. The chamber may be either spherical or cylindrical in form.

Advantages—In the turbulence-chamber engine the energy necessary to effect a rapid and thorough intermixture of the air and fuel charges derives from two sources—the kinetic energy of the fuel jet and that of the swirl of air in the turbulence chamber. If fuel is injected directly into the main combustion chamber, in which there is little or no turbulence, all of this energy must be supplied by way of the fuel jet, from which it is natural to conclude that in a turbulence-chamber engine it is possible to work with lower injection pressures than in an engine having direct injection into the main combustion chamber. This is one advantage of the turbulence-chamber type.

Another advantage resides in the fact that if the rotational velocity of the air swirl in proportion to the engine speed is well chosen, and the injection nozzle is properly located and aimed, the fuel charge will be so well distributed through the air charge that the proportion of excess air required to effect smokeless combustion need not exceed 25-35 per cent. This means a high bmep and low fuel consumption.

The turbulence chambers usually are provided with a hot surface opposite the injection nozzle, which in a way corresponds to the "burner" in the precombustion-chamber type. This hot surface helps to heat the air during the compression stroke, which promotes ignition and complete combustion, and one advantage often claimed for these engines is that they are free from the pungent exhaust odor characteristic of some Diesel engines, which, if present, is indicative of incomplete combustion.

Less Objectionable Exhaust—When the combustion in an engine burning petroleum products is complete, the exhaust contains only carbon dioxide, steam, nitrogen and oxygen,

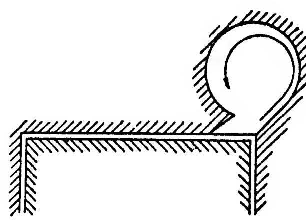


FIG 1—DIAGRAM OF TURBULENCE CHAMBER

all of which are odorless, and under conditions of complete combustion the exhausts from both gasoline and Diesel engines are practically odorless. It is well known, however, that under certain conditions the exhaust of each type of engine has a characteristic odor. This is not due, either, to carbon monoxide, which is always present in some proportion in the exhaust from gasoline engines, as this poisonous gas also is absolutely odorless. The odors are due to intermediate products of combustion.

In the combustion process, the first thing that happens is that the hydrocarbon molecule of the petroleum product is split up into carbon and hydrogen atoms, both of which eventually combine with the oxygen of the air to form carbon dioxide and steam. These transformations, however, are not effected directly; on the contrary, intermediate products are first formed, including peroxides, aldehydes, ketones and acids. If conditions are unfavorable, combustion is never completed, in which case these products appear in the exhaust, and they are the cause of the characteristic odor.

One reason for an interruption of the combustion process may be that the intermediate products of combustion come in contact with a cold wall, which chills them to such a degree that the final chemical reaction of the combustion process cannot take place, and since in the turbulence-chamber engine the surface with which the burning fuel jet is most likely to come in contact is at a high temperature, this cannot—or at least is not so likely to—occur there.

What is probably the greatest advantage of the turbulence-chamber engine is its flexibility, due to the relative independence of the ignition lag, in crankshaft degrees, of the speed (more rapid combustion at high speeds).

Disadvantages—The advantages discussed in the foregoing paragraphs are offset by some disadvantages. In the first place, the high rate of swirl of the air in the turbulence chamber, which is at least partly water-cooled, results in an increased rate of heat loss during the compression stroke, so that the temperature in the chamber at the end of the stroke is not as high as it would be in an engine with undivided compression chamber having the same compression ratio. This makes it somewhat more difficult to start the turbulence-chamber engine from cold, and starting aids, such as glow plugs, are usually required. There is greater loss of heat due to this cause also in normal operation, from which a loss in thermal efficiency might be expected, but if this loss does occur it is offset in part by greater combustion efficiency.

The objection also has been raised against the turbulence-chamber type that its combustion pressure tends to be very high and that the engine is rough in operation. The rapidity of combustion in the turbulence-chamber engine and, consequently, the rate of pressure rise, is dependent on what has been called the "swirl ratio," that is, the ratio of the angular velocity of the swirl to that of the engine crankshaft.

Measurement of Swirl Velocity—If it is granted that an air swirl has beneficial effects, the problem of the relation between the velocity of the swirl and the output, economy, and smoothness of operation of the engine arises. The swirl starts as soon as compression in the cylinder and flow into the turbulence chamber begin, but its velocity varies throughout the compression stroke, and this, together with the fact that it is influenced by the elasticity and viscosity of the air, makes it very difficult to calculate.

A number of attempts have been made, therefore, to determine this velocity experimentally. What it is particularly desirable to know, of course, is the velocity of the swirl during the injection period. Dr. Friedrich Sass of the German General Electric Co. developed a swirl indicator applicable more particularly to large, direct-injection engines. It comprises a vane located in the compression chamber close to the cylinder wall, mounted on a shaft which extends through a bushing in the cylinder head. At its outer end the shaft is fitted with an arm carrying a pencil adapted to record its motion on a revolving drum. Normally the indicator is held in the zero position by a spring, the vane then occupying a radial position in the cylinder. The air swirl exerts a pressure on the vane, deflects the spring, and causes the pencil to scribe a record on the drum. By rotating the drum around its axis by connection to the engine crankshaft, it is possible to obtain a diagram of the variation of the swirl velocity

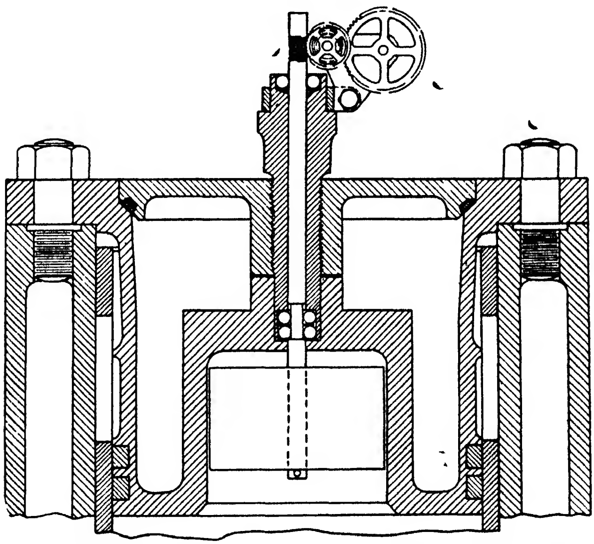


Fig. 2.—Ricardo Swirl Meter Applied to Engine with Vortex Combustion Chamber.

throughout the engine cycle. A weak feature of this instrument is that the presence of the vane in the combustion chamber is likely to change the air velocity therein materially.

Ricardo Swirl Meter—Harry Ricardo has devised a swirl meter which records the number of revolutions made by the air in a given time, and therefore gives the mean velocity of the swirl during the whole cycle. Like Sass' instrument, it is applicable only to engines with a particular form of combustion chamber. Ricardo has designed a number of engines with what he calls a Vortex combustion chamber (Fig.

2). In these engines the air is admitted to the cylinder through tangential ports, and is compressed during the following up-stroke into a cylindrical combustion chamber coaxial with the cylinder but of smaller diameter. The Ricardo swirl meter comprises a vane on a shaft carried in ball bearings in a bushing located centrally in the cylinder head, connected to a counting mechanism at its top. The velocity of swirl is measured while the engine is being "motored."

Engines with Ricardo Vortex combustion chambers are not turbulence-chamber engines as defined in the early part of this chapter, being akin to the direct-injection type shown in Fig. 5 of Chapter IX, as the fuel is injected parallel with the axis near the wall of the chamber, so that the spray cuts across the air swirls. However, the effects of air flow or air swirl in this engine have been thoroughly investigated, and as this investigation threw much light on these effects, some of its results will be given here.

Swirl Ratio—The term "swirl ratio" was introduced by Ricardo to signify the ratio of the number of revolutions per minute of the swirl in the turbulence chamber or combustion chamber, to the number of engine revolutions per minute. An investigation of the effect of changes in the swirl ratio on engine performance, made in the laboratory of Ricardo & Co., was dealt with in a paper presented to the Institution of Automobile Engineers by J. F. Alcock in 1935, from which the following information is taken.

The tests were carried out on a single-cylinder 5 by 7-in. engine with Vortex combustion chamber (Fig. 2) and with means at the inlet ports permitting of varying the swirl ratio in the cylinder from zero to 6.3, and in the combustion chamber from zero to a final value of 16.3. Measurements showed that the volumetric efficiency was not affected by the means employed to change the swirl velocity, hence the changes in performance were due entirely to effects of changes in the swirl ratio on the efficiency of combustion. The tests were carried out at 1300 rpm.

Fig. 3 shows the effect of the swirl ratio on the maximum bmep obtainable with the "exhaust just visible" (full line) and "moderately dirty" (dashed line), as well as the effect on the fuel consumption. It will be seen from these curves that a swirl ratio of 10-10.5 gives both maximum bmep and minimum fuel consumption. In these tests the beginning of injection was fixed, and Mr. Alcock pointed out that if the injection timing had been varied to give a constant peak pressure, which would have been more logical, the optimum swirl ratio would have been found to range between 9.5 and 10.

Effects of Swirl Ratio on Performance—Additional results from these tests are given in Fig. 4, which shows that the maximum pressure and the rate of pressure rise increase with the swirl ratio. This makes the engine rather rough in operation at high swirl ratios, for which reason it is generally advisable to operate at a swirl ratio about 10 per cent lower than that giving maximum engine performance. Not much is sacrificed thereby in the way of performance, and the engine operates noticeably smoother.

From the relationship of the two bmep curves in Fig. 3,

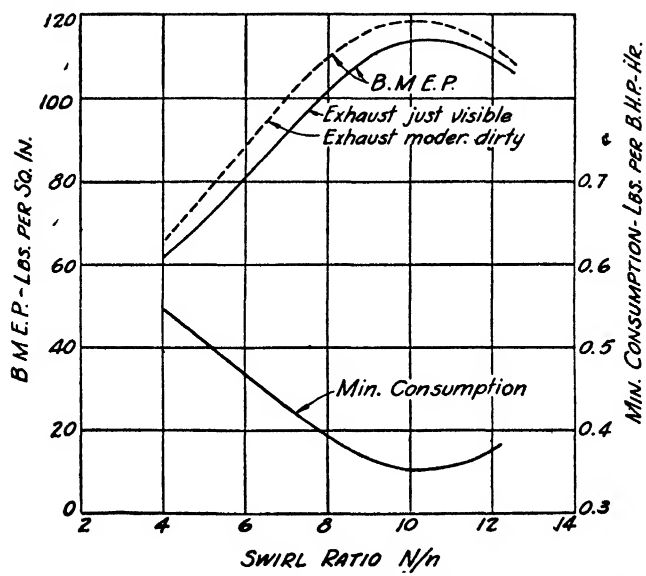


FIG. 3.—EFFECT OF SWIRL RATIO ON BMEP AND FUEL CONSUMPTION.

both peaking at substantially the same swirl ratio and dropping symmetrically from the peak values, the conclusion was drawn that the optimum swirl ratio is substantially the same for all loads. In the tests on this engine, the speed was varied between rather narrow limits only, 900 and 1300 rpm, but it was found that one swirl ratio gave best results throughout this range. That the optimum swirl ratio is independent of the speed was confirmed on other engines with Vortex-type combustion chamber, in which the speed was varied through a wider range.

It was further found that as the swirl ratio increases, the heat loss to the jacket increases and, as a consequence, the compression pressure drops, which accounts for greater difficulty in starting with a high swirl ratio. What is referred to as "the heel of the load-range curve," that is, the full-load end of the curve of specific fuel consumption vs. bmep, rises more abruptly if the swirl ratio is decreased below the optimum value, and the exhaust begins to show smoke at a

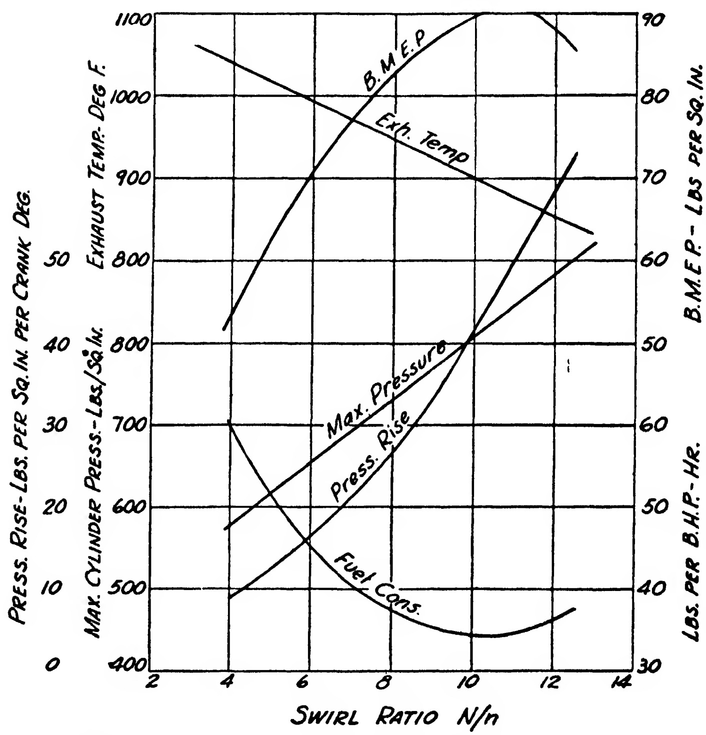


FIG. 4.—EFFECT OF SWIRL RATIO ON VARIOUS ENGINE CHARACTERISTICS. The scale for fuel consumption is in hundredths of a pound.

lower bmep. Tests on engines with a wide range of cylinder sizes showed that cylinder size has no effect on the optimum swirl ratio.

Mr. Alcock pointed out in this connection that the swirl velocity obtained with the Ricardo swirl meter is much lower than that obtained from calculations (evidently based on the tangential component of the air velocity through the inlet ports and the circumference of the combustion chamber). In one particular case it was only one-third as large. The difference is thought to be due to wall friction. In a turbulence chamber like that of the Ricardo Comet engine, the friction is not likely to be nearly as large, and it is believed that the actual swirl ratio in such an engine ranges between 20 and 25.

Size and Form of Turbulence Chamber—Extensive tests with turbulence chambers were made by the N.A.C.A. on a single-cylinder 5 by 7-in. experimental engine which was fitted successively with cylinder heads having turbulence chambers of different form and size, and different connecting passages (N.A.C.A. Technical Note No. 514, by C. S. Moore and J. H. Collins, Jr.). In the earlier tests the chambers were of

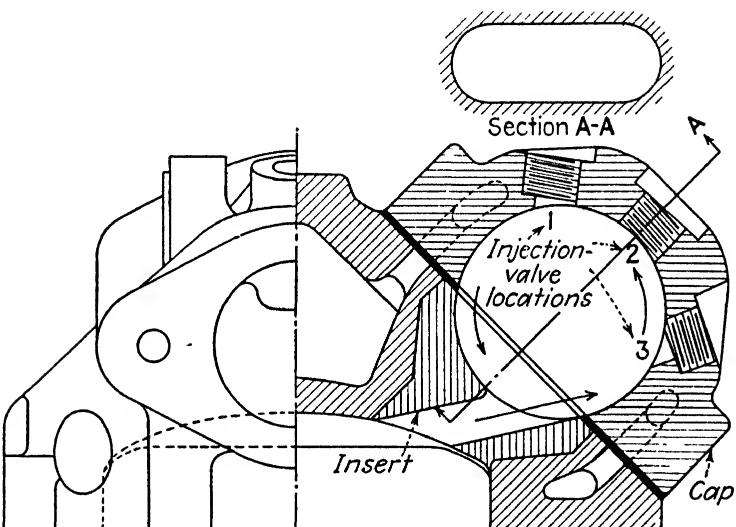


FIG. 5.—N.A.C.A. TURBULENCE-TYPE COMBUSTION CHAMBER.

spherical form and had a radial connecting passage, and the compression ratio used was 13.5. It was found that the output and economy increased as the volume of the turbulence chamber was increased from 20 to 70 per cent of the total clearance. On the other hand, the smoothness of the engine decreased over the greater part of the range, the engine running smoothest with a turbulence-chamber volume equal to 35 per cent of the total clearance. The total heat loss to the jacket increased from 20.5 to 28.5 per cent with increase in the turbulence-chamber volume through the range given.

Using a spherical chamber of 50 per cent of the total clearance volume, the best size of the connecting passage, for a speed of 1500 rpm, was found to be that of a circle of between 0.45 and 0.53 in. diameter. A decrease in the size of the passage, which naturally increased the velocity of the swirl, cut down the ignition lag and speeded up the combustion.

A flat-cylindrical turbulence chamber of 50 per cent the total clearance volume and with tangential connecting passage (Fig. 5) gave better performance than a spherical chamber of the same volume and radial connecting passage. Starting, idling, and combustion shock were affected little by the form of the turbulence chamber. Rounding the cylinder end of the connecting passage improved the performance, while

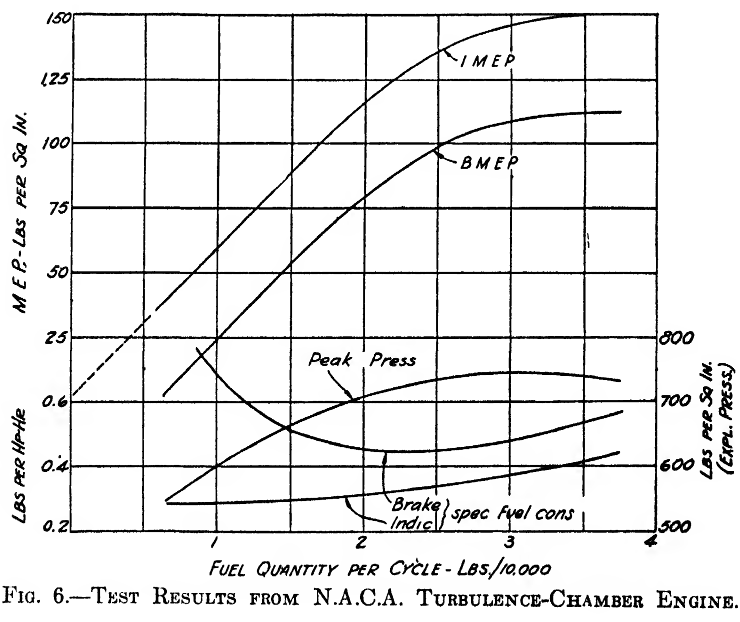


FIG. 6.—TEST RESULTS FROM N.A.C.A. TURBULENCE-CHAMBER ENGINE.

rounding the chamber end impaired it slightly, evidently because it interfered with the air swirl in the chamber. Best performance was obtained when the connecting passage entered the cylinder radially. As regards injection nozzles, results were best with a single-orifice nozzle injecting the fuel substantially at right angles to the direction of air flow.

Fig. 6 shows performance curves obtained with a flat-cylindrical turbulence chamber of 50 per cent the total clearance volume, with a tangential connecting passage flared at the cylinder end, and a single 0.050-in. spray from the middle one of the three tapped holes for the nozzle. This is believed to represent substantially the best performance obtainable with this type of turbulence chamber, though a slight improvement might result from an increase in the chamber volume.

An increase in the compression ratio of this engine, from 13.5 to 17.5 (made by decreasing the cylinder clearance, so that the percentage of turbulence-chamber volume increased with the compression ratio), resulted in an increase in the imep of only 4 psi, and as the friction also increased with the compression, the full-load brake output remained substantially constant. It was concluded that for this type of cylinder and head the optimum compression ratio is 15.5.

Boosting tests made on this engine showed that the bmep increased 2.5 psi for every inch of mercury column boost below 10 in. At part load the specific fuel consumption (on the brake-horse-power basis) was greater when boosting, because of the power consumed by the blower.

Ricardo Pintaux Nozzle—Harry Ricardo, who originated the turbulence-type of combustion chamber, also devised a special injection nozzle to facilitate the starting of engines equipped with it. It is known as the Pintaux nozzle and is being manufactured by C.A.V. Ltd.

It was found that with engines having the Ricardo "Comet" type of combustion head, where the injection nozzle is located opposite the entrance to the swirl chamber, the all-round performance is best if the fuel is injected tangentially in the direction of the air swirl. But while this results in maximum power, it makes starting of the engine difficult, for one reason because, since both the fuel and the air travel in the same direction, the spray jet is not broken up so energetically, and none of the fuel reaches the center of the spherical chamber, where the temperature probably is the highest. Before the introduction of the Pintaux nozzle it was common practice to employ starting aids with turbulence-type engines, either glow plugs in the swirl chambers or an electric heating element in the inlet pipe.

As shown in Fig. 7, the Pintaux nozzle is of the pintle type, and a small, inclined hole is drilled in the nozzle body immediately below the valve seat. The pintle is a fairly close fit in its hole. At starting and when idling, the pressure developed by the injection pump is sufficient to lift the nozzle valve off its seat, but not sufficient to lift it so far that the pintle is entirely out of its hole. Under these conditions most of the fuel is forced through the auxiliary injection orifice directly below the valve (Fig. 8, left) and meets the air as it enters

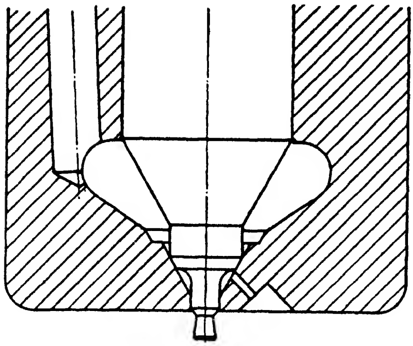


FIG. 7.—RICARDO PINTAUX NOZZLE.

the turbulence chamber, where it certainly is hotter than after it has nearly completed the round of the chamber. As soon as the engine gains speed, the pintle is lifted out of its hole at each injection, and the bulk of the fuel (about 90 per cent) is then injected in the direction of the nozzle axis (Fig. 8, right). It is claimed that the Pintaux nozzle lowers the minimum starting temperature of the "Comet" engine without starting aids by about 25 F.

Hercules Engines—An extensive line of two-, four-, six-, and eight-cylinder turbulence-type engines is being manufactured by Hercules Motors Corporation of Canton, O. A side-sectional view of one of these engines, the Model DJX, is

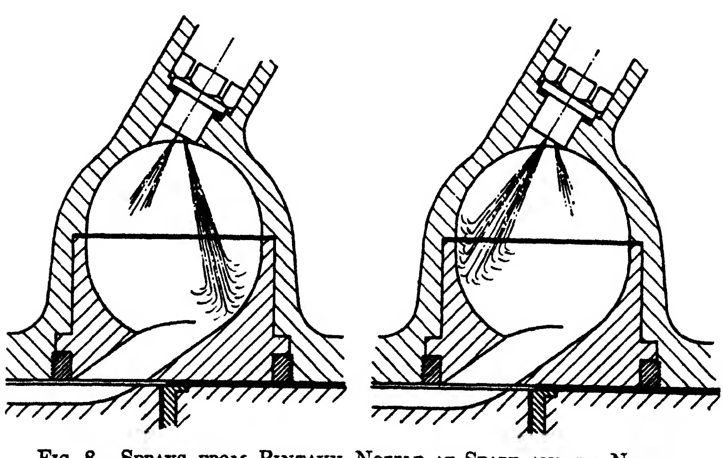


FIG. 8.—SPRAYS FROM PINTAUX NOZZLE AT START AND IN NORMAL OPERATION.

shown in Fig. 9, and a cross section in Fig. 10. Hercules engines range in displacement from 113.7 to 1468 cu in. and in maximum horse power from 12 to 500. In these engines there is a spherical turbulence chamber at the side of the cylinder, as clearly shown in the cross section. This chamber is formed in a pocket cast on the engine block and is provided with a lining held in place between the block and the cylinder head. The turbulence chamber has a narrow tangential inlet, which causes the air to swirl therein. Toward the end of the compression stroke, when injection takes place, the velocity of the swirl is increased, because the air then enters through a narrower passage, part of the inlet being masked by the piston. That, naturally, increases the velocity of inflow.

The engines are of very rugged design. Except in the smallest model, which is designed for hand-cranking (although an electric starter is supplied on demand), the compression ratio is 14.8 to 1. A somewhat higher ratio, 15.5 to 1, is used in the small engine to facilitate starting. "Dry" liners are used in most models.

Inlet and exhaust valves are located in the cylinder head and operated through pushrods and tappet levers from the camshaft in the crankcase. Throughout the line the exhaust valves have smaller diameters than the inlet valves. Thus the inlet valves of the $4\frac{5}{8} \times 5\frac{1}{4}$-in. engine have a port diameter of 2 in., while the exhaust ports are of $1\frac{3}{8}$-in. diameter. Crankshafts are of very rigid design, that of the six-cylinder, $4\frac{5}{8} \times 5\frac{1}{4}$-in. engine having main bearings of $3\frac{1}{2}$-in. diameter. Crankshafts are drop forgings and are surface-hardened by the Tocco process.

Fuel-injection pumps and the pintle-type nozzles are of standard make. The nozzles are clamped to the sides of the turbulence-chamber pockets and mounted in sleeves passing through the water jackets. Lubrication of all bearings, including those of the floating piston pins, is by the pressure system. Injection pressures (nozzle-opening pressures) vary from 1650 psi in the smallest to 2000 psi in the largest sizes. The shipping weight of the six-cylinder, $4\frac{5}{8} \times 5\frac{1}{4}$-in. engine is 1600 lb; its maximum rated horse power, 150 at 2200 rpm.

Waukesha Engines—The Waukesha Motor Company, Waukesha, Wis., manufactures a complete line of Ricardo-type turbulence-chamber engines, piston displacements ranging from 129 to 2894 cu in. and rated horse powers from 35 to 385. A cross section of the Model 148-DK is shown in Fig. 11. This is a six-cylinder engine of $5\frac{1}{4}$-in. bore by 6-in.

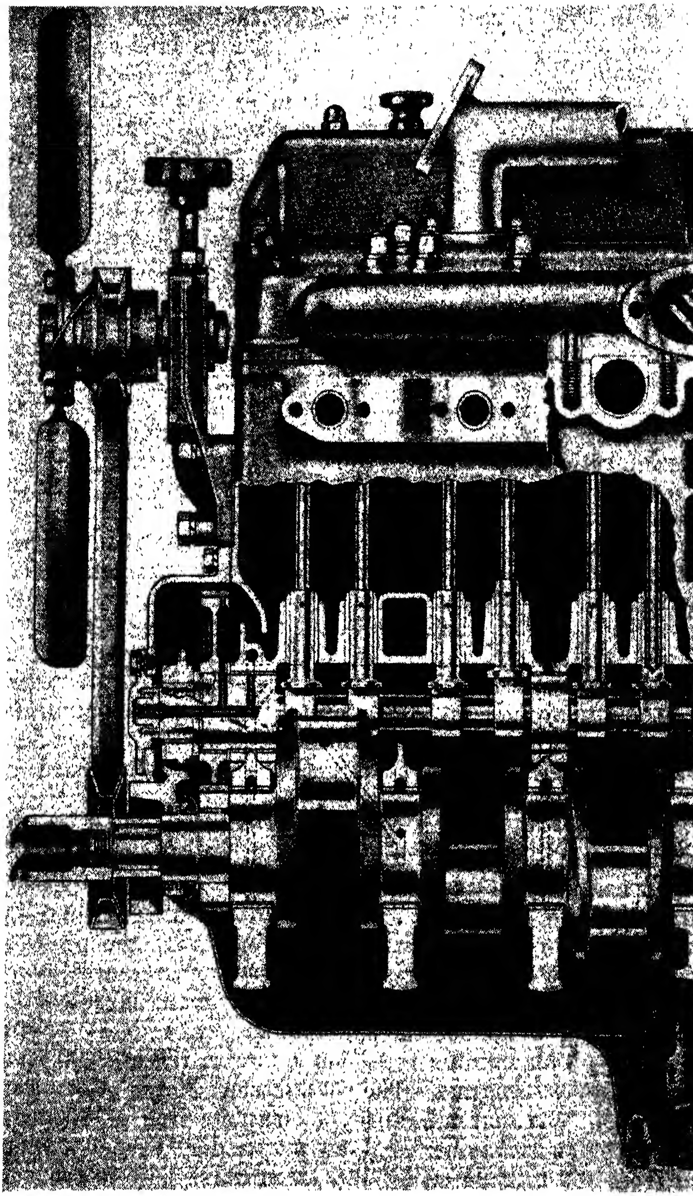


FIG. 9A.—HERCULES ENGINE, FORWARD END.

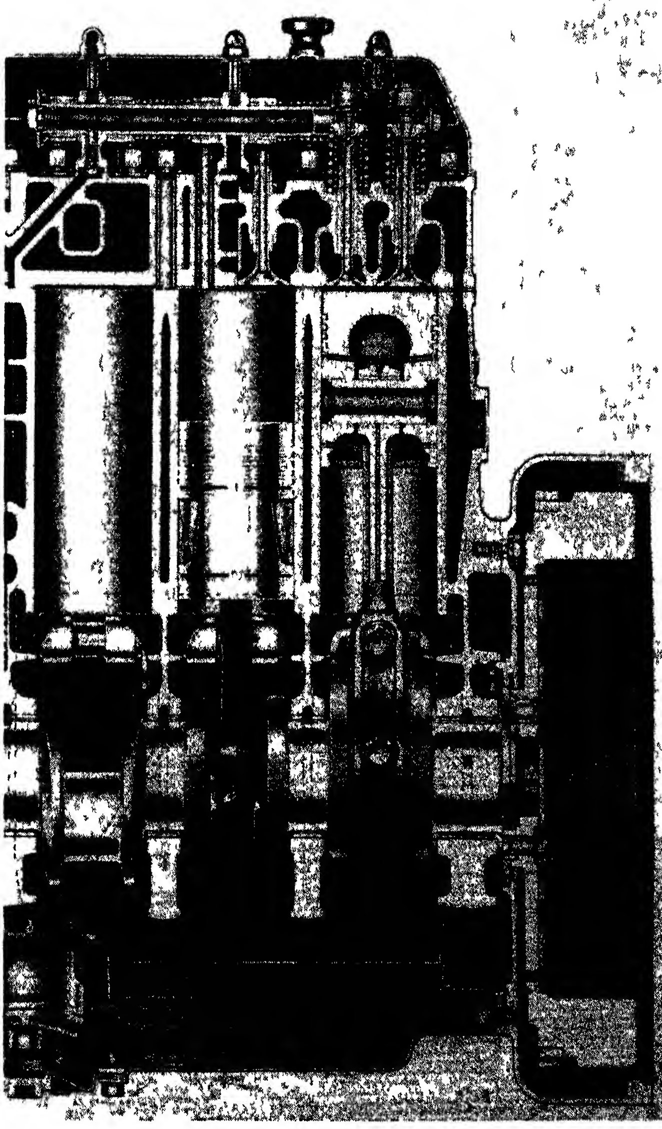


FIG 9B.—HERCULES ENGINE, REAR END.

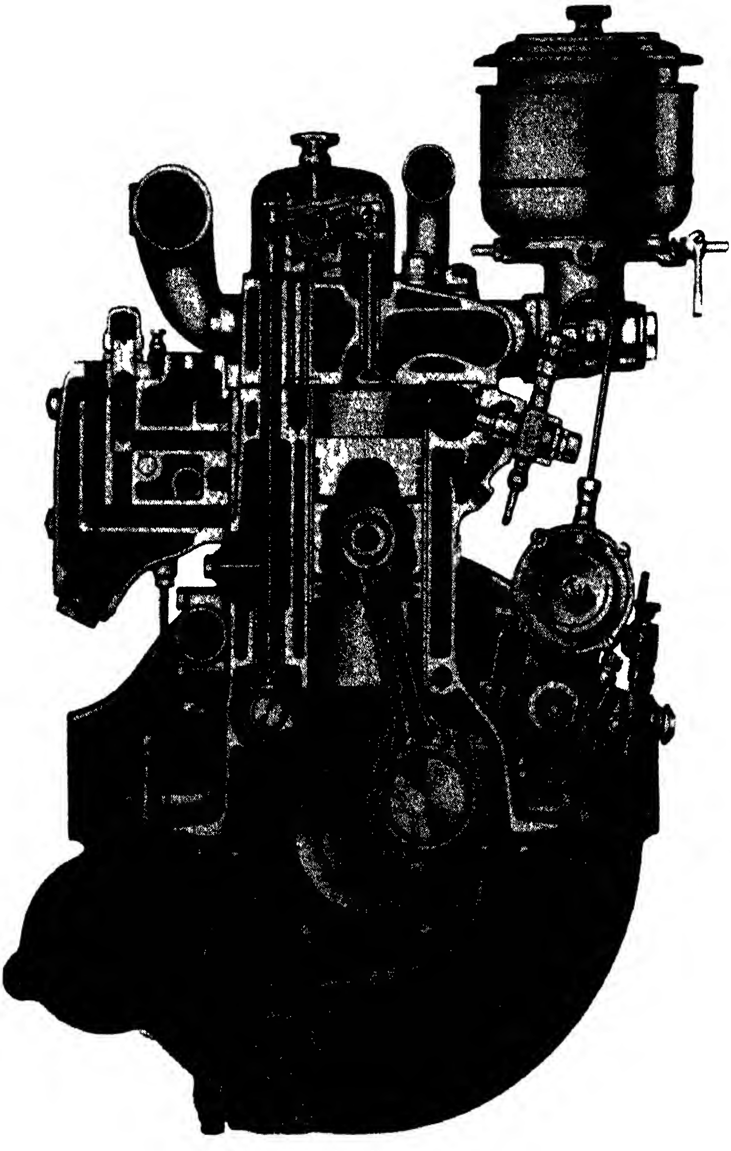


FIG 10—CROSS SECTION OF HERCULES ENGINE

stroke and is rated 200 hp at 2100 rpm (97 psi bmep). The weight of the engine is approximately 2320 lb.

The turbulence chamber, which is removable from the outside, consists of a lower cup made of a heat-resistant alloy, and a water-cooled cap which receives the injector. There is an air space between the cup and the metal of the cylinder head surrounding it. Approximately one-half of the air charge passes into the chamber during the compression stroke and absorbs heat from the cup. The temperature of the cup increases with engine speed, and the resulting increase in the air temperature reduces the ignition delay at high speeds and prevents excessive combustion pressures.

From the cross section it can be seen that the cylinder block and crankcase are a single casting into which "wet" cylinder sleeves are inserted. The sleeves are flanged at the top, and the block is counterbored to such a depth that when the sleeves bottom in the counterbores a "fire land" on top of them projects a few thousandths of an inch above the top surface of the block. This projection provides a dependable sealing action at the sleeve flange, preventing leakage and retaining the sleeve firmly in the block.

Injection equipment is of American Bosch make. The injectors fit into the turbulence-chamber caps, and adjacent to each injector there is provision for an electric glow plug, which serves as a starting aid. Pistons are of aluminum alloy. Each of them carries three keystone-type compression rings (of which the topmost is chromium-plated), one keystone-type oil-scraper ring above the piston pin, and two straight-sided oil-control rings, one above and one below the pin. Piston crowns are formed with ∞-shaped depressions in them. When ignition occurs in the turbulence chamber, the blow-off from that chamber enters the depression through a radial channel, is split in two, and forms two distinct, oppositely directed swirls in the combustion chamber.

The pistons are oil-cooled. A nozzle secured into the top of the connecting rod throws a stream of oil from the lubricating system against the piston crown. Most of this is caught in an oil chamber formed in the top of the piston by an integral inner flange and a flanged ring secured thereto, the latter acting as a dam. The oil pump, which is of sufficient size to meet both the lubricating and cooling requirements, is mounted on the outside of the crankcase and is shown at the right in the sectional view. An oil cooler, shown to the left of the cylinder, is standard equipment on this model.

The crankshaft is carried in seven main bearings of $4\frac{1}{4}$-in. diameter. Bolted-on-type counterweights are provided for

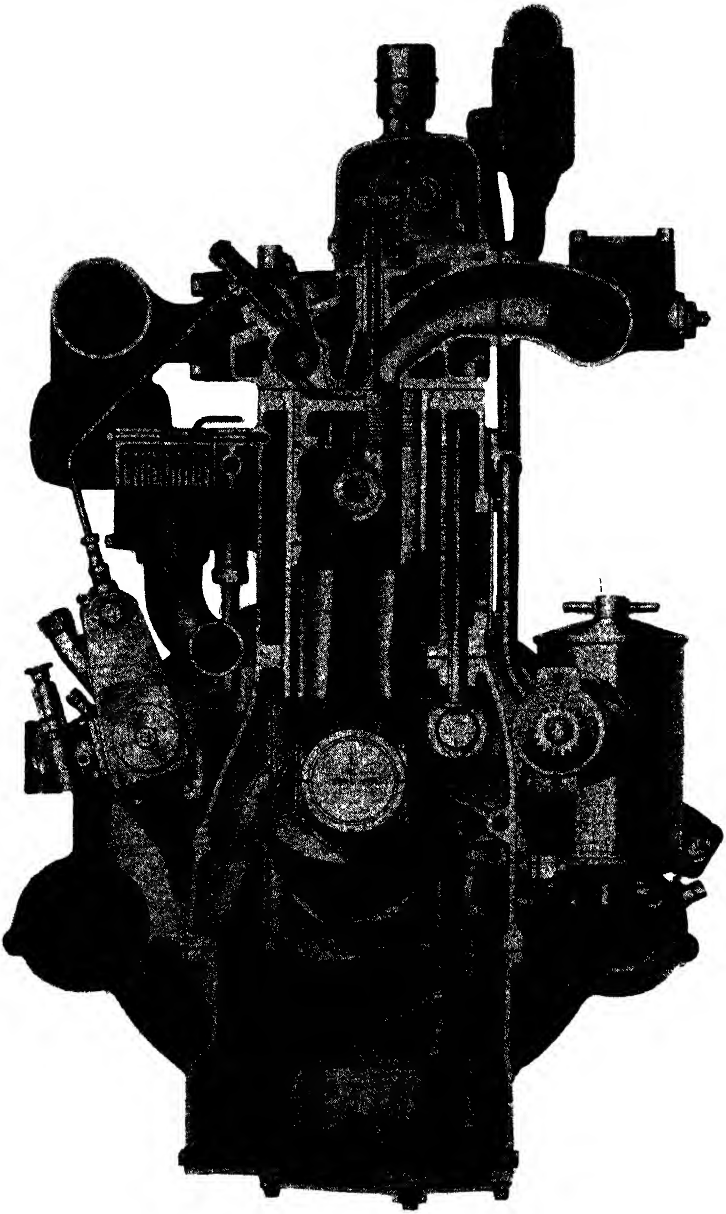


Fig. 11.—Cross Section of Waukesha Engine.

high-speed applications. Main journals and crankpins are surface-hardened. Caps of the connecting rods are secured to the rods by four heat-treated bolts each. The high-speed model usually is furnished with electric starting equipment, comprising a 12-volt generator, a 24-volt starting motor, glow plugs, electric intake heaters, and series-parallel controls. Batteries used with this equipment are connected in parallel for charging and in series for starting. For the moderate-speed industrial model a four-cylinder gasoline starting engine is available. With this the inlet manifold of the Diesel is heated by the exhaust from the gasoline engine before starting. This installation includes an automatic pinion shift

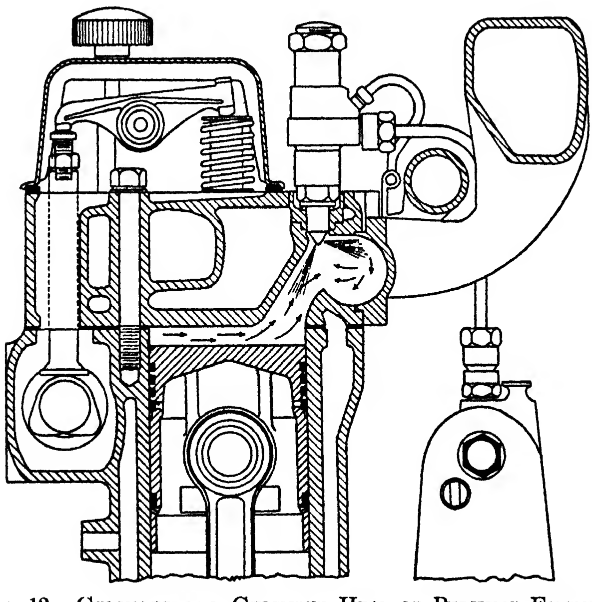


Fig. 12.—Cylinder and Cylinder Head of Perkins Engine.

and a manually operated clutch. No glow plugs and electric heaters are used with this starter.

Perkins Engine—Fig. 12 is a partial cross section and Figs. 13A and 13B are a longitudinal section of an engine with the Ricardo "Comet" head, manufactured by F. Perkins, Ltd., of Peterborough, England.

The six-cylinder engine has a bore of 3.5 and a stroke of 5 in. (288.5 cu in. displacement), and is rated 85 hp at 2600 rpm, the governed speed. This corresponds to a bmep of 90 psi. The maximum torque of 193 lb-ft is developed at

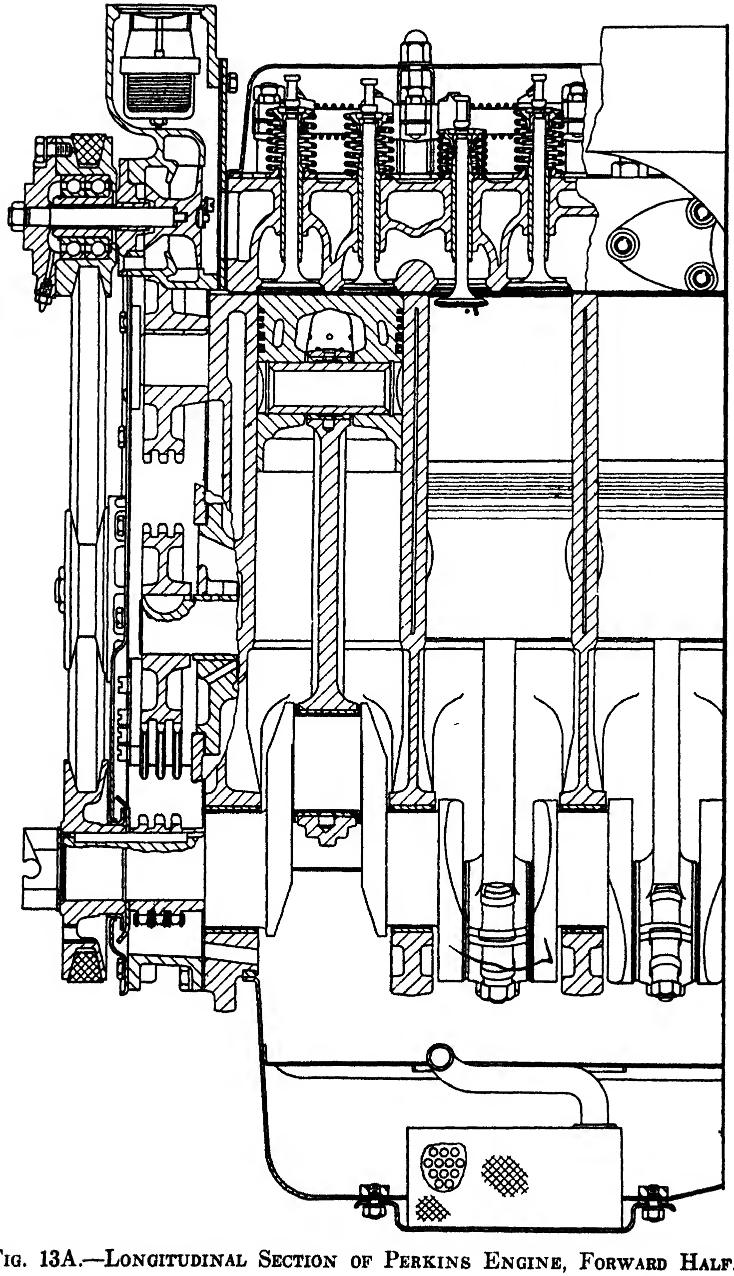


FIG. 13A.—LONGITUDINAL SECTION OF PERKINS ENGINE, FORWARD HALF.

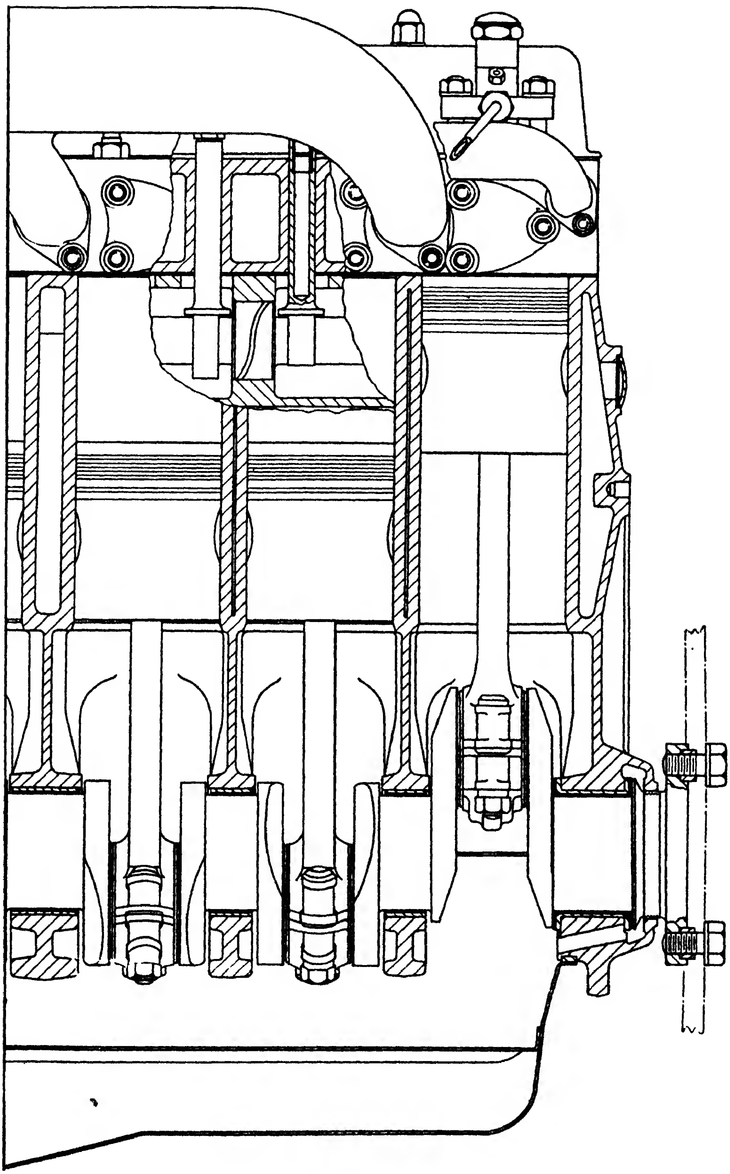


FIG. 13B.—LONGITUDINAL SECTION OF PERKINS ENGINE, REAR HALF.

1500 rpm. Complete with flywheel, bell housing, starting motor and fuel pump the engine weighs 712 lb. Its compression ratio is 15.8. The cylinder block and crankcase top-half are cast of chrome-alloy cast iron, with full-length water jackets. Cylinder walls are made sufficiently heavy to permit of fitting dry liners when reconditioning becomes necessary. The crankshaft is a drop forging of nickel-chromium-molybdenum steel and is electrically hardened. The oil sump, valve cover and valve-gear cover plate are steel pressings.

There are seven bearings on the crankshaft, $2\frac{3}{4}$ in. in diameter, while crankpins are of $2\frac{1}{4}$-in. diameter. Those halves of both sets of bearings which take the load due to the gas pressure are lined with copper-lead, while the remaining halves are lined with babbitt. The camshaft, which is located in a trough extending lengthwise of the cylinder block near its top, is driven through a triple-strand roller chain, and it actuates the valves through short tappet rods and rocker levers. Inlet and exhaust-valve ports are of $1\frac{13}{32}$ and $1\frac{3}{16}$-in. diameter, respectively. The engine is governed by means of a pneumatic governor.

An unusual system of crankcase ventilation is employed. Pipe connections are made at opposite sides of the throttle in the inlet passage, at points between which there is normally a pressure difference of $\frac{1}{2}$-in. water column. This pressure difference is made use of to force a considerable proportion of the incoming (filtered) air through the crankcase and the valve chamber.

CHAPTER XI

Auxiliary-Chamber Engines

An auxiliary-chamber engine may be defined as one in which part of the compression space is in a chamber separated from the space directly above the piston by a throat or restricted channel, and in which the fuel nozzle is located outside of this chamber. This distinguishes it from the precombustion-chamber and turbulence-chamber types, in which also part of the compression space is separated from the space above the piston, but the nozzle is located in the separated chamber, so that all of the fuel injected must pass through the latter. Sponsors of some of the engines coming under the above definition have referred to them as air-chamber engines, while others call theirs energy-cell engines.

In the original engine of this type the auxiliary chamber was comparatively large, comprising approximately 70 per cent of the total compression volume. It was known as an air-chamber engine, and combustion was supposed to take place in the throat separating the chamber from the cylinder, where the fuel spray was believed to meet air rushing out of the chamber during the early part of the expansion stroke. If this theory is correct, the outflow from the chamber is due solely to the piston motion, which creates a pressure difference between the chamber and the cylinder.

Energy-Cell Engine—In the energy-cell engine the separated chamber is much smaller, comprising from 10 to 18 per cent of the total clearance space. The fuel nozzle is directed at the inlet to the chamber, and since injection begins before the compression stroke is completed, a certain amount of fuel is carried along with the air flowing into the energy cell and apparently forms a good combustible mixture therein. Ignition starts in the space between the tip of the injection nozzle and the inlet to the energy cell, but is communicated to the cell, and combustion produces in the cell a pressure much higher than that in the combustion chamber, with the result that gases of combustion are blown from the cell with great force. The energy of this gaseous jet is taken advantage of to create a swirling motion in the combustion chamber.

Returning to the air-chamber type, it is rather obvious that if a large part of the total air charge is in the air chamber at the beginning of the expansion stroke, and enlargement of the combustion chamber by the downward movement of the piston is depended upon to draw that air into the throat, where the combustion is supposed to take place, much of the fuel will be burned rather late in the power stroke. Under these conditions it would be impossible to efficiently convert the heat energy of the fuel into mechanical energy, and that may explain why air-chamber engines are no longer being produced, at least not on a large scale. These engines, how-

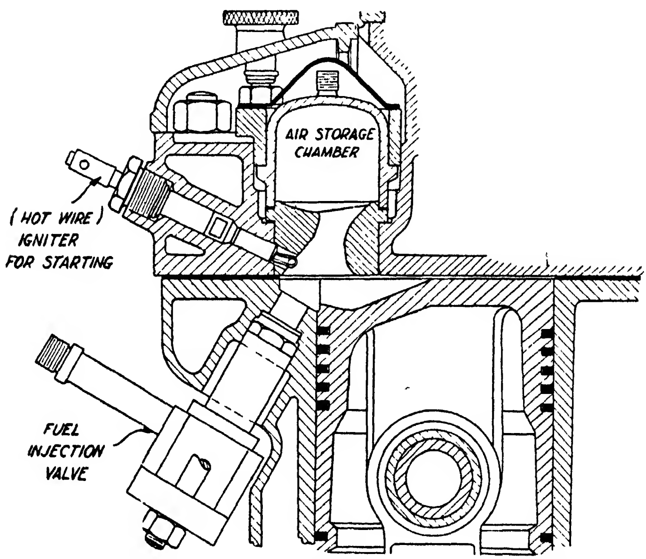


Fig. 1.—Combustion Chamber of Acro-Type Saurer Engine.

ever, played a rather prominent role in the early history of the high-speed Diesel, and a brief account of their development may be of interest.

Acro Engine—The first representative of the air-chamber type was the Acro engine, which was invented by Franz Lang of Munich, who for many years was connected with the Maschinenfabrik Augsburg-Nürnberg, the original manufacturer of Diesel engines. Lang assigned his American patents to the American Crude Oil Motors Corporation, organized in 1923, and the word "Acro" was formed from the initials of the corporation to serve as a trade name. Some time later the

Acro Motor Company of Küssnacht, Switzerland, was formed and took over all European rights under the Lang patents. In 1925 this company came under the control of the Robert Bosch Company of Stuttgart, Germany.

The Acro engine had the auxiliary chamber either in the cylinder head or in the piston. Fig. 1 shows the combustion chamber of an Acro engine with the auxiliary chamber in the cylinder head. The compression space is divided into three parts, one being in the auxiliary chamber, the second in the venturi-shaped throat connecting it with the cylinder, and the third in the cylinder. This latter part is constituted principally by the shallow space between the cylinder head and the piston when the latter is at the end of its up-stroke. The injection nozzle is so located that it sprays directly into the venturi throat.

Air stored in the auxiliary chamber was supposed to start flowing from that chamber into the cylinder at the beginning of the expansion stroke, at a time when fuel was being sprayed into the throat in the opposite direction, and from this counterflow a thorough intermingling of fuel and air was expected. However, in a high-speed engine, in order to ensure efficient combustion, all or nearly all of the fuel must be injected before the beginning of the down-stroke, or before the outflow from the chamber can begin, and the conditions for fuel distribution in the air charge therefore are not as favorable as would seem to be the case at first sight.

The Robert Bosch Company, as manufacturer of ignition equipment for gasoline engines and injection equipment for Diesels, had close connections with engine manufacturers throughout Europe, and quite a number of British and Continental firms undertook the manufacture of engines under the Acro patents, some of them placing the air chamber on the cylinder head, others in the engine piston. However, all of these licensees later abandoned the system.

M.A.N. Air-Chamber Engine—A rather different type of air-chamber combustion head was developed by the Maschinenfabrik Augsburg-Nürnberg (M.A.N.) and is illustrated in Fig. 2. The piston is slightly dished, and its outer portion comes close to the cylinder head at the end of the compression stroke. The major portion of the combustion chamber is in the cylinder head; it is of substantially conical form and the injection valve is located at the outer, small end, hence the spray cone conforms substantially to the form of the combustion chamber. One reason for locating the combustion chamber toward one side of the cylinder is that it permits of the use of larger valves, as required for high-speed operation. It will be seen that the auxiliary chamber is located alongside the combustion chamber and discharges through three orifices, the air escaping from these orifices mingling with the rich mixture formed in the combustion chamber. It is claimed that the substantially horizontal discharge of the auxiliary chamber creates a certain amount of turbulence in the combustion chamber, which assures clean combustion up to maximum load.

Air-chamber type truck engines were successfully produced by M.A.N. over a considerable span of years, and this combustion system was adopted for the German standard Army truck in the thirties and used throughout World War II. However, after the war the firm switched to the direct-injection type with spherical combustion chamber in the piston, described and illustrated in Chapter IX, on account of its higher fuel economy. This was the second change in combustion systems by M.A.N., which began the manufacture of truck-type Diesel engines in 1923 and originally produced an engine with direct injection into a combustion chamber of flat-cylindrical type, through two horizontal, oppositely located open injectors.

Lanova Energy-Cell Engine—Franz Lang, who originated the Acro combustion head for Diesel engines, some years later took out patents on another design, to which he applied the name Lanova. The original development work was done in Switzerland, and licenses under Lanova patents were issued in different countries. In this country further development of the combustion system and its commercial exploitation were undertaken by Lanova Corporation of Long Island City, N. Y. Diesel engines under Lanova license are being manufactured by Buda, Continental, Mack, Waukesha, and Fairbanks-Morse. Kohler, Chrysler (Dodge), Electric Boat, and Nordberg also are licensees, but are inactive at the time of this writing.

Vertical and horizontal sections of the Lanova double-lobe combustion head are shown in Fig. 3. The greater part of the compression space is in the cylinder head and is open to the cylinder. In horizontal section it has the approximate shape of a figure 8. Inlet and exhaust valves are located directly above this chamber. To one side of the main combustion chamber there is a small auxiliary chamber or energy cell, and directly opposite the latter is located the injection nozzle. The energy cell is divided into two compartments by a throat, the compartment nearest the inlet being the smaller one.

Action of Energy Cell—The spray from the nozzle passes through the main combustion chamber, and some of the fuel enters the energy cell with the air flowing into it during the latter part of the compression stroke. Ignition takes place somewhere between the nozzle tip and the inlet to the energy cell, because there the air is at the greatest distance from cooling walls and therefore reaches the highest temperature. But combustion is quickly communicated to the energy cell. In the design of the combustion chamber and the injection system the aim is to so arrange matters that the contents of

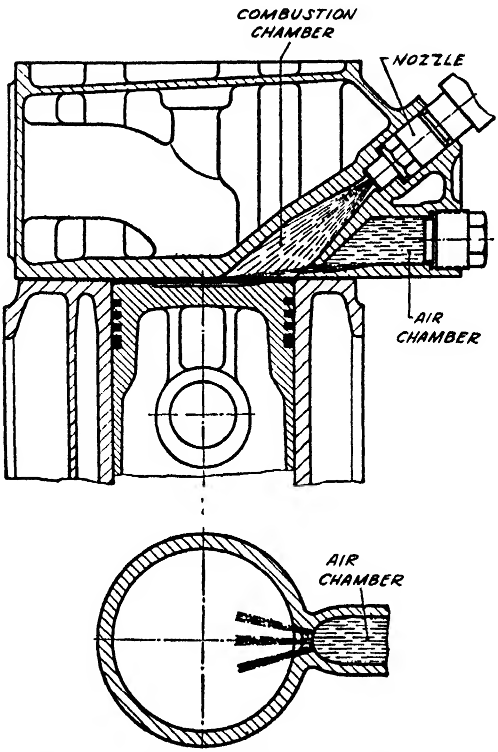


FIG. 2.—M.A.N. AUXILIARY-CHAMBER COMBUSTION HEAD.

the energy cell at the moment of ignition will be a combustible mixture of the proper proportion for complete combustion. When such a mixture, at from 14 to 15 times atmospheric density, is ignited, a very high pressure is generated, and although the charge in the combustion chamber is ignited in advance of that in the energy cell, the pressure rise is less rapid in the former, on account of the greater local richness of the charge there. Consequently, an excess pressure is produced in the energy cell, and this causes the products of combustion to be discharged with great force from the energy cell into the combustion chamber. The gaseous jet issuing from the energy cell is split and deflected by the "nose" of the nozzle tip, and this results in energetic swirls in the lobes of the combustion chamber. These air swirls, of course, are helpful in distributing the remainder of the fuel charge through the air in the combustion chamber.

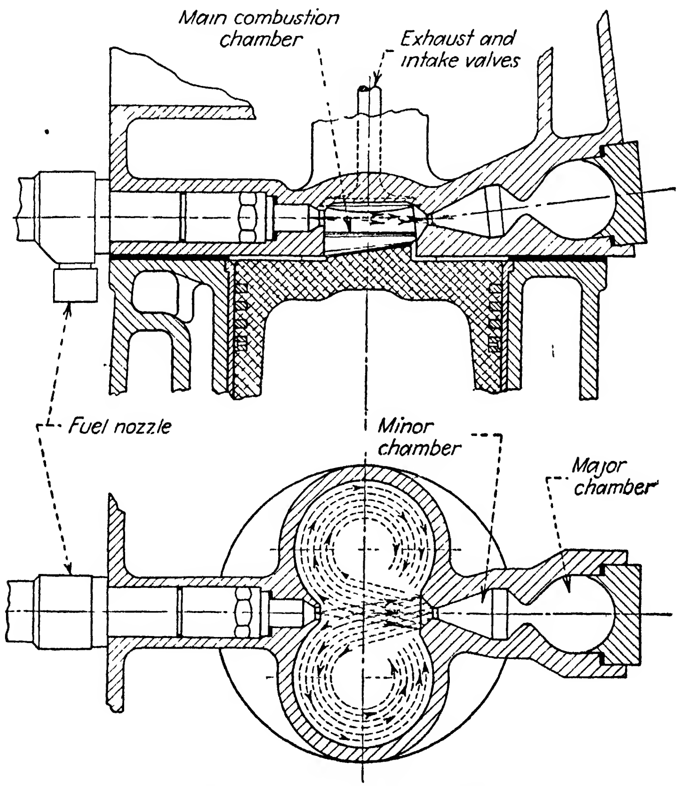


Fig. 3.—Lanova Energy-Cell, Dual-Lobe Combustion Head.

Two-Compartment Design of Energy Cell—It can be seen from the drawing that the energy cell is divided into two compartments, which communicate with each other through a small passage or throat. A cell of somewhat similar shape was used in the original Lanova engine, in which the outer compartment could be shut off by means of a valve, for the purpose of increasing the compression ratio for starting. When this starting aid was later abandoned and a single-compartment cell was tried, it was found that the inlet or throat of the cell corroded rapidly. With a single-compartment cell, the mixture in the cell at the time of ignition evidently is much leaner than the mixture in the compartment nearest the inlet of the double-compartment cell, and it is understandable that with the double-compartment energy cell the blow-off gases may be neutral and even reducing, whereas

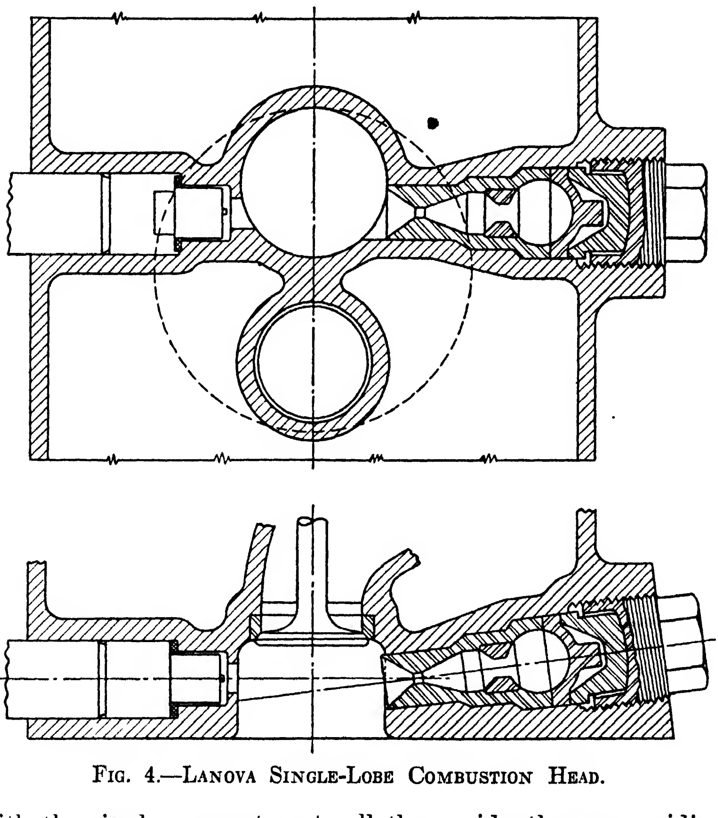


FIG. 4.—LANOVA SINGLE-LOBE COMBUSTION HEAD.

with the single-compartment cell they evidently were oxidizing. It also appears reasonable to suppose that with the double-compartment cell the blow-off occurs earlier and produces air swirl in the combustion chamber at a time when it is of more benefit.

Single-Lobe Combustion Chamber—After the double-lobe construction described in the foregoing had been in use for some years, a simpler design, known as the single-lobe chamber, was developed. Engines employing the single-lobe combustion chamber are cheaper to produce and are said to have a somewhat higher thermal efficiency, and Lanova licensees are gradually turning to the single-lobe construction. This type of combustion head is illustrated in Figs. 4 and 5. With this design there is less machine work on the cylinder head, and, besides, there is no need for the small displacers on the piston crown, which with the double-lobe type are needed to reduce the combustion chamber to the required volume. In most

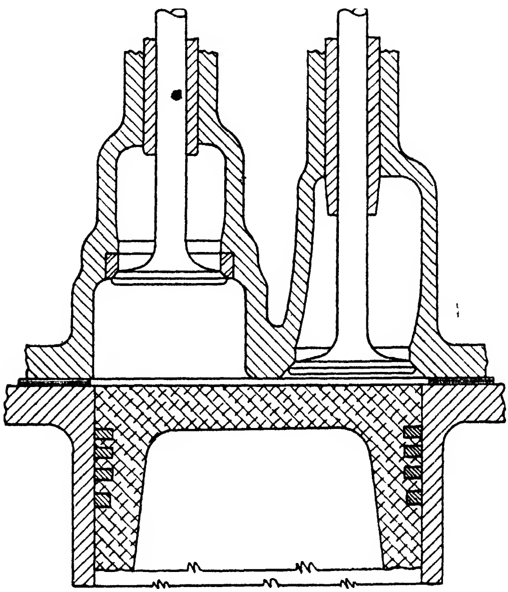


FIG. 5.—ANOTHER VIEW OF THE LANOVA SINGLE-LOBE COMBUSTION HEAD.

Lanova engines the exhaust valve is located at the top of the cylindrical combustion chamber, while the inlet valve is nearly flush with the bottom of the head. The reason for placing the exhaust valve in the combustion chamber, rather than the inlet valve, is that it is of smaller diameter and gives a more favorable form to the chamber. In the chamber shown in the drawings the energy cell and the nozzle are in the same vertical plane through the cylinder head, but the cell may be offset slightly toward the center of the cylinder. The energy cell is so located that it discharges tangentially into the combustion chamber, while the injection nozzle is aimed at the discharge orifice of the cell. With this design, too, combustion starts in the main chamber and is communicated to the energy cell, and the resulting blow-off sets up a swirling motion, breaks up the spray, and carries fuel to all parts of the chamber.

Advantages of the Lanova Combustion System—Among the advantages claimed for the Lanova system are the following: Owing to the manner in which combustion is initiated, the rate of pressure rise and the maximum combustion pressure are relatively low, and the engines therefore are not harsh in operation. Maximum pressures usually are below 850 psi, though they may rise to 900 psi in small engines at the speed of maximum torque. The strong turbulence created by the blow-off from the energy cell ensures efficient combustion, and the fuel consumption is relatively low. Curves said to represent average performance of Lanova medium-size truck engines at up to 2500 rpm show a minimum specific fuel consumption of 0.42 lb per bhp-hr, and a consumption of about 0.48 lb at full load; this together with a bmep dropping from a maximum of 108 psi to 90 psi at full load.

Self-cleaning pintle nozzles are generally used in Lanova engines, and are set to open at between 1700 and 2000 psi.

Tests of Energy-Cell Engine—When the Lanova combustion process was first made available to the Diesel-engine industry, comprehensive tests were made on a six-cylinder truck engine of 4.53-in. bore by 6.70-in. stroke by Dr. A. Loschge of Munich. Pressure-time diagrams were taken with a Farnboro electric indicator from both the pressure cell and the combustion chamber, without fuel injection and also when the engine was idling and operating under different loads. Fig. 6 shows two such diagrams from the two chambers superposed, these having been taken when the engine developed a bmep of 85 psi.

It will be seen that throughout the compression stroke the pressure is lower in the energy cell than in the combustion chamber. In crossing the dead-center line, both pressure lines are upwardly inclined, which indicates that combustion has already begun, but the rate of pressure rise is still relatively slow. About 5° past top center there occurs a very rapid rise in the energy cell, to a peak pressure of nearly 1200 psi. A similarly rapid rise in pressure starts in the combustion chamber about 5° later, but there the pressure increase is much smaller, the maximum value attained being only about 630 psi. In the energy cell, after the pressure has attained its peak value, it drops again very rapidly, which is no doubt due to the rapid blowing-off effect, and for a brief space of time this pressure drops below that in the combustion chamber, which must induce a reverse flow. The great excess of pressure in the energy cell during the early part of the expansion stroke must result in a violent swirl in the combustion chamber.

Combustion Characteristics of Energy-Cell Engine—A series of tests on two Lanova engines was run at Munich by Dr. S. J. Davies of London. In Fig. 7 the author has plotted some of the results obtained by Dr. Davies from a four-cylinder 4.72 by 7.09-in. engine at 1400 rpm, on a basis of bmep. At this speed the engine developed a maximum bmep of 94.6

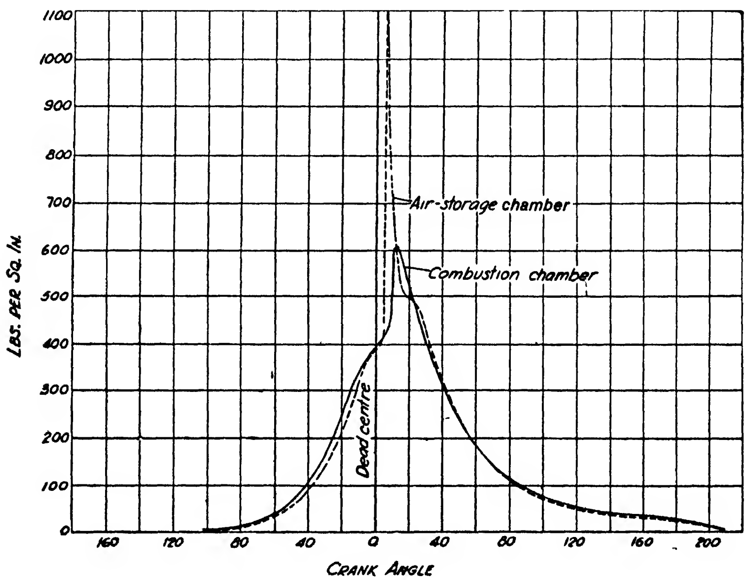


FIG. 6.—PRESSURE-TIME CURVES FROM ENERGY CELL AND COMBUSTION CHAMBER OF LANOVA ENGINE.

psi. To those familiar with gasoline-engine exhaust temperatures, the temperatures of the exhaust from this engine will seem very low. The specific fuel consumption at this speed reached a minimum value of 0.42 lb per hp-hr. In determining the volumetric efficiency, a correction was made to account for the fact that Munich has considerable altitude, and the atmospheric pressure at the time of the tests was only 13.8 psi. Another factor determined that is not covered in the chart was the degree of air utilization; in other words, the ratio of the amount of air theoretically required to combine with the fuel consumed, to the amount of air actually taken in by the engine. This factor naturally varies with the load, and for maximum load it attained a value of 0.808. It is more common in Diesel practice to express this relationship in the form of an excess-air factor, and an air-utilization coefficient of 0.808 corresponds to an excess-air factor of 1.24.

Referring to Fig. 7, the scale marked "Vol. Efficiency, Per Cent" is really a scale of relative charge density. The volumetric efficiency, as now generally defined, does not vary with the density of the atmosphere, but the mass of charge taken

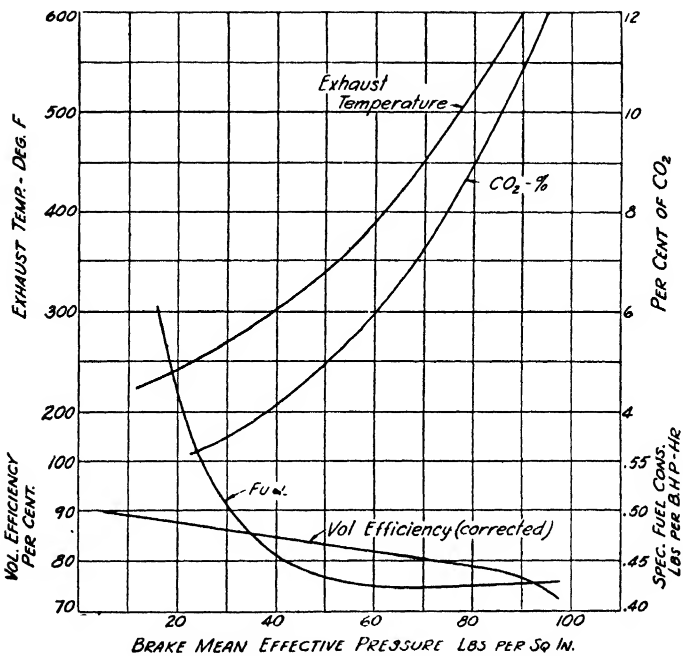


FIG. 7.—TEST RESULTS FROM LANOVA ENGINE (DAVIES).

in by the engine does, being proportional to the product of the volumetric efficiency and the relative atmospheric density.

Mack Engine—Mack Motor Truck Corporation since 1938 has been manufacturing Diesel engines embodying the energy-cell combustion principle. These engines are fitted into trucks, and a large number of Diesel-engined trucks were delivered to the armed services during World War II. A cross section of the END-672 truck engine is shown in Fig. 8. With a bore of 4⅞ and a stroke of 6 in., the six-cylinder engine has a piston displacement of 672 cu in. and is rated 165 bhp at 2000 rpm.

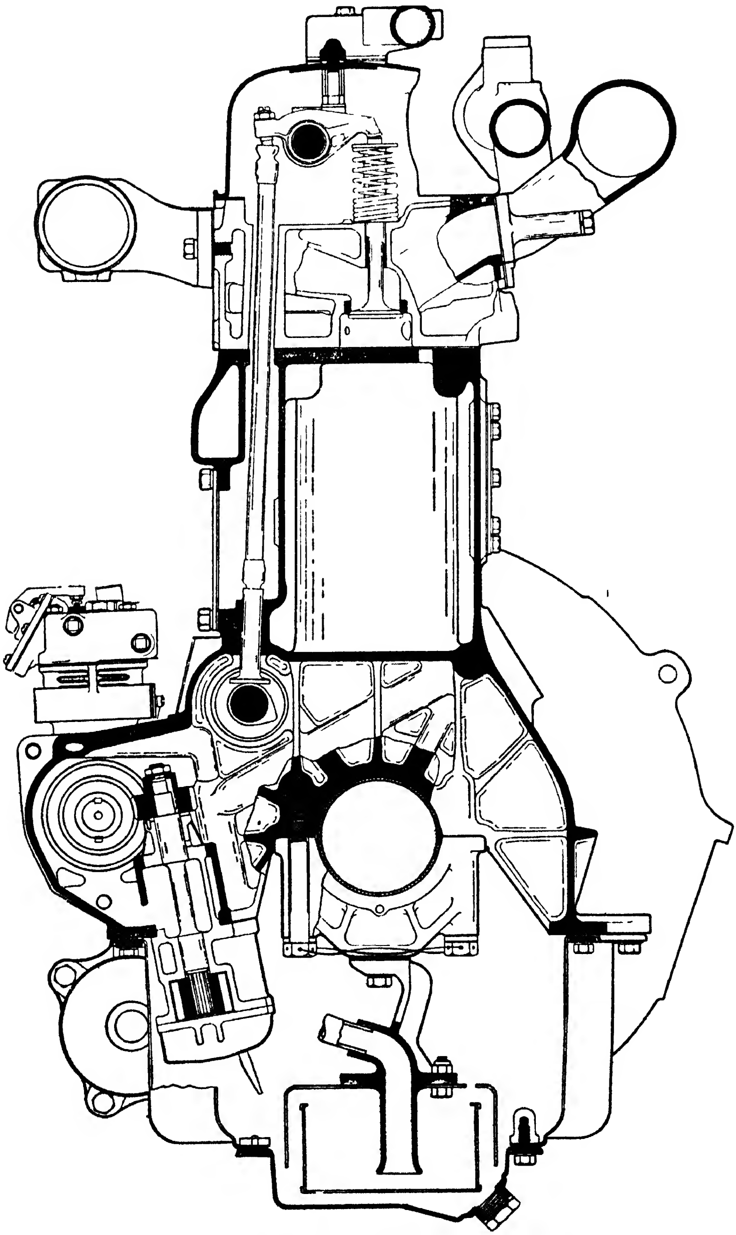


Fig. 8.—Cross Section of Mack-Lanova Engine.

The cylinder block is cast of nickel-chromium semi-steel, in a single block that forms a compact unit of great vertical stiffness. Cylinder heads are cast in two interchangeable blocks of three, and are held to the cylinder block by 32 studs. The cylinders are provided with "dry" liners of nickel-chromium semi-steel. Asbestos-copper gaskets are interposed between the cylinder heads and the block.

Both intake and exhaust valves have 45-degree seats. The exhaust valves seat on Niferite inserts that are copper-plated and provided with Stellite facings. Valve covers are of pressed steel, and to make it unnecessary to exert great pressure on the covers to prevent oil leakage from the compartments, the cylinder heads are recessed so that the oil in the compartments never reaches the gasket level. There are seven main bearings on the crankshaft, of 4-in. diameter, while crank-pin bearings are of 3-in. diameter. All bearing surfaces are case-hardened.

Pistons are of aluminum alloy, of the cam-ground, T-slot type, which permits of a relatively close cold clearance. Piston pins are lubricated with oil scraped off the cylinder wall, that gets to them through drain holes in the piston. The camshaft and accessories are driven through a train of four gears. These gears are made from drop-forged steel blanks, which are cut with helical teeth that are ground in generating machines. The equipment of the engine includes Bosch injection units, a coolant-temperature-control valve with by-pass, an oil-temperature regulator, an oil filter, and a crank-case ventilator with an air filter on the inlet.

CHAPTER XII

Two-Stroke Engines

The operating cycle of the two-stroke engine was described in Chapter I. It was there explained that, since in this engine there is no separate exhaust stroke, the burnt gases must be blown out with a scavenging medium. In carburetor-type two-stroke engines combustible mixture serves as a scavenging medium. Some of this mixture escapes through the exhaust ports with the burnt gases, with the result that the specific fuel consumption is relatively high. Carburetor-type two-stroke engines therefore are used only in services where the number of hours of operation per year is small and the investment in the engine must be kept to a minimum. In designing a carburetor-type two-stroke engine, the aim always is to provide such porting that practically all of the burnt gases will be swept from the engine with a minimum loss of combustible mixture. In a Diesel two-stroke engine, on the contrary, a considerable amount of fresh air often is purposely blown through the engine during the scavenging period, because of its cooling effect on the piston and the combustion-chamber walls.

Crankcase Scavenging—The simplest type of two-stroke engine, and the one which has been most widely used as a carburetor engine, is that employing the crankcase as a scavenging pump. Crankcase scavenging can be used also for Diesel two-stroke engines, and such engines have been used to a certain extent in the past for industrial locomotives, for driving electric generators, oil-drilling rigs, pumps, etc. But the bmep obtainable with crankcase scavenging is only about one-half that of the average four-stroke Diesel engine, and where high weight efficiency is required, crankcase scavenging is impractical.

Fig. 1 is a cross section of a two-stroke Diesel engine with crankcase scavenging. During the up-stroke of the piston air is admitted to the crankcase through a check valve fitted over a large opening cast in its side (the right side in the drawing). This air is compressed in the crankcase during the down-stroke, and when the piston is at the bottom of the stroke, as shown in the drawing, the air-inlet valve is closed and there is an over-pressure in the crankcase. Since both

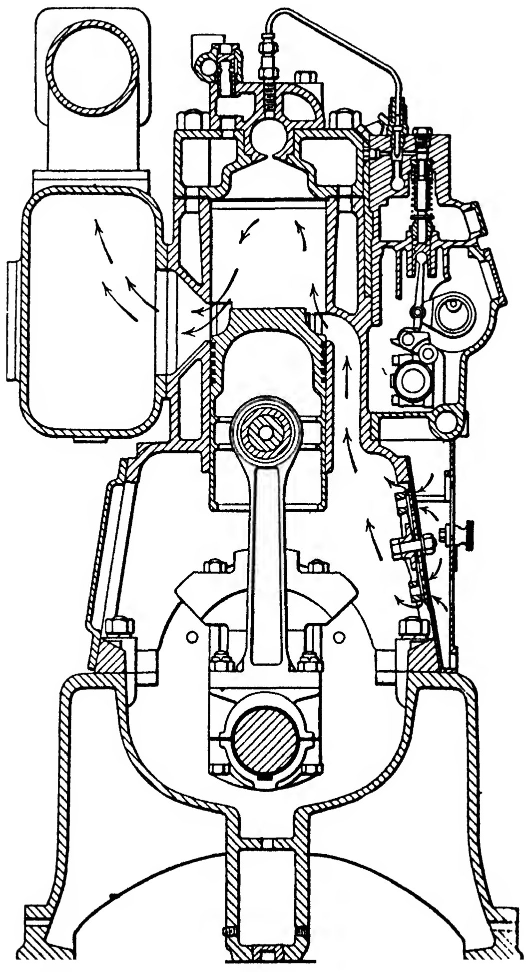


FIG. 1.—CROSS SECTION OF CRANKCASE-SCAVENGED TWO-STROKE ENGINE.

the transfer ports and the exhaust ports are now fully uncovered by the piston, air flows from the crankcase through the transfer passage into the cylinder. Upon entering the cylinder at its lower end, the current of air is directed upward by a reduced upward extension of the piston, now directly opposite the ports. While air is thus entering the cylinder through ports on the right, the burnt gases escape from it through ports at the left, as indicated by the arrows.

Advantages of Two-Stroke Cycle—Certain advantages of the two-stroke cycle for Diesel engines have long been recognized, and whereas in the gasoline-engine field the two-stroke engine is almost extinct, except for small sizes or where periods of use are short and separated by long intervals, two-stroke Diesel engines have been extensively used in large sizes, especially in marine work. In 1941 a listing of American manufacturers showed 34 to be producing four-stroke and 14 two-stroke engines. It is true that in the field of high-speed engines the number of manufacturers of the four-stroke type predominates greatly, but that may be due to the fact that most of these manufacturers were producing gasoline engines of the four-stroke type before entering the Diesel field, and therefore were more familiar with that type.

A definite advantage of the two-stroke engine is that practically the same degree of torque uniformity can be obtained with half the number of cylinders as in a four-stroke engine. If the engine had only half the number of cylinders, only one-half the number of injection units—pump elements and injectors—would be needed, and that would materially reduce the cost of the injection equipment. In practice, however, this advantage is not fully realized, for most truck and bus engines of between 100 and 200 hp have six cylinders, whether they operate on the two-stroke or four-stroke cycle, and the mean number of cylinders of American railroad two-stroke Diesel engines is only 15 per cent less than that of the four-stroke engines.* However, as there are nearly twice as many power strokes in the two-stroke engine in unit time, its torque curve must be smoother.

Two-stroke engines of the high-speed type develop practically the same maximum bmeps as equivalent unsupercharged four-stroke engines, and at practically the same rpm, hence their output per unit of displacement is about twice as great. The weight per horse power is considerably less for the two-stroke, even though the latter must be provided with a blower. The blower usually is so designed that it delivers nearly 50 per cent more air per cycle than the cylinder could draw in by natural induction, but most of the excess air deliv-

* Shortly after this was written GMC Truck and Coach Division announced a line of light trucks equipped with the three-cylinder Type 71 GM two-stroke engine.

ered is blown through the cylinders to produce an internal cooling effect, and is not used to supercharge the engine. The cylinders, however, are completely scavenged, and if they are filled to atmospheric pressure at the moment the inlet ports close, they contain more air than the cylinders of an equivalent four-stroke engine with atmospheric induction. This greater air charge seems to just about compensate for the losses resulting from earlier opening of the exhaust port in the two-stroke and from power absorbed by the blower, so that the bmeps are substantially the same in both engine types.

Jacket Losses—Owing to the fact that the time for heat transfer to the water jacket is shorter in the two-stroke engine, and also because an engine equipped with a blower is internally cooled, the jacket losses bear a smaller proportion to the engine output than in the four-stroke type. Experimental data from a four-stroke and a two-stroke engine, both of 140-

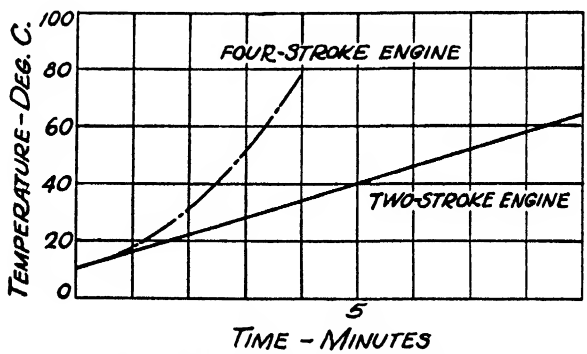


FIG. 2.—RATES OF HEAT TRANSFER TO WATER JACKET IN TWO- AND FOUR-STROKE ENGINES.

hp rating, are plotted in Fig. 2, which shows that the temperature of the jacket water rose nearly twice as fast in the four-stroke as in the two-stroke.

With some types of two-stroke engine, such as the Junkers, in which the combustion takes place between two opposing pistons, only a small portion of the combustion-chamber wall area is in direct contact with water at the beginning of the power stroke. Most of the wall area is made up by the two piston heads, which are at a much higher temperature than the water-cooled cylinder wall, and the rate of heat absorption per unit of area is materially lower for the piston head than for the cylinder wall. This, of course, is not a direct result of the use of the two-stroke cycle, but rather a result of the particular design. In the conventional engine a large part of the heat carried off by the jacket water enters the jacket through the cylinder head, which is in contact with the burning gases longest.

Where the exhaust is through piston-controlled ports, a slight reduction in the jacket losses might be expected also from the fact that such ports usually open more rapidly than cam-actuated poppet valves. In any case, in the two-stroke engine the exhaust is completed some 50 to 60 deg past bottom dead center, whereas in a four-stroke engine it continues for substantially 180 deg past that point.

Heat Stresses—From the foregoing it must not be inferred that cooling problems are less troublesome in the two-stroke engine, for the exact opposite is the case. In any engine, the maximum temperatures attained by such parts as piston head, valve head, etc., vary directly with the frequency at which combustion cycles follow one another, and since for the same speed there are twice as many combustion cycles per unit of time in a two-stroke engine, it can be readily understood that the parts directly exposed to the flame must attain materially higher temperatures. Many two-stroke engines, of course, have no poppet valves, but cooling of the piston heads presents quite a problem, which is the more difficult the greater the bore and the higher the speed of operation.

If the exhaust takes place through ports uncovered by the pistons, during the exhaust period all of the hot gases sweep by the piston head at high velocity, in consequence of which the rate of heat absorption by the piston is much greater than in an equivalent four-stroke engine. Trouble is caused also by the fact that when the exhaust ports begin to open, the hot gases come in contact with the outside of the piston over a length which at the start is substantially equal to the depth of the ports. This tends to burn the oil in the ring grooves and to cause the rings to stick. Once the rings are "frozen" in their grooves, they can no longer perform their sealing function efficiently, excessive blow-by occurs, and the piston is quite likely to be seriously injured, especially if it is a light-alloy piston.

Piston Cooling—One way to keep down the piston temperature is to make the head of quite heavy section, so that the heat absorbed can flow off more readily. However, the most effective way is to cool the piston artificially, and some method of artificial cooling is quite essential in high-speed two-stroke engines of large bore. Such cooling can be effected by throwing a jet of lubricating oil against the under surface of the piston head; by forming a closed chamber under the piston head and circulating oil through it by means of sliding connections with stationary pipes, or by carrying some of the heat from the head to the lower part of the skirt by a method similar to that employed in sodium-cooled valves; that is, filling cavities formed in the piston about half full with metallic sodium, which melts at the operating temperature and then conveys heat from the hot crown to the cooler skirt.

It is most important to protect the top piston ring against excessive temperatures, so as to prevent the oil in its groove from coking and "freezing" the ring. To this end a wide land may be provided above the top ring and the section at this land made thin, so it will act as a heat dam.

Types of Scavenging Pumps—In a two-stroke engine with crankcase precompression, which is the oldest type, the displacement of the scavenging pump is, of course, exactly the same as that of the working cylinder, since both displacements are effected by the same piston or pistons. Owing to the fact that there is always a certain loss of scavenging medium through the exhaust ports, and also because the clearance volume in the crankcase is large, the amount of charge received by the working cylinder per cycle is less than a similar cylinder would draw in under equivalent conditions of speed, port-timing, etc., when working on the four-stroke cycle. This is reflected by the lower bmep of two-stroke engines as compared with four-stroke engines of the same general design. For instance, the crankcase-scavenged engine shown in Fig. 1 developed a bmep of only 40.4 psi at its rated speed, whereas four-stroke engines develop from 70 to 100 psi at their speeds of maximum output.

Crankcase precompression for scavenging purposes is generally recognized as unsuitable for use where light weight or high specific output is desirable, and in such cases separate scavenging pumps are always used. There is a choice of at least five types of pumps, viz., the piston type, the Roots-blower type, the axial-flow positive-displacement type, the vane type, and the centrifugal type. The latter, however, is not well suited to bus and truck engines, as it raises the high-speed torque of the engine more than its low-speed torque, causing the torque to increase with the speed through practically the whole operating range, which deprives the engine of all "lugging" power. A number of the earlier two-stroke Diesel engines were provided with piston-type scavenging pumps, probably because the designers were most familiar with that type. At present the Roots type of blower is in most extensive use in this country, while abroad the vane type has a certain vogue. Centrifugal blowers can be used to advantage where the torque load decreases rapidly with the speed (propeller load).

Roots Blower—Blowers in which the motion of the parts is purely rotary naturally offer advantages, and considerable use has been made of what is known as the Roots blower (originated many years ago for industrial purposes by the Roots-Connersville Corporation, Connersville, Ind.). It is essentially a gear pump, two "rotors" with either two, three or four lobes each being enclosed in a suitable housing. As shown in Fig. 3, which represents a Roots blower with two-lobed rotors, the inlet is located centrally on one side and the outlet on the opposite side of the housing, the air being carried around from the inlet to the outlet side in the spaces

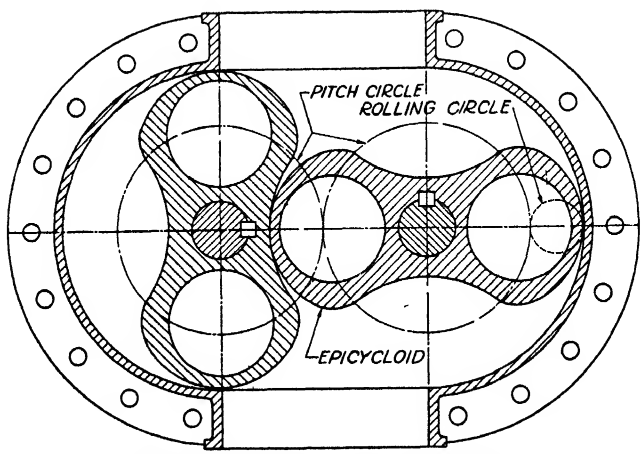


FIG. 3.—TRANSVERSE SECTION THROUGH ROOTS BLOWER.

between the lobes and the housing. The rotors are so designed that they do not come in contact with each other or with the housing, but the clearances are made as small as manufacturing and operating conditions permit. They are usually of the order of 0.004-0.006 in. Leakage through the clearances tends to reduce the volumetric efficiency at low speeds. However, owing to the resistance to air flow of the air cleaner and inlet pipe, the density of the air in the spaces between the rotors and the housing decreases as the speed increases, and this has a preponderating effect on the volumetric efficiency. F. G. Shoemaker has shown that the volumetric efficiency of the tri-lobed blower on the General Motors Type

71 engine decreases from slightly over 85 per cent at 500 rpm engine speed (970 rpm blower speed) to 83 per cent at 2000 rpm engine speed.

One of the two rotors receives the drive directly. Since there must be no contact between rotors, it is impossible to transmit the drive to the second rotor through the lobes or teeth, as is done in a gear pump, and the shafts of the two rotors must be placed in driving connection by means of gears, which are enclosed in a separate housing at one end. It is essential that the two rotors should be accurately spaced angularly; that is to say, in a blower with two-lobed rotors, they should be accurately at right angles to each other. To facilitate fine adjustments of the angular relation, the gears may be secured to the shafts by splines, and the number of teeth in the gears and the number of splines be so chosen that a vernier adjustment is possible. For instance, if there are twelve splines, and each of the gears has, say, 25 teeth, then moving one gear around two teeth relative to its mate moves one of the shafts through an angle of $(2 \times 360)/25 = 28.8°$, and by then moving the rotor one spline on the shaft, it is turned through an angle of $360/12 = 30°$ in the opposite direction, so that an adjustment of $30 - 28.8 = 1.2°$ is obtained. Still finer adjustments may be had with larger numbers of teeth and splines.

Form of Rotor Lobes—The lobes of the rotors are made of epicycloidal form. That is to say, the outline of the lobe is generated by a point in the circumference of a generating circle rolling on the pitch circle. Since the two rotors have the same number of lobes, the radius of the pitch circle is equal to one-half the center distance between rotors, and for two-lobed rotors the radius of the generating circle is made equal to one-quarter the radius of the pitch circle. To reduce the inertia of the rotors, the lobes must be made hollow. This reduces their resistance to deformation under centrifugal force, and to stiffen them they may be ribbed internally, or, alternately, circular discs may be welded into the ends of the openings in the lobes. Owing to the fact that the load on the rotors is a pulsating one, there is a tendency for the gear teeth to wear, and when the teeth are worn the rotors are likely to come in contact, which always causes abnormal noise. The volumetric efficiency usually decreases somewhat with length of service. Small blowers cannot be made quite as efficient as larger ones, owing to the fact that the clearance cannot be reduced in direct proportion to the linear dimensions. One disadvantage of early Roots blowers was that they were usually very noisy in operation. The noise can be reduced by using rotors with a larger number of lobes than two, using rotors with helical lobes together with ports whose lengthwise edges are parallel to the rotor axis or, alternately, rotors with straight lobes together with ports whose edges are inclined to the rotor axes.

Characteristics of Blower—The power consumed by a blower when compressing 1000 cu ft of free air per minute

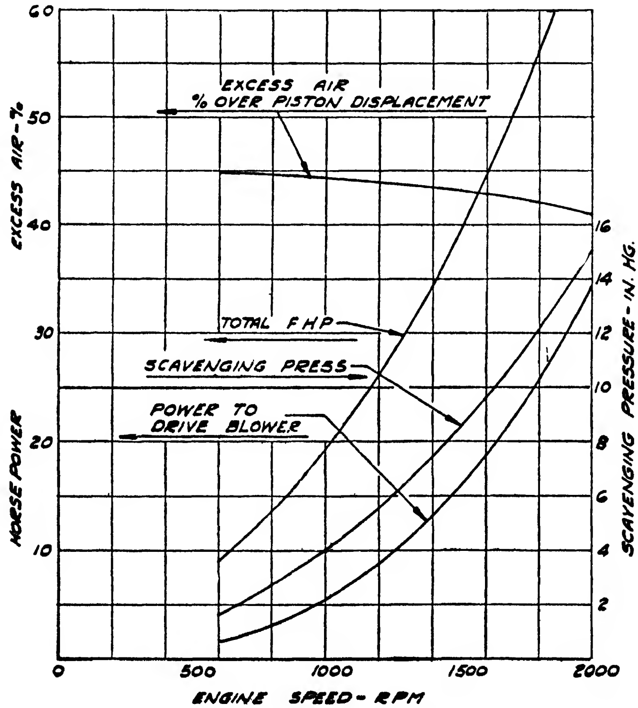


FIG. 4.—PERFORMANCE CURVES OF G.M. ROOTS-TYPE BLOWER.

from normal atmospheric pressure of 14.7 psi to an absolute pressure p psi is given by the equation

$$P = \frac{221.5}{\eta}\left[\left(\frac{p}{14.7}\right)^{0.29} - 1\right]\text{hp},$$

where η is the so-called adiabatic efficiency, which usually is somewhere between 70 and 75 per cent. At other rates of delivery the power consumed is directly proportional.

Performance curves of a three-lobed Roots-type blower used on General Motors two-stroke six-cylinder engines of 425.4 cu in. piston displacement are shown in Fig. 4. This blower turns at 1.94 times crankshaft speed, and its air delivery at 2000 engine rpm is 943 cu ft per minute. It will be seen that the delivery pressure (scavenging pressure) varies substantially as the square, and the horse power consumed nearly as the cube of the speed. Since the ratio of engine piston displacement to blower displacement is constant, the curve of excess air in the chart may serve also as

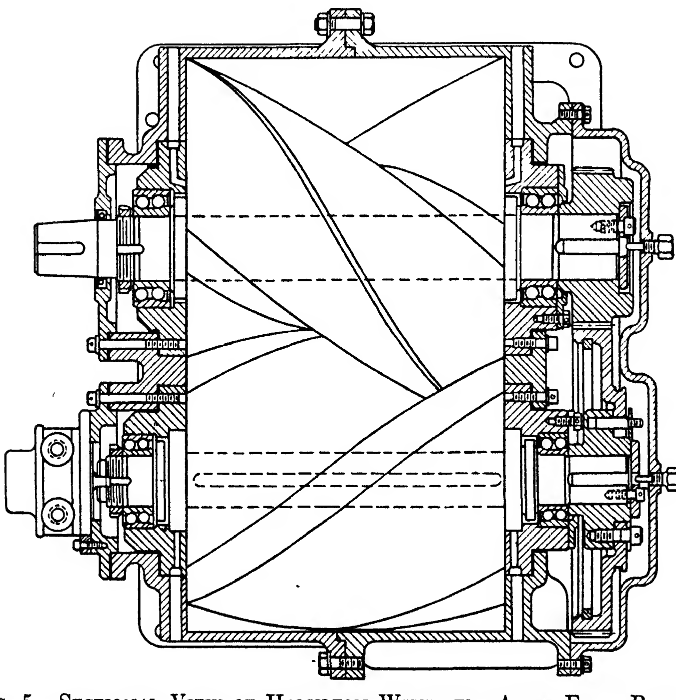


Fig. 5.—Sectional View of Hamilton-Whitfield Axial-Flow Blower.

a curve of blower volumetric efficiency, which latter in this particular case is given by the expression 0.59 (100 + per cent excess air).

Axial-Flow Rotary Blower—In the Roots-type blower the air may be considered to flow circumferentially through the housing. It is possible also to so arrange two intermeshed rotors that they will displace air trapped in the tooth spaces or thread spaces axially. One such axial-flow blower is the Hamilton-Whitfield, which is being manufactured by the

General Machinery Corporation of Hamilton, Ohio. A section through the axes of the rotors is shown in Fig. 5. This blower has two helically threaded members mounted in ball bearings in a housing, with their threads intermeshed to form a continuous sealing line over the whole length of their engagement. The housing seals the perimetral edges of the rotors, and the only path through the blower is through the troughs of the threads. Inlet and outlet ports are located at opposite ends of the housing and communicate directly with the troughs in the threads.

The larger or main rotor, which is driven directly, has two helical threads with a helical advance of about 180 deg over the length of the rotor. It meshes with a smaller rotor known as the gate, which has four helical threads with a helical advance of about 90 deg from one end of the rotor to the other. As in the case of helical gears, the threads of the intermeshing rotors are of opposite hand. Both rotor-thread forms are symmetrical and are generated. The sides of the threads on the main rotor are described by the continuous edges of the helical threads of the gate rotor, while the troughs of the gate rotor are described by the edges of the crests of the helical threads on the main rotor. The two rotors are connected by a pair of 2:1 gears located in a separate compartment of the housing opposite the driving end.

When the rotors are in motion, their threads intermesh to form pockets which advance axially from the inlet to the outlet end. If the direction of rotation is reversed, the flow is reversed, and the inlet becomes the delivery port. The blower ports are diagonally opposed on opposite sides of the plane through the rotor axes, and formed partly in the heads and partly in the side walls. The adiabatic efficiency of this blower is said to range between 70 and 76 per cent, and the volumetric efficiency between 88 and 92 per cent.

Elliott-Lysholm Blower—Another type of rotary blower, in which the air is transported diagonally by a pair of intermeshing, helically lobed rotors, is the Elliott-Lysholm, which was invented by Alf Lysholm, chief engineer of the Ljungstrom Steam Turbine Company of Stockholm, Sweden, and is being manufactured in this country by Elliott Company, Jeanette, Pa. Two sectional views of the blower are shown in Fig. 6. The distinguishing feature of this device is in its action on each air charge after the latter is sealed off from the inlet and before it is brought into communication with the discharge. A charge is first trapped in the space bounded by the thread flanks, the casing and end walls. On the convex-lobed rotor the helices have a wrap of about 240 deg, and a particular thread space is filled and sealed off from the inlet port at the very moment when a coacting lobe on the other rotor enters it at the opposite end. As rotation continues, an axial seal is established which separates this charge from charges in succeeding thread spaces. The seal moves

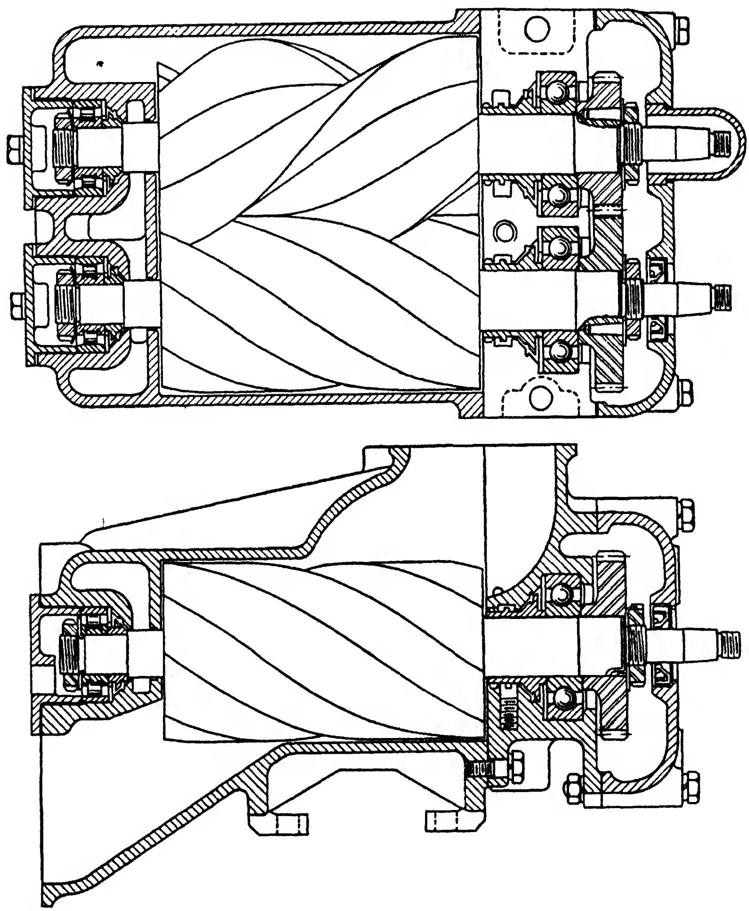


FIG. 6.—TWO SECTIONAL VIEWS OF ELLIOTT-LYSHOLM AXIAL-FLOW BLOWER.

axially, and as a result the volume of the charge is reduced and the latter is compressed substantially adiabatically. When the leading lobe of the space passes the edge of the discharge port, the space begins to discharge.

Compressing the air adiabatically is claimed to be more efficient than compressing it at constant volume, as in a Roots blower, for instance. In the latter a charge of air at nearly atmospheric pressure is carried around from the inlet to the discharge side of the blower. When the leading lobe passes the edge of the discharge port, air from the discharge rushes back into the space between lobes, and fills it to discharge pressure. Thereafter the trailing lobe is subjected to discharge pressure on one side and to suction pressure on the other, and it must be moved against this constant pressure difference. The pressure-volume diagram of its compression cycle is therefore a rectangle. In the Lysholm blower, on the other hand, the pressure of the air in the lobe spaces

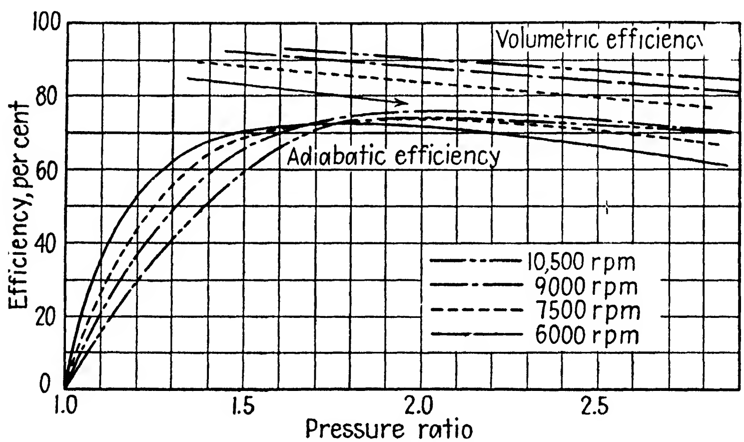


Fig. 7—Efficiencies of Elliott-Lysholm Blower.

increases gradually as the volume of the sealed chamber decreases, and at the moment the chamber is opened to the discharge, the pressure in it is slightly above that in the reservoir into which the blower delivers. The ratio of the work which must be done when air is compressed adiabatically to that which must be done when it is compressed by the constant-volume process varies from 1.00 for a pressure ratio of 1.00 to 0.80 for a pressure ratio of 1.8 in a substantially straight-line manner. Volumetric and adiabatic efficiencies of a Lysholm 1100 cfm compressor for different speeds and different pressure ratios are shown in Fig. 7.

Vane-Type Blowers—Another type of blower that can be used for supercharging is the vane type, of which the Zoller and the Centric are examples. The latter is illustrated in longitudinal and transverse sections in Figs. 8 and 9 respectively. Each of the four vanes is separately hinged on a central spindle by means of ball bearings. The vanes rotate at speeds of up to 5000 rpm, but as they are individually counterweighted there is little load on the ball bearings. Very small clearances are maintained between the vanes and the housing, and the volumetric efficiency is said to be well maintained at low speeds.

The vanes are driven against the pressure head through the displacement drum, which is mounted eccentric to the housing and spindle. The drum, which is mounted on its own ball bearings, drives the vanes through trunnion blocks and is in turn driven from the central spindle through a pair of internal and external gears. This arrangement has the advantage of affording a "through drive," but it reduces the effective speed to two-thirds that of the driving shaft. The trunnion blocks, which are slotted, cylindrical rods of Fabroil, drive the vanes and permit of a limited amount of rocking or wobble, and seal the working space from the central compartment.

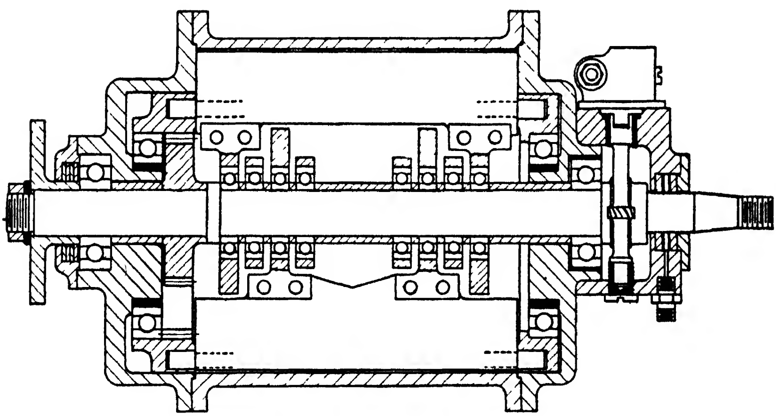


FIG. 8.—LONGITUDINAL SECTION OF CENTRIC VANE-TYPE BLOWER.

Owing to the fluctuations of the rates of displacement of the engine pistons and the blower rotors or vanes, these blowers cannot be connected directly to the inlet ports of the engines, but must deliver into an air-storage chamber, which latter communicates with the inlet ports of all cylinders.

Blowers are discussed further in the chapter on Supercharging.

The Scavenging Process—In the two-stroke engine the bmep is reduced somewhat by the fact that the exhaust ports (or valves) begin to open when between 30 and 35 per cent of the "in" stroke of the power piston is still to be completed, and the pressure in the cylinder then drops rapidly, because of the rapid increase in the port-opening area. In the conventional crankcase-scavenged two-stroke engine both the inlet and the exhaust ports are located at the bottom end of the cylinder, the top edge of the exhaust ports being somewhat higher than the top edge of the inlet ports, so that the exhaust will open some time before the inlet opens, during which time the pressure of the burnt gases in the cylinder drops to that of the scavenging air. Most of the remaining burnt gases are swept out by the scavenging air. Since the inlet and exhaust ports are located on opposite sides of the cylinder, to prevent the incoming scavenging air from passing right across from the inlet to the exhaust ports, it is common practice in low-compression (carburetor-type) two-stroke engines to provide a deflector on the piston head directly opposite the transfer ports. In a high-compression engine with its shallow compression space there is not enough room for such a deflector, and a similar effect is then obtained by providing channels in the piston opposite the ports, as in Fig. 1. The current of air entering the transfer port is supposed to be directed upward by the deflector or its equivalent, and upon striking the cylinder head, to be turned back toward the exhaust ports, sweeping the burnt gases before it. In practice, however, this scavenging action is far from being perfect, and there is always a tendency for the burnt gases to be trapped in the upper part of the cylinder, while the scavenging air takes a short cut from the inlet to the exhaust ports. For this reason the practice of placing the inlet and exhaust ports at opposite ends of the cylinder, which results in what is referred to as uniflow scavenging, has come into use in high-speed two-stroke Diesel engines.

Port Capacities—To be able to operate at high speeds, a two-stroke engine must be provided with adequate port capacity. The capacity of a port varies more rapidly than the depth of the port in the direction of the cylinder axis, because any depth that may be added at the upper end is effective for a longer time than an equivalent depth at the lower end. The capacity of any small element of the port of depth dh and width w is proportional to the expression $\alpha w\, dh$, where α is the angle of crank motion during which this element is uncovered by the piston. The values of w and dh being given in inches and that of α in degrees, the port capacity is measured in square inch-degrees. The angle α, through which the crank passes from the time the port begins to open until the lower dead-center position is reached, is dependent upon the relation between the length of stroke l and the distance h from the upper edge of the port to the top edge of the piston when at the bottom of its stroke. We will assume that the lower edge of the port is on a level with the top edge of the

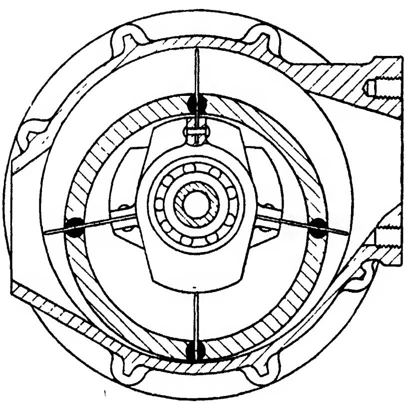
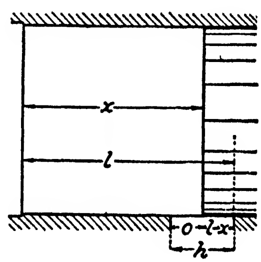


FIG. 9 (*Left*).—CROSS SECTION OF CENTRIC BLOWER.
FIG. 10 (*Below*).—PORT-OPENING DIAGRAM.



piston when the latter is at the bottom of the stroke, which is the usual arrangement.

It is shown in books dealing with the mechanics of the internal combustion engine that the distance x which the piston moves down on the stroke while the crank turns through an angle θ is given by the equation

$$x = \frac{l}{2}(1 - \cos\theta) + \frac{l}{8n}\sin^2\theta.$$

Referring to Fig. 10, the top of the piston evidently begins to uncover the port when $x = l - h$, l being the length of stroke and h the height of the port. Then

$$\frac{l}{2}(1 - \cos\theta) + \frac{l}{8n}\sin^2\theta = l - h$$

and

$$h = l(0.5 + 0.5\cos\theta) - \frac{l}{8n}\sin^2\theta.$$

Now let us designate by α the angle through which the crank passes from the time the port begins to open until the bottom dead center is reached. Then $\theta = 180° - \alpha$; $\cos\theta = -\cos\alpha$, and $\sin\theta = \sin\alpha$. Making these substitutions and on the assumption that n, the ratio of connecting-rod length to length of stroke, is equal to 2 we get

$$h = l(0.5 - 0.5\cos\alpha - 0.062\sin^2\alpha).$$

It will be seen from this equation that for a given angle α of port opening the depth of port h is directly proportional to the length of stroke l, and the following ratios of l/h have been calculated:

α =	40°	42°	44°	46°	48°	50°	52°
l/h =	13.0	11.6	10.4	9.35	8.48	7.70	7.00
α =	54°	56°	58°	60°	62°	64°	66°
l/h =	6.42	5.96	5.50	5.10	4.77	4.45	4.17

Any difference there may be between the actual value of the ratio n and the value here assumed (2) will make practically no difference in the port timing.

Port-Opening Integral—From Fig. 10 it can be seen that the width of opening o of the port when the piston has moved down a distance x from the top end of the stroke is

$$o = h - (l - x) = h + x - l.$$

Substituting the value of x given in the foregoing we have

$$o = h - \frac{l}{2} - \frac{l}{2}\cos\theta + \frac{l}{8n}\sin^2\theta.$$

In Fig. 11 is shown a coordinate diagram in which the abscissas represent crank angles and the ordinates length of port opening. Let $d\theta$ represent any small increment in the crank angle and o the corresponding length of opening of the port. Then the product $o\,d\theta$ represents the area of the narrow strip of width $d\theta$ and height o. The total area enclosed by the port-opening curve and the horizontal axis may be called the port-opening integral and denoted by I. Inserting the value of o in the expression for the area of the narrow strip, we have

$$o\,d\theta = h\,d\theta - \frac{l}{2}d\theta - \frac{l}{2}\cos\theta\,d\theta + \frac{l}{8n}\sin^2\theta\,d\theta.$$

To find the area of the entire diagram we have to integrate this expression between the limits θ_1 and θ_2, these being the crank angle corresponding respectively to opening and closing of the port.

$$\int_{\theta_1}^{\theta_2} h\,d\theta - \frac{l}{2}\,d\theta - \frac{l}{2}\cos\theta\,d\theta + \frac{l}{8n}\sin^2\theta\,d\theta$$

$$= \left(h - \frac{l}{2}\right)(\theta_2 - \theta_1) - \frac{l}{2}(\sin\theta_2 - \sin\theta_1)$$

$$+ \frac{l}{16n}(\theta_2 - \theta_1) - \frac{l}{16n}[(\sin\theta_2\cos\theta_2 - \sin\theta_1\cos\theta_1)].$$

In applying this equation, angles θ_1 and θ_2 must be entered in radians and the results will be in inch-radians. To

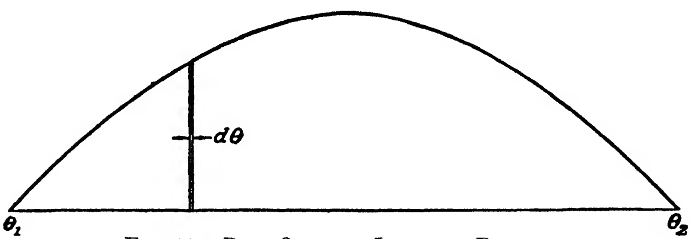


FIG 11—PORT-OPENING-INTEGRAL DIAGRAM.

change to inch-degrees it is only necessary to multiply by 57.3 ($= 360/6.28$). However, the equation can be materially simplified. If we denote by α the angle through which the crank must turn from the time the port begins to open to the moment it reaches the lower dead-center position, then

$$\theta_1 = 180° - \alpha; \quad \theta_2 = 180° + \alpha; \quad \theta_2 - \theta_1 = 2\alpha;$$

$$\sin\theta_1 = \sin\alpha; \quad \sin\theta_2 = -\sin\alpha;$$

$$\cos\theta_1 = -\cos\alpha; \quad \cos\theta_2 = -\cos\alpha.$$

Making these substitutions and giving n its usual value, 2, we obtain

$$I = \left(h - \frac{l}{2}\right)2\alpha + \frac{l}{2}(2\sin\alpha) - \frac{l}{16n}(2\alpha - 2\sin\alpha\cos\alpha)$$

$$= l\sin\alpha\left(1 - \frac{\cos\alpha}{8n}\right) - \left(l - 2h - \frac{l}{8n}\right)\alpha.$$

In this equation α must be inserted in radians and the result will be in inch-radians. To obtain the result in inch-degrees directly, α is inserted in degrees in the right-hand term, and the left-hand term, which involves functions of α, is multiplied by 57.3 ($= 360/6.28$). This leads to the equation

$$I = 57.3l \sin \alpha \left(1 - \frac{\cos \alpha}{8n}\right) - \alpha \left(l - 2h - \frac{l}{8n}\right) \text{ in.-deg.}$$

It was shown previously that for any given angle α, h is directly proportional to the length of stroke l, hence I also is proportional to l. Values of the ratio I/l for different angles α are given in the following table, and in calculating these values n again has been assumed to be equal to 2.

Angle α =	40°	42°	44°	46°	48°	50°	52°
I/l =	3.72	4.35	5.15	6.00	7.00	8.20	9.50
Angle α =	54°	56°	58°	60°	62°	64°	66°
I/l =	10.80	12.25	13.75	15.25	16.85	18.60	20.60

Graphic Method—Those who prefer graphic to algebraic methods can readily determine the port-opening integral practically without calculation. First the crank train is laid out with the crank at various angles from the lower dead-center position (in steps of 10°, for instance). Next a port-opening diagram is drawn, with the crank angles as abscissas to any convenient scale, and the widths of port opening as ordinates, preferably to a scale about four times actual size. A series of simple geometrical figures are then superimposed on this diagram, so chosen that their aggregate area is the same as that of the diagram (see Fig. 12). The "area" of each geometrical figure is then obtained by multiplying one-half of the represented width of opening, in inches, by the base in degrees in the case of triangles, and the mean width of opening in inches by the base in degrees in the case of quadrilaterals, the product in each case being in inch-degrees. The areas of all of the figures are then added together, and the sum multiplied by the width of the port is the port-opening integral represented by the diagram.

Sometimes the lower edge of the port does not coincide with the top of the piston when at the bottom of its stroke, but is located higher. In that case the port-opening integral can be found by first determining the value the integral would have if the port extended all the way down to the top edge of the piston when in its lowest position; then determining a port-opening integral for an imaginary port extending from the lower edge of the actual port to the top of the piston when in its lowest position, and finally subtracting the last value from that first obtained. This, of course, applies to both the graphic and the algebraic methods. The graphic method can be used also to check the results of computation.

Circular Ports—With engines of the uniflow type fitted with cylinder liners it has become customary to have the transfer ports in the form of one or more rows of drill holes extending all around the liner at the lower end of the stroke. This makes machining of the ports easier, obviates the stress

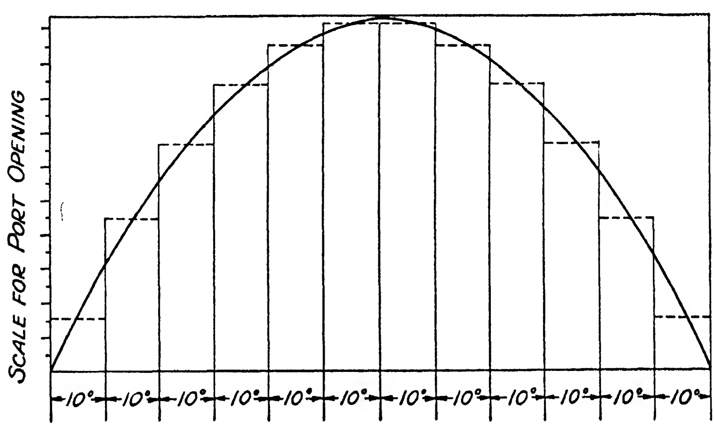


FIG. 12.—GRAPHICAL DETERMINATION OF PORT-OPENING INTEGRAL.

concentration which undoubtedly occurs at the corners of rectangular ports, and makes it practically impossible for the piston rings to catch on the edges of the ports. Capacities of such ports also can be determined either analytically or graphically, but the process naturally is somewhat more complicated, since both the width and the height of opening vary simultaneously.

Symmetrical and Unsymmetrical Timing—When a port in the cylinder wall is opened and closed by the piston, it closes the same number of crankshaft degrees after dead center as it opens ahead of dead center, and this is referred to as symmetrical timing. With such timing, since the exhaust port must open before the transfer port, it will close after the transfer port. This has the disadvantage that some of the air which has been forced into the cylinder by the scavenging pump during the charging period escapes through the exhaust ports during the period between transfer-port closing and exhaust-port closing. Moreover, supercharging is quite impossible with such a timing, because as long as the transfer port is open and the blower delivers air to the cylinder, the exhaust port is wide open and practically prevents the building up of a super-atmospheric pressure. High-speed, two-stroke engines therefore usually are given an unsymmetrical timing. The exhaust port or valve then opens ahead of the transfer port, but it also closes ahead of the transfer port. Such a timing can be obtained in various ways. For instance, instead of using ports uncovered by the piston for the exhaust, cam-actuated exhaust valves may be placed in the cylinder head, which can be timed as desired, regardless of the piston motion. Alternately, each cylinder or combustion chamber may be provided with two pistons connected to crank throws making a slight angle with each other, so that there is a small phase difference between the motions of the two pistons; one piston then is made to control the transfer port, the other the exhaust port. A third possibility lies in the use of a rotary valve in the passage just outside the transfer port, the latter being made as high as, or even higher than, the exhaust port. The transfer port then will open simultaneously with, or ahead of, the exhaust port, but no burning gases can get into the transfer passage, because the rotary valve is closed at the time. The beginning of the scavenging period then is determined by the timing of the rotary valve; it follows the opening of the exhaust port at an interval equal to the blowdown period, that is, the time required for the pressure in the cylinder to drop to the value of the scavenging pressure.

Non-Return Valve in Transfer Passage—The various schemes outlined in the preceding paragraph all involve a degree of complication, and to secure some of the advantages they offer in a simpler manner, check valves or non-return valves sometimes are placed in the transfer passage just outside the transfer port, which latter is then arranged to open as early as, or earlier than, the exhaust port. Such an arrangement is illustrated in Fig. 13. When the transfer port is uncovered by the piston, the gas pressure in the cylinder closes the non-return valve, and the burning gases cannot get into the transfer passage. Upon opening of the exhaust ports, the pressure in the cylinder drops rapidly, and when the pressures in the cylinder and the air chamber are equalized, the check valve opens and scavenging begins.

Port Capacities Required—In a two-stroke engine the port-opening periods, expressed in crankshaft degrees, are little more than one-half as long as the inlet and exhaust periods in a four-stroke, and for equal rotative speeds the port-opening areas of a two-stroke therefore must be proportionately larger than the valve-opening areas of a four-stroke engine with cylinders of the same size. In determining the port areas and port timing of a high-speed two-stroke engine, care must be taken that the exhaust-port capacity during the blowdown period is sufficient to permit the cylinder pressure to drop to the scavenging pressure by the time the transfer port opens. If we assume the exhaust ports to be of constant width, a plot of the instantaneous port-opening area vs. crank angle measured from the beginning of opening is substantially a right-angled triangle, the hypotenuse bulging outwardly very slightly, because the piston motion per degree of crank motion decreases as dead center is approached. If we designate the circumferential width of the exhaust ports by w and the height of the opening of the port at the moment the transfer port begins to open by h, then with conventional timing the port capacity during the blowdown period is approximately 0.55 $wh\phi$ sq in.-deg, where ϕ is the crank angle corresponding to the blowdown period. The port capacity can be accurately calculated by means of the equation given on page 306, but this simple rule saves time.

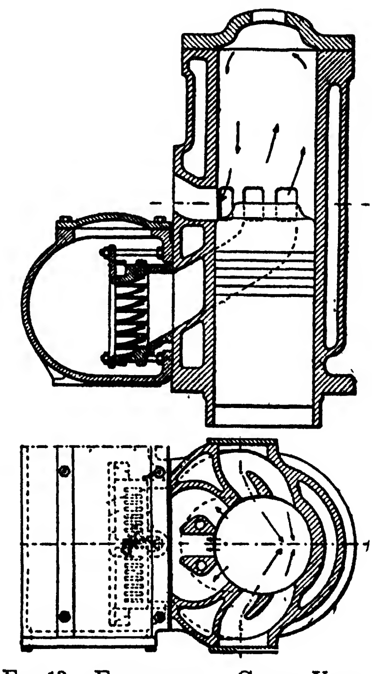


FIG. 13.—ENGINE WITH CHECK VALVE IN INLET PASSAGE.

The quantity of air or gas that must pass through the ports per cycle varies directly as the piston displacement. The quantity which actually flows through a port varies as the area of opening (measured at right angles to the direction of flow) and as the time of opening. The latter factor in turn varies directly as the period of opening in angular measure and inversely as the engine speed. Therefore, $D \sim A\phi/n$, where D is the piston displacement of the cylinder; A, the mean area of the port during the opening period; ϕ, the angle of port opening, and n, the engine speed in rpm. But $A\phi$ is the port capacity in sq in.-deg, hence

$$A\phi = \frac{Dn}{c} \text{ sq in.-deg.}$$

For high-speed engines (1200-1800 fpm piston speed) with unsymmetrical timing, the divisor c has a value of about 4500 for the blowdown period, 525 for the intake, and 575 for the total exhaust period. In these engines, therefore, the same as in most four-stroke engines, the intake-port capacity is greater than the exhaust-port capacity. The exact opposite holds in the case of two-stroke engines with symmetrical timing. Since the exhaust ports must be of materially greater depth than the inlet ports, the total time of opening of the exhaust ports is greater, and the maximum area of opening often also is greater, so that in some cases the area-time integral of the exhaust ports is about twice that of the inlet ports.

In making use of the above equation for the minimum port capacity required it must be remembered that to obtain substantially the full capacity, the edges of the ports must be rounded. Also, if the top of the piston is domed or cone-shaped, the port capacity is reduced in the proportion of unity to the cosine of the angle which the outer portion of the piston head makes with the plane of a flat head.

While it is customary to assume that the delivery is directly proportional to the area of the port and to the time of flow, owing to the inertia of air and gas there is undoubtedly a slight time lag. All motion starts with a velocity of zero, and during the opening period, while the opening area has reached a certain value, the instantaneous mean velocity of flow through that area undoubtedly is somewhat less than that corresponding to steady flow through the same area under the same pressure drop. The effect of this lag on the mean rate of flow throughout the period of port opening would be greater the higher the speed of the engine and the shorter, therefore, the total time of flow. This would have to be compensated for by increasing the area-time integrals (reducing the values of the divisor c) for really high-speed engines. No port-capacity data for Diesel engines of very high speed are available, but in a two-stroke gasoline racing engine designed for a top speed of 6500 rpm the value of c for the blowdown was 2600 and that for the exhaust period about 250.

The above simple equation will give a first approximation to the minimum port capacities required. However, there are additional factors which have an effect on the charging operation, such as the scavenging pressure, the delivery ratio, the temperature of the scavenging air, etc., and for accurate results these must be taken into account. The subject is covered in "Scavenging of Two-Stroke Cycle Diesel Engines" by Paul H. Schweitzer (The Macmillan Company, New York).

Crank Arrangements—Two-stroke multi-cylinder engines require crank arrangements different from those of corresponding four-stroke engines. In the latter, cranks at equal distances from the center generally are in line or in phase with each other. That results in phase equality of their respective inertia forces and eliminates bothersome rocking couples. With a similar crank arrangement in a two-stroke engine two cylinders would have to fire simultaneously, which would largely nullify the advantages of multi-cylinder construction.

In the selection of a crank arrangement the aims should be to obtain equal or nearly equal firing intervals, and to eliminate unbalanced inertia forces and rocking couples; and if that cannot be done, to reduce these forces and couples to a minimum. Another item that deserves consideration in this connection is that of main-bearing loads, which depend largely on the inertia forces on the reciprocating parts in the cylinders on opposite sides of the particular bearing. If the reciprocating parts in these cylinders are in phase, the bearing loads produced by their inertia forces add together, whereas if they are opposed in phase, they neutralize or cancel each other. In a two-stroke engine the reciprocating parts of adjacent cylinders always are out of phase, and the greater their phase difference the smaller will be the inertia load on the main bearing between them.

Shaking Forces and Rocking Couples—Firing intervals, primary and secondary unbalanced inertia forces (shaking forces), and primary and secondary rocking couples of two-stroke in-line engines with from one to ten cylinders are given in Fig. 14, and similar data for two-stroke V engines with from four to twenty cylinders in Fig. 15. The data given in each case apply only to engines designed to have a definite firing order (determined by the crankshaft design), which can be obtained from the diagram in the column at the left by reading off the cylinder numbers in the counterclockwise direction, starting with 1. In the chart for V-type engines the same procedure gives the firing order for the left bank of cylinders. Each cylinder in the right bank follows that of the same number in the left bank at one firing interval if the angle between cylinder banks is equal to the firing interval, and at three firing intervals if the angle between banks is equal to three times the firing interval. The symbols in the charts have the following significance:

C, centrifugal force in pounds produced if the reciprocating mass of one cylinder were rotated with crankshaft speed at crank radius.

W, centrifugal force produced by the counterweight necessary to cancel all or part of the inertia forces. The primary weight must turn at the same speed and in the same direction as the crankshaft, and its angular position is indicated in the diagrams by P.

H, horizontal unbalanced inertia remaining when the counterweight indicated is used.

V, vertical unbalanced inertia remaining when counterweight indicated is used.

ϕ, crank angle measured from the position indicated in the diagram.

a, distance in inches between cylinder axes.

M_v, rocking couple in pound-inches in the vertical plane. A positive couple turns clockwise in a side view with the forward end of the engine at the left.

M_h, rocking couple in pound-inches in the horizontal plane (perpendicular to the drawing). A positive couple turns clockwise in the top view.

λ, ratio of crank radius to connecting-rod center-to-center length.

All crankshafts are shown in front view in the column at the left. Rotation is supposed to be clockwise in the front view. The subscripts "v" and "h" refer to vertical and horizontal planes through the crankshaft axes.

Neutralizing Rocking Couples—Primary rocking couples can be neutralized by two pairs of unbalanced masses, one pair at each end of the engine, the two masses of each pair rotating in opposite directions at crankshaft speed. Fig. 16 is a diagram of such an arrangement. The masses must be driven positively from the crankshaft, and the gearing is preferably combined with that for the camshaft and accessories. In Fig. 16 A,A are the masses at the driving end, and are here shown as cast on the driving gears, while B,B are equivalent masses on the gear shafts at the opposite end of the engine. When the engine is running, each unbalanced mass is subjected to a centrifugal force in the direction from the axis of rotation through its center of gravity. This force

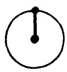
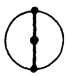
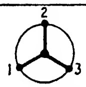
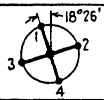
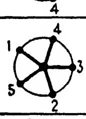
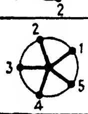
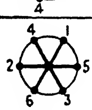
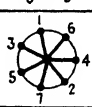
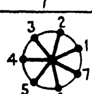
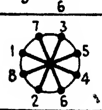
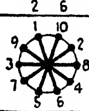
Inertia balance and firing intervals						
Cylinder arrangement	No. of cyl.	Primary		Secondary		Firing interval, deg.
		Shaking force	Rocking couple	Shaking force	Rocking couple	
Two-stroke in-line engines						
	1	$V = C\cos\theta$	$M = 0$	$V = \lambda C\cos 2\theta$	$M = 0$	360
	2	$V = 0$	$\vec{M}_V = a C\cos\theta$	$V = 2\lambda C\cos 2\theta$	$M = 0$	180
	3	$V = 0$	$M_V = 1.73 a C\sin\theta$	$V = 0$	$M_V = 1.73 a\lambda C\sin 2\theta$	120
	4	$V = 0$	$M_V = 3.16 a C\cos\theta$	$V = 0$	$M = 0$	90
	5	$V = 0$	$M_V = 0.45 a C\cos\theta$	$V = 0$	$M_V = 4.92 a\lambda C\sin 2\theta$	72
	5	$V = 0$	$M_V = 4.25 a C\cos\theta$	$V = 0$	$M_V = 0.41 a\lambda C\sin 2\theta$	72
	6	$V = 0$	$M_V = 3.46 a C\cos\theta$	$V = 0$	$M = 0$	60
	7	$V = 0$	$M_V = 3.59 a C\cos\theta$	$V = 0$	$M_V = 6.82 a\lambda C\sin 2\theta$	$51\frac{3}{7}$
	7	$V = 0$	$M_V = 8.07 a C\cos\theta$	$V = 0$	$M_V = 1.48 a\lambda C\sin 2\theta$	$51\frac{3}{7}$
	8	$V = 0$	$M_V = 0.45 a C\cos\theta$	$V = 0$	$M = 0$	45
	10	$V = 0$	$M = 0$	$V = 0$	$M_V = 0.9 a\lambda C\sin 2\theta$	36

FIG. 14.—CHART OF CRANK ARRANGEMENTS FOR TWO-STROKE IN-LINE ENGINES

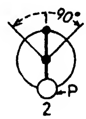
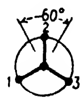
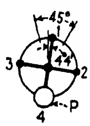
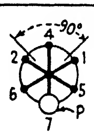
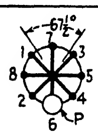
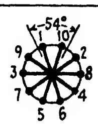
Inertia balance and firing intervals						
Cylinder arrangement	No. of cyl.	Primary		Secondary		Firing interval, deg.
		Shaking force	Rocking couple	Shaking force	Rocking couple	
Two-stroke V-type engines						
	4	W=C V=0 H=0	M=0	V=0 H= $0.283\lambda C$ $\times \sin 2\theta$	M=0	90
	6	W=0 V=0 H=0	$M_v =$ $2.6\,aC \sin\theta$ $M_h =$ $0.866\,aC\cos\theta$	V=0 H=0	$M_v =$ $2.6\,a\lambda C\sin 2\theta$ $M_h =$ $0.866\,a\lambda C\cos 2\theta$	60
	8	W=0.293C V=0 H=0	$M_v =$ $4.48\,aC\cos\theta$ $M_h = 0$	V=0 H=0	M=0	45
	12	W=C V=0 H=0	M=0	V=0 H=0	M=0	30
	16	W=0.616C V=0 H=0	$M_v =$ $0.345\,aC\cos\theta$ $M_h = 0$	V=0 H=0	M=0	$22\frac{1}{2}$
	20	W=0 V=0 H=0	M=0	V=0 H=0	$M_v =$ $0.943\,a\lambda C\sin 2\theta$ $M_h =$ $0.626\,a\lambda C\cos 2\theta$	18

FIG. 15.—CHART OF CRANK ARRANGEMENTS FOR TWO-STROKE V-TYPE ENGINES

ordinarily has both a horizontal and a vertical component. If the phase relations are as shown in the drawing, the horizontal components of the forces at each end cancel each other, while the vertical components add together. The result is a vertically reciprocating force at each end, and since the two reciprocating forces are 180 deg out of phase, they produce a rocking couple. This rocking couple must be so proportioned and so timed that it exactly neutralizes that due to the engine reciprocating parts. A similar mechanism can be used to neutralize a secondary rocking couple, but in that case the gears must be driven at twice crankshaft speed.

Primary rocking couples are easier to deal with than secondary, because the unbalanced masses required to neutralize the former must rotate at crankshaft speed instead of at twice that speed. In many cases it is possible, by the use of a

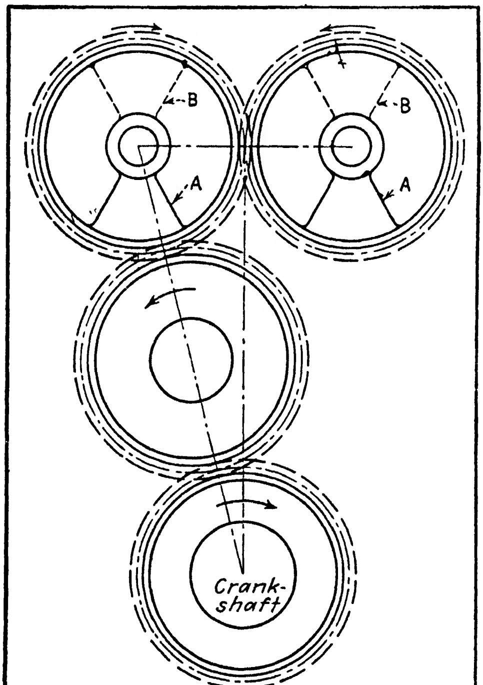


FIG. 16.—BALANCING MEANS FOR PRIMARY ROCKING COUPLE.

particular firing order, to either eliminate the secondary couple or to reduce it greatly, and if the primary couple is then neutralized by means of a mechanism of the type described, engine vibration due to this cause can be minimized.

Rotating Couple—In a multi-cylinder, two-stroke engine, unless all rotating parts are completely balanced by counterweights on the crank arms, there will also be a rotating couple. The centrifugal force on the rotating parts always is in line with the crank arms, and since in a six-cylinder engine, for instance, cranks *1* and *6* extend in opposite directions from the axis, the centrifugal forces on these cranks alone form a rotating couple. There are similar centrifugal forces on the rotating parts of the other four cranks, but their moment arms are shorter, and although the forces on cranks *2* and *3* and on cranks *4* and *5* are partly in opposition to those on cranks *1* and *6*, respectively, they reduce the moments due to the latter only moderately. For instance, in an engine with the firing order 1-4-2-6-3-5 the resultant rotating couple will be about 70 per cent of that produced by the centrifugal forces on cranks *1* and *6* alone, and will lag about 30 deg of crank motion behind it. The exact value and the phase angle of the resultant couple can be readily determined by trigonometry. This couple rotates at the same rate as the crankshaft, and therefore can be neutralized by providing one balancing weight at each end of the crankshaft, so proportioned and so located angularly as to produce a rotating couple that will exactly neutralize that due to the rotating parts of the crank trains. This method of eliminating a rotating couple is preferable to that of balancing the rotating parts of each crank train individually, as the latter method would call for the use of rather large counterweights on the crank arms. These might be difficult to accommodate in the crankcase and might make it necessary to lengthen the connecting rods to prevent interference of the counterweights with the pistons. In addition, they would lower the critical speeds of torsional vibration of the engine.

Five- and Seven-Throw Crankshafts—Five- and seven-cylinder engines have been produced by a number of manufacturers abroad to make it possible to offer a series of engines with a considerable range of output with a minimum investment in tooling, all engines of the series having cylinders of the same size. Alternate crank arrangements for each engine type are shown in Fig. 14. In each case the first, which produces the smallest primary and the largest secondary rocking couple, is preferable if no means to neutralize the rocking couples are to be provided, while the second is preferable if the primary rocking couple is to be neutralized and the secondary disregarded.

Six- and Eight-Throw Crankshafts—The crank arrangements for six- and eight-cylinder in-line engines shown in Fig. 14 produce no secondary rocking couples, and in the eight-cylinder the primary rocking couple is very small. The six-cylinder crank shown has the same firing order as the General Motors 6-71 engine, in which the primary rocking couple is neutralized by rotating unbalanced weights. In the General Motors 6-110 engine, which is designed for mobile, stationary and marine applications, where freedom from vibrating forces and couples is not so essential as in road vehicles, a different crank arrangement is used, giving a firing order 1-5-3-4-2-6. This introduces a small secondary rocking couple but reduces the primary rocking couple as compared with the firing order of the 6-71, and in this engine the couple-neutralizing mechanism is dispensed with.

An index of the relative magnitude of the rocking couple is obtained by dividing the coefficient in the expression for the couple by the number of engine cylinders. For the three-cylinder engine we get 0.577 for this index, while for the eight-cylinder we get 0.056, or less than one-tenth the former value.

Crank Arrangements for V-Type Engines—Of the V-type engines represented in Fig. 15, the four-cylinder has a certain appeal where a very compact powerplant is desired. Engines with this crank arrangement are being (or have been) manufactured by Graef & Stift in Austria and Krauss-Maffei in Germany. In such engines there are no rocking couples, but there is a small horizontal shaking force, which can generally be disregarded. Theoretically this could be neutralized by means of a pair of meshing unbalanced gears at crankshaft height and midlength, rotating at twice crankshaft speed; but the installation and drive of such gears involves difficulties. A German truck engine of this type (Krauss-Maffei) is without such balancing means.

Six-cylinder V engines are free from shaking forces, but have primary and secondary rocking couples in both the vertical and the horizontal plane through the crankshaft axis. Ordinarily the angle between cylinder banks would be 60 deg, as this gives uniform firing intervals. The strongest rocking couple is the primary in the vertical plane, and this can be readily neutralized by means of a mechanism producing an equivalent rocking couple of opposite phase. By reducing the angle between cylinder banks to 45 deg, to minimize the width of the engine, the horizontal rocking couples would be reduced by about 41 per cent, while the vertical couples would be increased about 11 per cent. However, with this angle of V the firing intervals are unequal—45 deg and 75 deg.

In the eight-, twelve-, and sixteen-cylinder engines the cranks are provided with counterweights of sufficient size to eliminate the horizontal rocking couples. These counterweights, which are in addition to those required to compensate for unbalanced rotating masses, also reduce the vertical rocking couples. The twenty-cylinder engine requires no counterweights and produces only a small secondary rocking couple. A twenty-cylinder, 2000-hp two-stroke engine having the firing order shown in Fig. 15 was produced during World War II by the Tokyo Engineering Works for the Japanese Navy, and was brought to this country after the war.

Brake Mean Effective Pressures—The full-load mean effective pressure obtainable with two-stroke engines varies within wide limits, depending on the method of scavenging and the degree of supercharge employed. In an engine with crankcase scavenging (which would not be considered for application to transportation units), it runs as low as 40 psi. If a separate blower is used, phenomenally high bmeps can be obtained by making the blower capacity large relative to the cylinder displacement, that is, by using a high supercharge ratio. But in practice a limit is set to the supercharge ratio and the quantity of fuel injected by overheating difficulties. With an increase in the specific power the rate of waste-heat generation rises, such working parts as pistons and piston rings reach higher temperatures, and lubrication and other difficulties are run into.

General Motors rates its Type 71 engine (with uniflow scavenging) on the basis of 106 psi at the maximum-torque speed of 1200 rpm; of 93 psi at the governed speed of 2000 rpm for intermittent load, and of 74 psi at the governed speed of 1800 rpm for continuous load. The three corresponding values of the bmep for the Type 110 engine (with cylinders 55 per cent larger) are 99, 92, and 75 psi. Rail-car engines built in Sweden by Atlas Diesel and employing the loop-scavenging system illustrated in Fig. 13, are said to develop a bmep of 73.5 psi at 1800 rpm. Engines built by the Manufacture d'Armes de Paris, of the double-piston type and similar to the Sulzer described further along in this chapter, except that a Roots-type blower is used instead of a piston-type scavenging pump, develop a bmep of 78 psi at 2000 rpm, the cylinder dimensions being 3.46-in. bore by 4.02-in. stroke of each piston.

General Motors Type 71 Engine—General Motors Corporation in 1938 placed on the market a line of two-stroke engines in a single cylinder size of 4¼-in. bore by 5-in. stroke (71 cu in. piston displacement). These engines are built with from two to six cylinders, and the four- and six-cylinder units are combined into eight- and twelve-cylinder "twins", twelve-cylinder "tandem twins," and twenty-four-cylinder "quads." Outputs of the series extend from 65 to 800 hp. The engine was designed primarily for use on trucks and

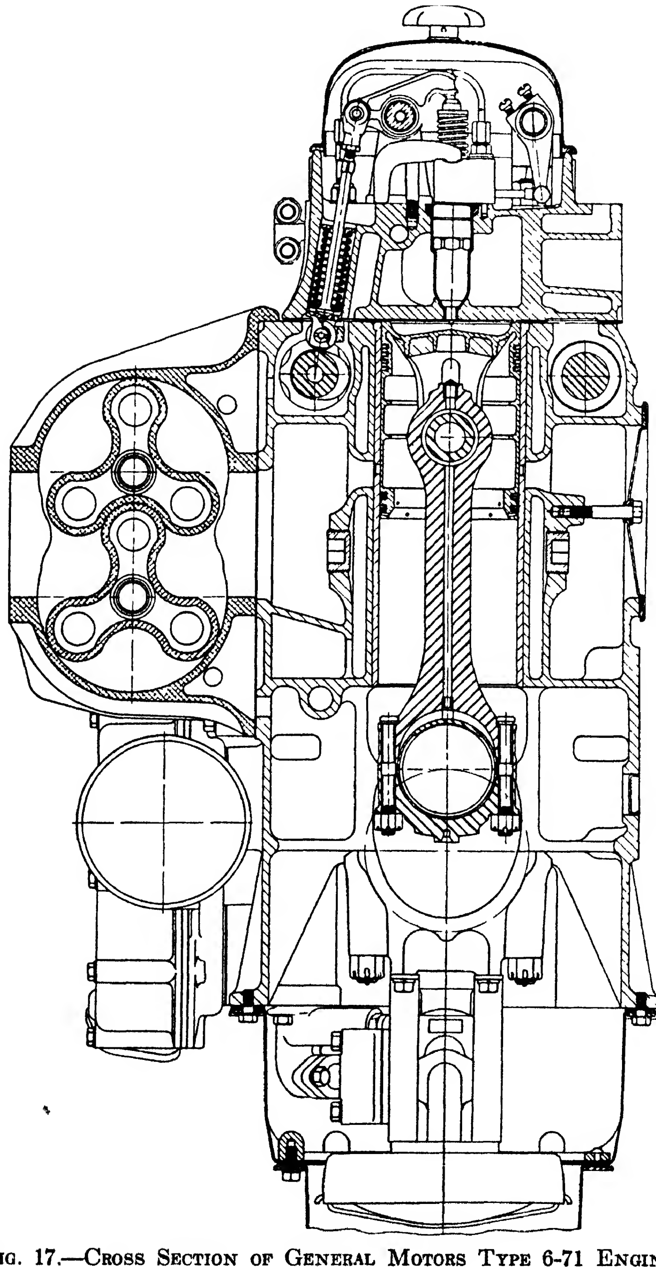


FIG. 17.—CROSS SECTION OF GENERAL MOTORS TYPE 6-71 ENGINE.

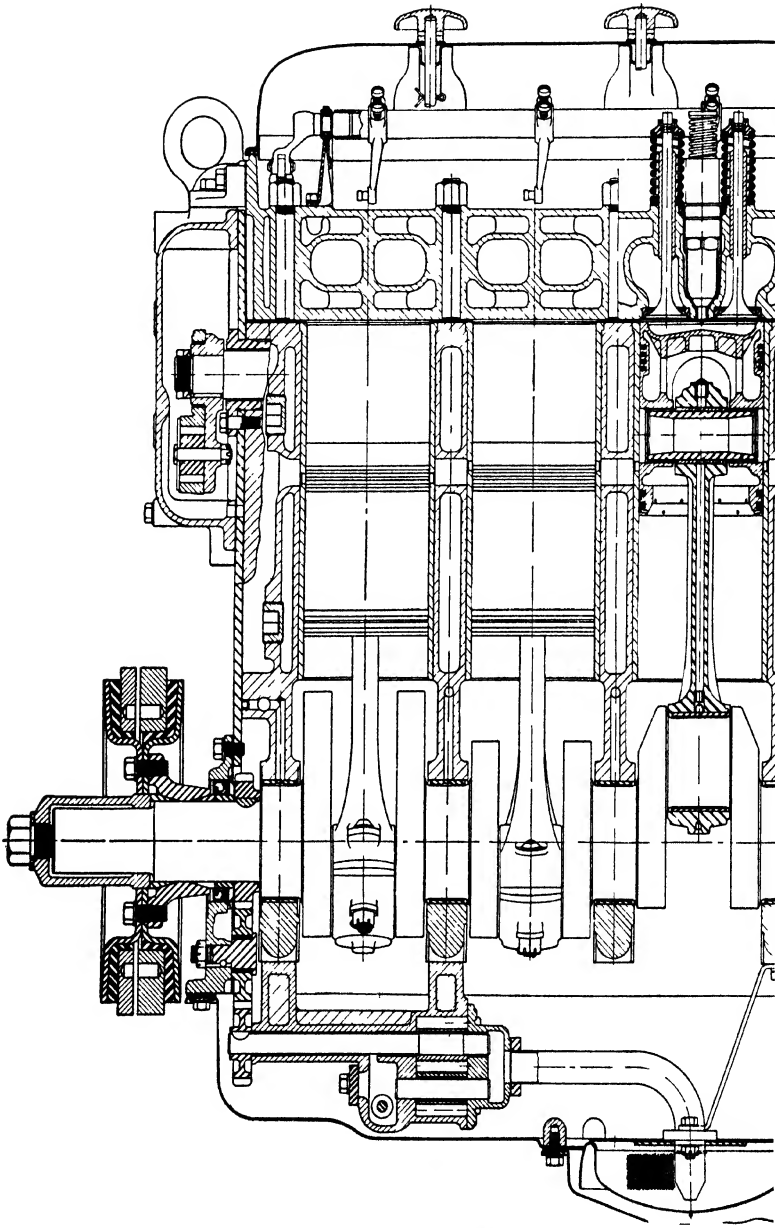


FIG. 18A.—LONGITUDINAL SECTION OF GENERAL MOTORS TYPE 6-71 ENGINE, FORWARD HALF

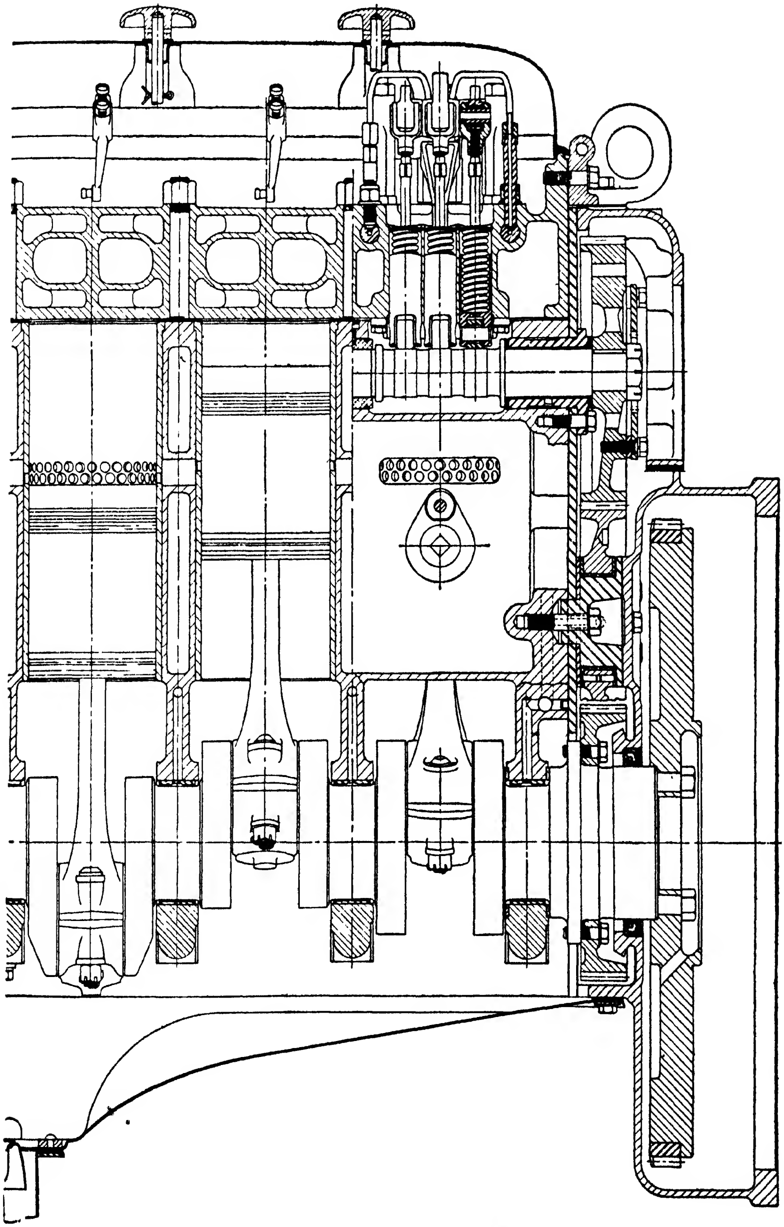


FIG. 18B.—LONGITUDINAL SECTION OF GENERAL MOTORS TYPE 6-71 ENGINE, REAR HALF.

buses, but application to road vehicles naturally is confined chiefly to the three-, four- and six-cylinder models.

These engines are scavenged by Roots-type blowers with two three-lobed helical rotors each. The cylinders are surrounded by a large air chamber, and the side walls of the block are practically plane and extend up vertically from the bottom flange, to which the pressed-steel oil sump is secured. Air enters the cylinders through ports in the cylinder wall at the bottom of the stroke, and exhaust takes place through dual exhaust valves in the cylinder heads, so that the engines operate on the uniflow principle. A combined injection pump and injection valve (unit injector) is located in the center of each cylinder head, the plunger of the injection unit being actuated from the camshaft, which extends through a tunnel running the length of the engine block near its top at the side opposite the blower. From the camshaft all of the exhaust valves and the injection units are operated. This being a two-stroke engine, the camshaft turns at crankshaft speed. It is driven from the crankshaft through spur gears located at the flywheel end of the engine. Extending parallel with the camshaft on the opposite side of the engine block is another shaft geared to the camshaft to turn in the opposite direction at the same speed. Both shafts carry balancing weights at opposite ends; these weights produce a rocking couple equal to and opposite in phase to the primary rocking couple due to the reciprocating parts of the engine, and therefore neutralize it.

From the sectional views of the six-cylinder engine, Figs. 17 and 18, it will be seen that the crankshaft is of very rigid design, its main bearings having a diameter of 3½ in. and its crankpins of 2¾ in. Pistons are of cast iron and carry four compression rings above the piston pin and two oil rings below it. Special interest attaches to the provisions made for cooling the pistons. It will be seen that there is a considerable distance between the top of the piston and the top ring, and there the wall of the piston is made quite thin, so it will act as a "heat dam." This is made possible by transmitting the pressure from the crown to the piston bosses through pedestals instead of through the piston skirt. Connecting rods are rifle-drilled and have a nozzle pressed into the top, from which jets of oil are thrown against the ribbed under side of the piston crown to carry off excess heat. The cylinders are provided with "dry" liners. The air chamber surrounding the cylinders is cast open at both ends, the openings being closed by steel plates.

All of the three automotive models (three-, four-, and six-cylinder) are governed at 2100 rpm for truck use. The compression ratio is 16, the maximum torque is developed at between 800 and 1000 rpm, and the fuel consumption is given as 0.45 lb per bhp-hr. Performance data of the three engines are given in the following table:

TABULATED DATA OF G.M. AUTOMOTIVE ENGINES

Model	*3-71*	*4-71*	*6-71*
Displacement, cu in.......	212.8	283.7	425.6
Maximum horse power * . .	105	145	220
Maximum torque, lb-ft....	300	400	600
Dry weight, lb †..........	1110	1245	1600

* At 90 F, 28.33 in. mercury column, 1500 ft altitude.
† Complete with starter and generator but without fan, transmission, and clutch.

For industrial use the two-, three-, four-, and six-cylinder engines are governed at from 1200 to 1800 rpm. At 1600 rpm they have "continuous" ratings of 45, 67, 89, and 134 hp. At 2100 rpm the friction mep is 38 psi, one-half of this being accounted for by the power required to drive the blower, the other half by internal friction. This makes the mechanical efficiency under full load approximately 72 per cent. At 2100 rpm the scavenging pressure is 6.5 psi. This pressure varies nearly as the square of the speed and is only 1.5 psi at 1000 rpm. Weights and performance data of the three blowers used for the automotive engines are given in the following table:

TABULATED DATA OF BLOWERS FOR G.M. AUTOMOTIVE ENGINES

Blower Output		*Rated Engine Output*		*Blower Weight, Lb*	*Blower Specific Weight, Lb per bhp*
Cfm	*Rpm*	*Bhp*	*Rpm*		
700	4200	220	2100	62.9	0.286
466	4200	145	2100	53.2	0.367
350	4200	105	2100	48.5	0.462

With the engine running at 2100 rpm the blower speed is 4200 rpm. Throughout the normal speed range the blower delivers free air at the rate of about 135 per cent of the piston displacement. A conventional four-stroke Diesel engine with atmospheric induction takes in about 85 per cent of its displacement volume of free air under normal load, hence about 60 per cent more air passes through a cylinder of the Type

71 two-stroke engine per cycle as through an equal-sized cylinder of a four-stroke engine. A considerable fraction of the air in the two-stroke engine is used for scavenging and passes out through the exhaust valves before compression really begins, the valves being open throughout the period the inlet ports are open.

Injectors Adapted to Type of Service—Engines intended for "continuous" service with relatively constant load and speed are equipped with what is known as the "60" injector (combined injection pump and fuel nozzle); while those intended for "intermittent" service with varying load and speed are equipped with either the "60", "70" or "80" injector. The pumps of these injectors have displacements of 60, 70 and 80 cubic millimeters, respectively. Nozzles of the smaller injectors have six; those of the larger ones have seven orifices.

The smaller injectors are used on engines for work boats and most industrial applications, while the larger ones go into engines for pleasure craft, certain industrial uses, and vehicle applications where the ability to materially increase the power for short periods is of importance. The "60" injector gives an overload capacity of 10 to 15 per cent and protects both the attached machinery and the engine from damage, thereby insuring economy and long life under any load conditions. With the "80" injector, the output can be stepped up to as much as 33 per cent above the "continuous" rating, at a certain sacrifice in economy. Users of the engines are cautioned, however, that the average load over a 24-hour period must not exceed the "continuous" rating. Owing to the large capacity of the blower, there is always a large excess of air in the cylinders, and the additional power can be obtained by merely supplying additional fuel. This large overload capacity is claimed to be a unique feature of the two-stroke engine.

Timing of the engine varies slightly with the injector equipment. With "60" injectors the exhaust valves, when having the specified "hot" lash of 0.009 in., begin to lift 86° ahead of bottom center and close 64° past bottom center. Intake ports open 46° ahead of and close 46° past bottom center. Exhaust cams have a 10-deg ramp at the beginning and a 16-deg ramp at the end of the lifting lobe, and the exhaust period increases as the engine heats up and the lash is reduced. With "60" injectors injection begins 12° ahead of top center, theoretically, and under full load ends 2° ahead of top center. The fuel-injection system of the engine was described in Chapter VII.

Adaptability of Engine—In designing the G.M. two-stroke engine, the engineers endeavored to make it possible to adapt it for truck, bus, marine and industrial installations with the least possible changes, so that most of the parts could be produced on a true quantity basis. It is held that the two-stroke, uniflow engine with unit injector lends itself particularly to this program. The engine block is symmetrical in design, which permits of turning it end for end and placing all of the accessories on the opposite side. Cylinder-head studs are distributed symmetrically around the cylinder, so that the head can be placed on the block to locate the exhaust manifold on either the right or left side, regardless of the location of the blower. There is an idler between the crankshaft gear and the camshaft gear, and the direction of rotation of the engine can be reversed by merely placing this idler gear between the crankshaft gear and the gear on the shaft running parallel with the camshaft at the top of the engine. The crankshaft then turns in the opposite direction, while all other shafts turn in the same direction as before. When the cylinder heads are reversed to change the location of the exhaust manifold, the camshaft and balancer shaft are interchanged in position.

Multiple Engine Units—The adaptability of the Series 71 design, referred to in the foregoing, is exemplified by the multiple units in which this engine is being offered, the "Twin 6" and the "Quad 6." The Twin 6 consists of two six-cylinder engines mounted side-by-side and driving through dry-disc clutches and helical gears to a single drive shaft. More than 20,000 of these units were used in tanks and tank destroyers in World War II. In a slightly modified form and fitted with a base, radiators and controls, this unit is being supplied for oil-well drilling and other purposes. The Quad 6 consists of four 6-71 engines mounted together on a common base and delivering their power to a single drive shaft. The gear box is located in the center and a power take-off shaft extends between one pair of engines. Each engine is provided with a friction clutch and one or more engines can be declutched. More than 10,000 marine "Quads" were built for the U. S. Navy for use in invasion boats.

After the Series 71 had been in production for about a decade, a number of improvements were made in the design. Ribs in the cylinder head and block were enlarged and increased in number, to make the castings more rigid. The holes for the cylinder-head studs were drilled and tapped ⅝ in. deeper, and counterbored, to prevent stress concentration at the point where the studs emerge from the block. A new method of cylinder-head packing was introduced, which is illustrated and described in Chapter XV. To prevent "brinelling" of block material under the flange at the upper end of the cylinder liners, the block is now counterbored and has rings of heat-treated material inserted under the flanges. The angle of the exhaust-valve seat has been reduced from 45 to 30 deg, to ease the flow of the exhaust. These improvements are not shown in the drawings reproduced herewith.

General Motors Series 110 Engine—In addition to the Series 71 the Detroit Diesel Engine Division of General Motors Corporation produces a Series 110, a six-cylinder of 5-in. bore by 5.6-in. stroke (110 cu in.). It was designed to meet a demand for a single unit of greater horse power than the 6-71 in vehicular, marine, and stationary applications. Design features are similar to those of the Series 71, with the following principal exceptions: Instead of a Roots blower, a centrifugal blower is used for scavenging purposes. The latter is well adapted to applications—such as propeller work—where the torque load increases rapidly with engine speed. The blower is mounted in back of the engine block over the flywheel housing, thus making the engine more compact in the transverse direction. A different firing order is used (1-5-3-4-2-6 instead of 1-4-2-6-3-5). This eliminates the primary rocking couple and obviates the need for a couple-neutralizing mechanism. Counterweights are provided on four of the twelve crank arms. Cranks on opposite sides of intermediate bearings other than the center bearing are spaced 120°, hence the inertia forces on their reciprocating parts cancel out to a certain extent, and the load on the bearing is reduced. A torsion damper of the viscous-fluid type is fitted. For industrial purposes this engine (as well as the Series 71) can be fitted with an integral hydraulic torque converter, which has torque-load characteristics similar to those of a propeller.

Junkers Double-Piston Engine—Diesel engines of the double-piston type have long been built by the Junkers Engine Works of Dessau, Germany. Sectional views of a two-cylinder truck engine of early design are shown in Fig. 19. There is a six-throw crankshaft, to which each of the two lower pistons is connected directly by the conventional rod, while each upper piston is connected to it through a balance lever and a pair of long connecting rods, one on each side of the cylinder. The upper piston forms a rigid unit with the piston of the scavenging pump.

Transfer ports are located at the upper end of the cylinder and are controlled by the upper piston, while exhaust ports are at the lower end and controlled by the lower piston. The cranks for the upper pistons are set 15° out of phase with those for the lower, so that while the exhaust ports open ahead of the inlet ports, they also close earlier.

The scavenging pump, of the reciprocating type, is located at the top, and the whole interior, including the crankcase, serves as an equalizing chamber to maintain a substantially

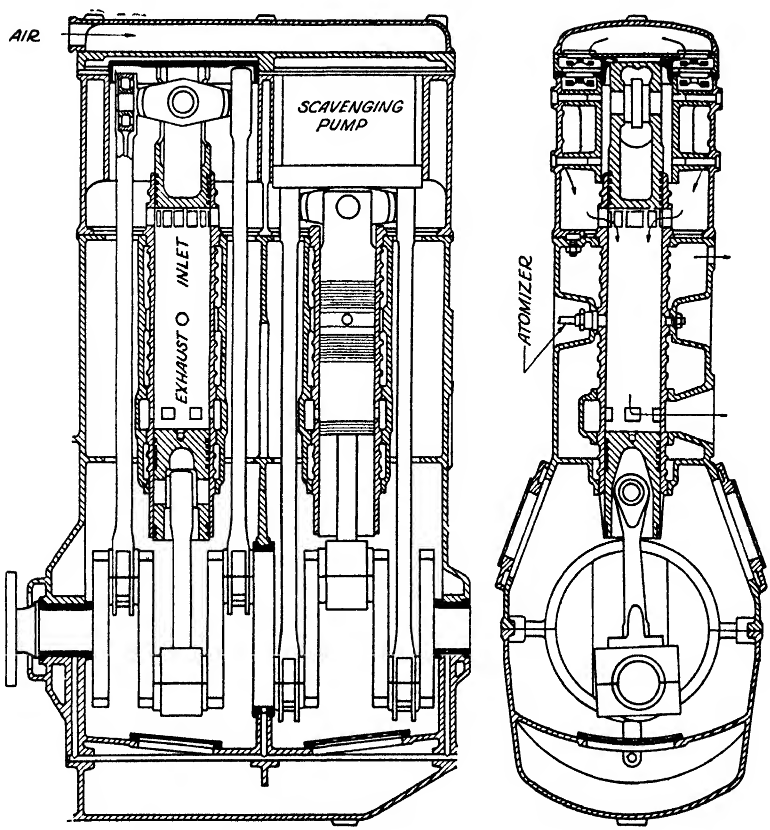


FIG. 19.—JUNKERS OPPOSED-PISTON-TYPE ENGINE.

constant pressure at the inlet ports. To compensate for the greater weight of the upper reciprocating parts, their stroke is made shorter than that of the lower ones. Tests carried out on a single-cylinder engine of 2.36-in. bore and a total stroke of 8.28 in. (3.54 in. upper and 4.74 in. lower) showed that it developed a bmep of 90.7 psi at 1000 rpm, with a specific consumption of 0.47 lb per bhp-hr. The compression ratio of the engine was 15.8, based on the effective length of stroke, which was 14 per cent less than the actual combined strokes. Timing of this engine was as follows: Exhaust opened 54° ahead of and closed 54° after bottom center; inlet opened 36° ahead of and closed 66° after bottom center.

One disadvantage of certain direct-injection Diesel engines is that the compression chamber is a cylinder of very small depth, with a very large surface in comparison with its volume, which results in rapid heat loss at the beginning of the power stroke. In the double-piston engine, owing to the great stroke/bore ratio, the compression space is not nearly so shallow as in the conventional Diesel, and, besides, the greater part of the surface enclosing it is furnished by the crowns of the two pistons, which are not water-cooled. These advantages are offset by the disadvantages of the engine's unwieldy form (abnormal height), of the heavy reciprocating parts, and of the difficulty of cooling the pistons in really high-speed operation. At the moment of ignition, about two-thirds of the combustion-chamber wall is constituted by the piston crowns, and a large percentage of the waste heat therefore must go into the pistons.

Junkers later also built aircraft Diesel engines of the double-piston type, but in these there was a crankshaft at both top and bottom, the two shafts being connected by a train of spur gears and the propeller mounted on the shaft of the topmost intermediate gear.

A modern American engine of the double-piston type is described and illustrated in Chapter XIII.

Scavenging by Gas Inertia.—It has long been known that when the exhaust valve or exhaust port of an engine is opened rapidly, the elastic force of the suddenly-released gas sets up a wave motion in the exhaust line, which is sometimes referred to as a pipe-organ effect. The cylinder contents at first rush into the exhaust pipe at great velocity, and the inertia of the gaseous column creates a momentary suction effect, so that the pressure in the cylinder drops below atmospheric. The vacuum thus created tends to fill up again, and a moment later a reverse flow sets in. The pressure wave has a definite period, which depends on the length of the exhaust line. If the air port is opened while the pressure is below atmospheric, air will be drawn into the cylinder, and a definite scavenging effect will be obtained without the use of a pump. The period of the wave motion set up by the exhaust is independent of engine speed, for which reason this effect can be taken advantage of for scavenging purposes over a narrow speed range only. There is no inertia-scavenging effect when the engine is being started, and one solution of the resulting problem consists in providing the engine with a manually-controlled valve which places the air-inlet manifold in communication with the crankcase, thus providing crankcase scavenging for starting. When the valve is thrown to the running position, it opens the air-inlet passage to the atmosphere.

Kadenacy Engine—Work on engines with inertia scavenging has been done by Michel Kadenacy, a French engineer, in collaboration with the Armstrong-Whitworth Securities Company, Ltd., in England. The Kadenacy engine has been built in single-cylinder and six-cylinder models, both for stationary installation. The single-cylinder model, of which a sectional view (from *Engineering*) is shown in Fig. 20, has no scavenging pump, but the six-cylinder model is fitted with a Roots-type blower bolted directly to the side of the cylinder block and delivering into an air chamber surrounding the lower portion of the cylinders.

This engine is of the "uniflow" type, having air ports at the bottom of the stroke and an exhaust valve in the center of the cylinder head. A peculiarity of the design is the shape of the combustion chamber, which is formed between a cup-shaped depression in the piston crown and the slightly-dished exhaust valve. Fuel is injected into this compact chamber by a nearly horizontal injector mounted in the cylinder head, through a channel formed in the rim portion of the piston crown.

As shown in Fig. 21, the air ports extend all around the cylinder and are cut tangentially, so that they will impart a swirling motion to the incoming air. Ordinarily the air is drawn in through two "straight-through" silencers mounted vertically at the side of the crankcase, and through the manually-controlled valve, which is shown in the "running" position in the drawing. When this valve is turned through an angle of 90 deg, it cuts off direct communication with the atmosphere and opens the air-inlet passage to the crankcase. There is an automatic check valve in the end of the "switch-over" valve, which admits air to the crankcase during the up-stroke of the piston.

Published test results from the single-cylinder engine ($2\frac{7}{8}$-in. bore, $4\frac{1}{8}$-in. stroke, 26.6 cu in. displacement) show that it developed 5 hp at 950 rpm, which corresponds to a bmep of 78 psi. At this output the specific fuel consumption was 0.405 lb per hp-hr, and the exhaust temperature, 480 F.

Burmeister & Wain Engine—The firm of Burmeister & Wain, Copenhagen, Denmark, well known for its larger Diesel marine engines, has in recent years built light, moderate-speed two-stroke engines for rail-car work. A transverse section of this engine is shown in Fig. 22. In addition to the power piston, each cylinder contains a piston valve coaxial

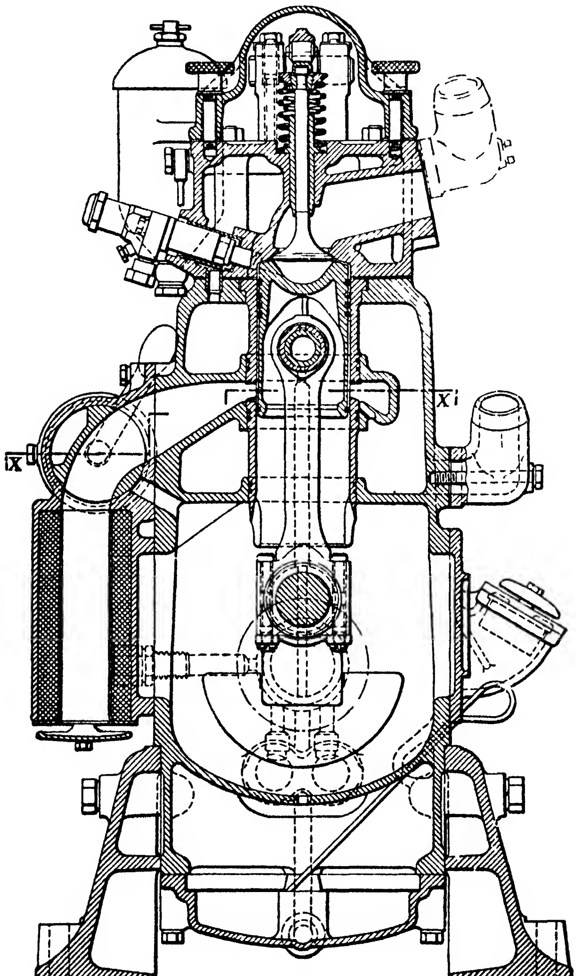


Fig. 20.—Cross Section of Kadenacy Single-Cylinder Engine.

with and above the piston. Scavenging and supercharging are effected by means of a Roots-type blower with four-lobed rotors. Scavenging air enters the cylinder through ports in the cylinder wall, which are uncovered by the piston when approaching the lower end of its stroke. These ports extend completely around the cylinder. Exhaust takes place through ports in an upward extension of the cylinder, of reduced diameter, which ports are masked and unmasked by the piston valve. As air enters the cylinder at the bottom and gases of combustion leave at the top, the engine operates on the uniflow principle. The piston valves are operated from the crankshaft through the intermediary of eccentrics, eccentric straps, crossheads, and pullrods. Piston and piston valve move sub-

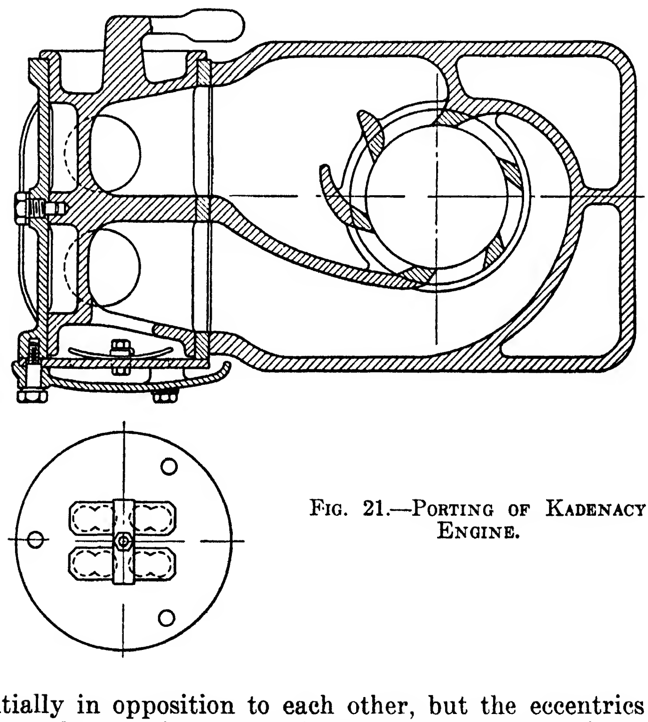


Fig. 21.—Porting of Kadenacy Engine.

stantially in opposition to each other, but the eccentrics are not exactly opposite the cranks, so that an asymmetric timing diagram is obtained.

In a general way the engine works on the same principle as the Junkers opposed-piston type. The piston valve also acts as a piston, as it moves most of the time in the direction in which it is being urged by the gas pressure, but the work done on it by the expanding gases is only about 10 per cent that done on the piston, the area exposed by it to gas pressure being 25 per cent and its stroke 40 per cent that of the piston.

Rocking-Beam Engines—In another type of double-piston engine the cylinders are horizontal and the two pistons in each cylinder act on a crankshaft located centrally below the cylinder or cylinder block, through the intermediary of

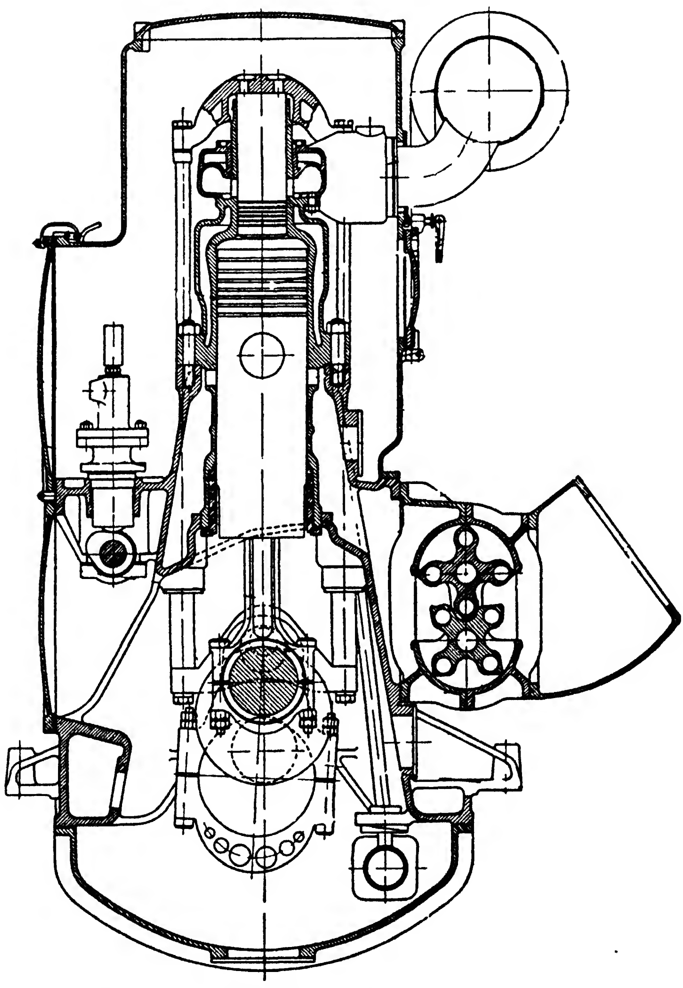


FIG. 22.—BURMEISTER & WAIN TWO-STROKE ENGINE IN CROSS SECTION.

two pairs of connecting rods and a pair of rocking levers. Diesel engines of this type were built in this country for a number of years by the Hill Diesel Engine Company of Lansing, Mich., but the design was abandoned again. It seems that the problem of lubrication under high-speed conditions gave rise to some difficulty. Such engines are now being built by Sulzer Brothers in Switzerland for motor trucks, farm tractors, and stationary power units. A sectional view of one of these engines is shown in Fig. 23. With this engine, an asymmetric timing diagram is obtained by

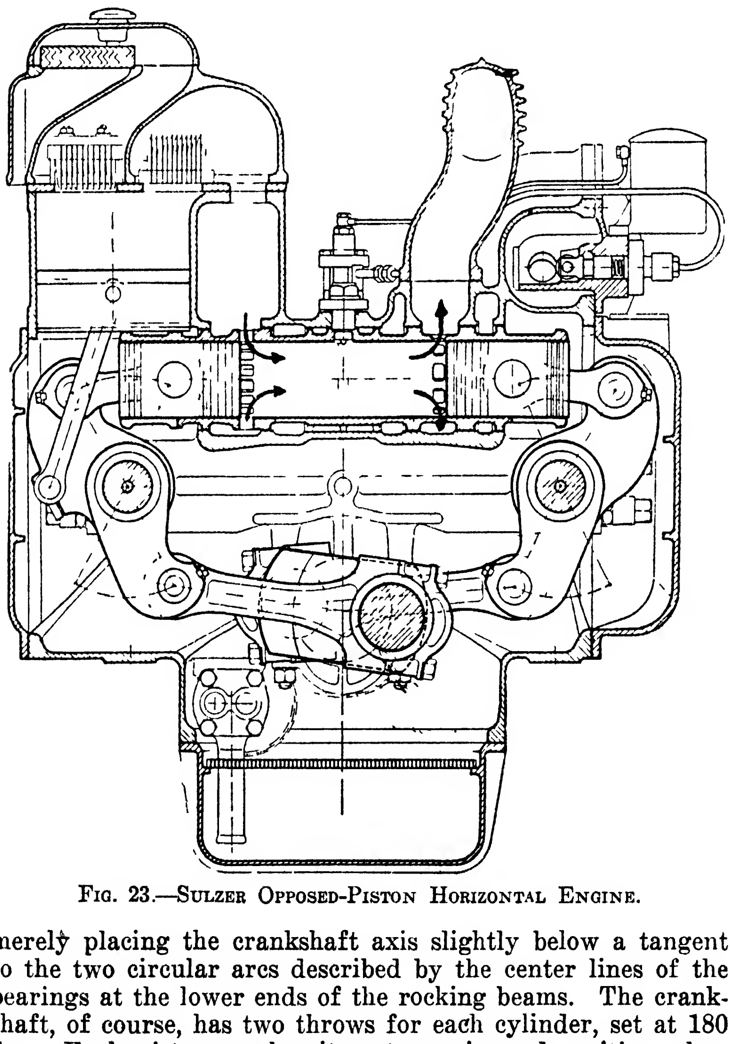


FIG. 23.—SULZER OPPOSED-PISTON HORIZONTAL ENGINE.

merely placing the crankshaft axis slightly below a tangent to the two circular arcs described by the center lines of the bearings at the lower ends of the rocking beams. The crankshaft, of course, has two throws for each cylinder, set at 180 deg. Each piston reaches its extreme inward position when its crank and the attached connecting rod come in line with each other, which occurs slightly before the crank reaches the horizontal position, for one piston, and slightly after it has passed through that position, for the other.

Sulzer rocking-beam engines are made in two cylinder sizes of 2¾ by 4 and 3½ by 4¾ in., and in two-, three-, and four-cylinder models, the small cylinder, in addition, being used in a single-cylinder model. Each cylinder is provided with a separate piston-type blower, driven from the rocking beam at the scavenging end. As in the double-piston vertical engine, scavenging air enters the cylinder through ports uncovered by one of the pistons, and exhaust takes place through ports controlled by the other. These engines are designed to be operated at a maximum speed of 1500 rpm in automotive use, and at 1000 rpm in heavy-duty service. The compression ratio is 22:1 and the maximum combustion pressure 1150 psi. Engines are rated on the basis of 84 psi bmep. Weights of these engines are comparatively high. For instance, the two-cylinder 2¾ by 4-in., which is used for tractor purposes and develops a maximum of 30 hp at 1500 rpm, weighs 1000 lb.

CHAPTER XIII

Railroad Engines

As pointed out in Chapter I, in the railroad field there are four distinct applications for Diesel engines—to switching locomotives, rail cars, streamlined articulated trains, and main-line locomotives. In the United States there has been little demand for rail cars since the early thirties. Articulated trains also did not come into extensive use, because with their limited passenger capacities they did not meet the requirements of main-line roads, and at present the use of Diesel engines in the railroad field is confined almost entirely to locomotives—switching, passenger, and freight.

Switching Locomotives—On American railroads Diesel engines so far have been used most extensively for switching locomotives. The first locomotive of this type in this country was built by the American Locomotive Company and was placed in service in 1925. It was equipped with an engine built by the Ingersoll-Rand Company, while the electrical equipment was furnished by General Electric Company. The four-stroke engine developed 300 hp at 600 rpm and weighed about 63 lb per hp. It may be pointed out in this connection that very light construction is not necessary for switching locomotives. These latter must have a certain total weight in order to be able to produce the traction effort wanted, and there seems to be no good reason why a good percentage of this weight should not be put into the engine.

The power required by switching locomotives varies considerably. In Europe the rolling stock is generally much lighter than in this country, and the switching locomotives therefore also can be lighter. Thus the London, Midland and Scottish Railway has in service a number of "shunting" locomotives of only 150 hp rating. These locomotives have a fluid flywheel (hydraulic coupling) and a two-speed planetary gear-set giving speeds of 4.5 and 9 mph at the governed engine speed of 900 rpm. The final drive is from a cross shaft with crank directly coupled to a crankpin on one of the wheels, the other two wheels being connected to this one in the usual manner. The locomotive develops tractive efforts of 12,000 lb and 6000 lb at the two speeds respectively.

Most of the Diesel switching locomotives delivered to American railroads in recent years are much heavier and more powerful. Being equipped with the flexible electric drive, they can be used also for light branch-line service. During World War II four different sizes of Diesel-electric locomotives were developed for the U. S. Army, of 25, 44, 65 and 127 tons, equipped with engines of 150, 380, 500 and 1000 hp, respectively, and geared for maximum speeds of 20, 35, 45 and 60 mph, respectively. The Army preferred Diesel-electric locomotives in its overseas operations because they require no large amounts of water, are instantly available after delivery, use a fuel that can be easily procured and transported, and can be operated by one man. Other advantages that count heavily in military operations are that these locomotives can be maintained without extensive repair facilities, are equally satisfactory in switching and road work, produce no smoke and fire-box glare at night, and are always instantly ready, which is of special importance in emergency operations. Some of these Army locomotives were used for the transportation of war equipment over the Trans-Iranian Railroad from the Persian Gulf to Russia. This line leads through a desert where water is unobtainable and where steam locomotives therefore are impractical.

At the beginning of 1952 a number of the leading railroads of the country had been almost completely "dieselized," and others were rapidly approaching this condition. Consequently, there was a strong demand for main-line Diesel locomotives as well as for switchers. Among new models introduced the previous year were three six-axled, six-motored (electric) road switchers. Locomotive-type Diesel engines were being produced with up to 16 cylinders, their outputs ranging up to 2500 hp. In some cases two engines were installed in a single cab.

Some European Developments—In Europe the application of Diesel engines to railway rolling stock previous to World War II was confined almost entirely to rail cars and switchers. There the private automobile had not cut into railway passenger traffic to the same degree as in this country, and the Diesel-powered rail car promised a solution of the problem of providing low-cost frequent services in congested areas. Among the more ambitious projects of that era was one by the German State Railways to install 300 Diesel rail cars on lines in the industrial Ruhr district. Owing to the outbreak of war this plan was not fully carried out at the time. Toward the end of the "forties" these railways again turned their attention to the Diesel for a solution of their problems, and three of the leading manufacturers of Diesel engines in Germany (Daimler-Benz, M.A.N., and Maybach) brought out railroad engines of new design to compete for this market. These engines, with ratings up to 1200 hp, are still generally referred to as "rail-car engines," although the larger ones would seem to be well adapted to locomotives. They are exceptionally light for their outputs, the Maybach, for instance, weighing only 6.7 lb per bhp on the basis of its factory rating. This low specific weight is made possible by the extensive use of light alloys, by supercharging, and by operating at relatively high speeds. The designers hope to approach the engine life (period between complete overhauls) of American engines of much greater specific weights. The argument that in locomotives weight is required for traction they meet with the observation that "if it is weight you need, ballast is much cheaper than high-grade machinery."

Alco Engines—American Locomotive Company in 1952 had in production a six-cylinder in-line engine of 12½-in. bore by 13-in. stroke, and twelve- and sixteen-cylinder V-type engines of 9-in. bore by 10½-in. stroke. The six-cylinder engine delivers 660 hp at 740 rpm with atmospheric induction, and 1000 hp at the same speed when turbosupercharged. Both of the V engines are equipped with turbosuperchargers, and deliver 1600 and 2250 hp at 1000 rpm, respectively. The in-line engines are installed in yard- and road-switching locomotives; the 12-cylinder V-type engines, in road switchers and freight-passenger locomotives, and the 16-cylinder, in passenger locomotives.

The vertical engine is of conventional design, having a cast-iron cylinder block with "wet" liners, a cast-iron crankcase to which the cylinder block is bolted, and individual cylinder heads secured to the block by heavy studs. There are four valves in each head, the injection nozzle being located centrally between them. Each cylinder is served by an individual injection pump. Engine speed is controlled by a Woodward governor. This engine weighs 33,700 lb without and 34,800 lb with the turbosupercharger.

The V engines are of more recent design and comprise welded-steel structural members, including a base, a free-end casing, and a cylinder block. The angle of the V block is 45 deg in both the twelve- and the sixteen-cylinder models. A photograph of the twelve-cylinder engine is shown in Fig. 1, and Fig. 2 shows the base and the free-end casing. The base provides mounting surfaces for the free-end casing, the generator adapter, and the cylinder block, and is equipped with two engine mounting pads, two additional pads being provided on the generator adapter. It forms the lower half of the crankcase and acts as an oil reservoir. Main and connecting-rod bearings, oil lines, cylinder liners, and piston skirts can be inspected through large openings in each side by removing the

FIG 1—ALCO TWELVE-CYLINDER SUPERCHARGED 1600-HP ENGINE.

crankcase covers. The free-end casing houses the viscous-fluid-type vibration damper and the pump gearing, and provides mounting surfaces for the water and oil pumps, the turbosupercharger support, and the fuel oil filters.

The cylinder block (Fig. 3) supports the crankshaft, rods, pistons, water jackets and liners, and cylinder heads. It also provides mounting surfaces for the turbosupercharger support, the inlet water header, the camshaft housing, and the generator adapter. The upper halves of the main-bearing saddles and a portion of the intake manifold are integral parts of the block.

Each cylinder unit is made up of two machined cylindrical shells, the water jacket and the cylinder liner. The liner extends above the water jacket at the top and is provided with a recessed rim which rests on the flanged top of the water jacket, a water-tight joint between the two being effected by means of a rubber ring gasket in a groove in the jacket. At the bottom the liner is a close fit in the jacket, the joint there being sealed by two ring gaskets. The top flange of the water

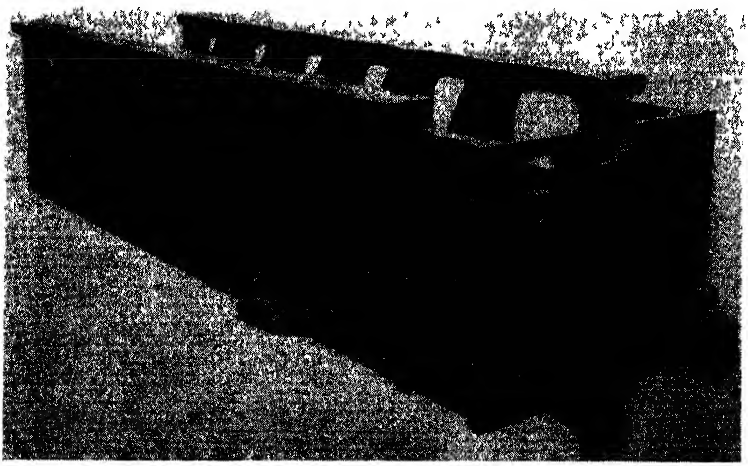


Fig 2—Base and Free-End Casing of Alco Engine.

jacket rests on the top surface of the cylinder block, the liner and jacket together with the cylinder head being held in place by means of the cylinder-head studs. Cylinder units with their pistons and connecting rods can be removed from the top of the block as shown in Fig. 4.

A system of positive-flow oil cooling is provided for the pistons. Referring to Fig. 5, a separate ring carrier is shrunk on the upper part of the piston, and oil-cooling grooves are machined in the piston before application of the ring carrier. Oil from the engine lubricating system is forced up the drilled connecting rod, through the hollow piston pin and radial holes in the wall of the pin, and through the hole in the piston parallel to its axis, into the lowest cooling groove.

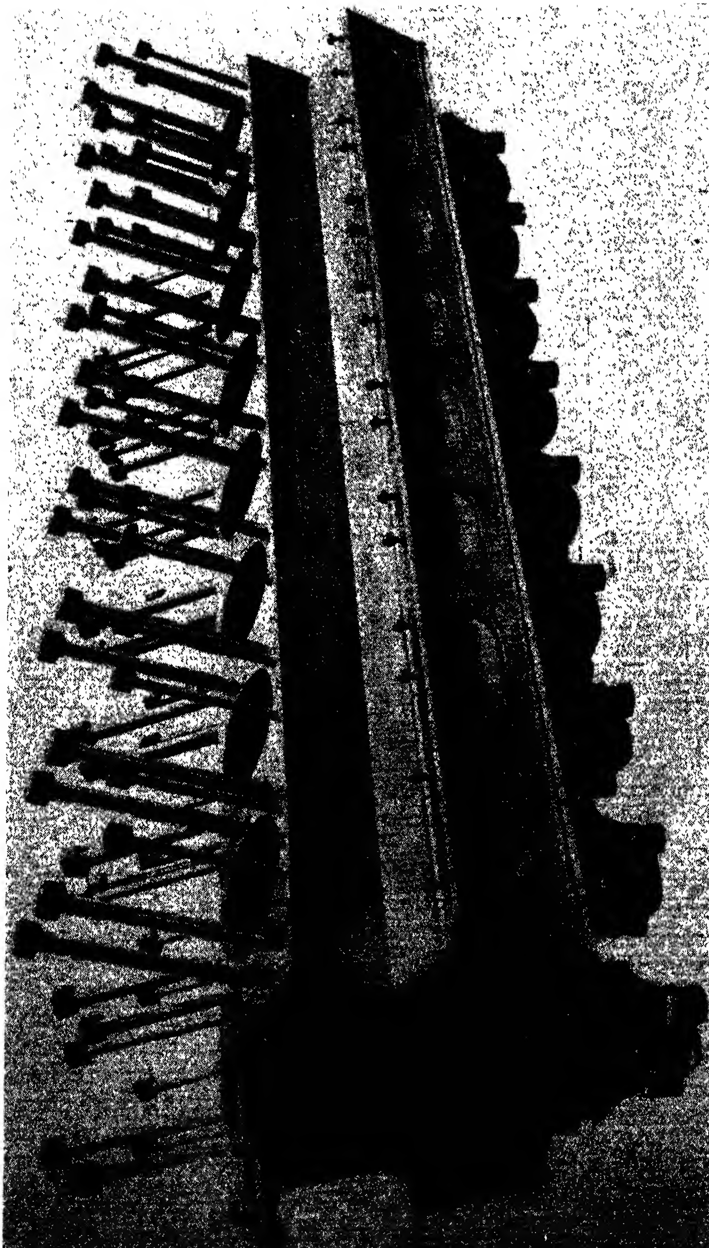


Fig. 3.—Alco Cylinder Block with Crankshaft Bearings and Cylinder-Head Studs.

It enters this groove at one side of the piston and passes on to the groove next above through a slot in the intervening land on the opposite side. In this way, the oil passes through the different grooves in succession, and then drains back into

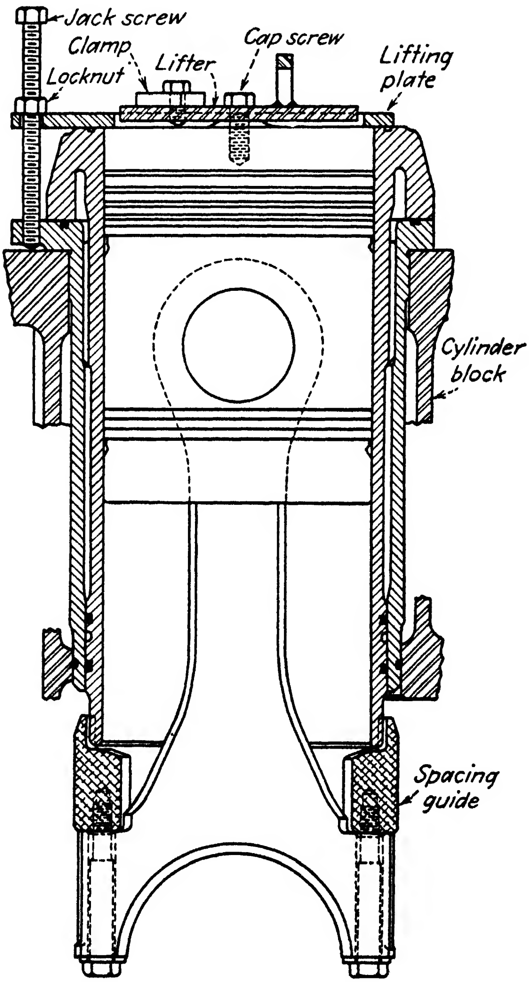


FIG. 4.—SHOWING METHOD OF REMOVING CYLINDER, LINER, PISTON AND CONNECTING ROD FROM BLOCK.

the base. Each piston carries three compression and three oil rings. The oil chamber within the piston pin is closed by pressed-in sheet-metal cups. Plugs are fitted into the ends of the piston-boss bores, and any oil finding its way into the spaces between the pins and these plugs drains back into the engine base.

Three cylinder-head studs pass through holes in the head, and in addition there are two studs between each pair of

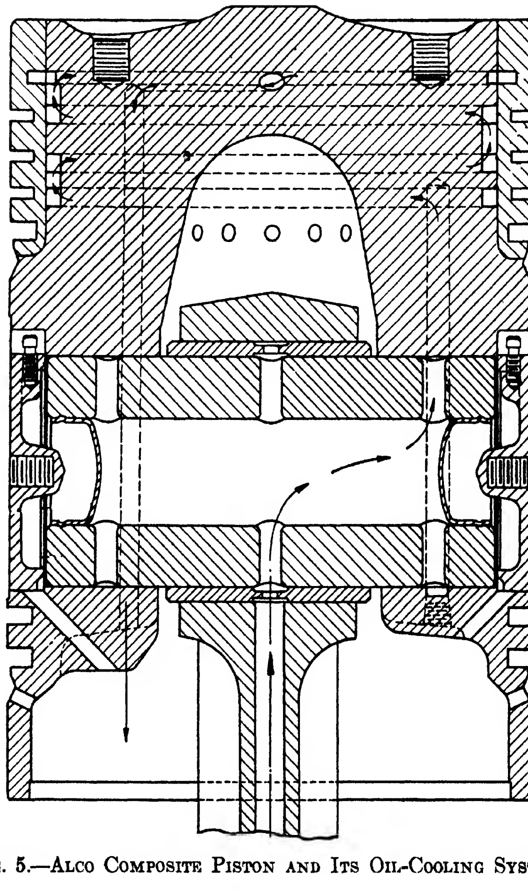


FIG. 5.—ALCO COMPOSITE PISTON AND ITS OIL-COOLING SYSTEM.

adjacent heads, by means of which pressure is exerted on the heads through the intermediary of clamping bars. Cylinder head-to-liner seals are metal-to-metal, the surfaces being lapped. Sealing gaskets are used at the water connections between the head and liner.

General Motors Railroad Engines—Two-stroke Diesel railroad engines are being manufactured by the Electro-Motive Division of General Motors Corporation at LaGrange, Ill. They are being built in six-, eight-, twelve-, and sixteen-cylinder models, all with the same bore and stroke of 8½ by 10 in. All are of the V-type, with an angle of 45 deg between the two cylinder banks. That gives uniform spacings between power impulses of 45 and 22.5 deg in the eight and sixteen, respectively, but uneven spacings in the other models—45 and 75 deg in the six, and 19, 26, and 49 deg in the twelve. The reason for using a uniform angle of Vee of 45 deg is that it results in a fairly narrow engine and leaves ample room in the cab to get at the engine from both sides; and the six is made a Vee rather than an in-line engine because more of the parts of the Vee can be made interchangeable with those of the other models. The displacement is 567 cu in. per cylinder, and the compression ratio is 16 to 1. Ratings of the different engines are as follows: Six-cylinder, 600 hp; eight, 800 hp; twelve, 1125 and 1200 hp; sixteen, 1500 hp. The six-, eight-, and twelve-cylinder engines are installed in switching locomotives, while the sixteen goes into freight, combination freight-passenger, and road switching locomotives. A cross section of the engine (known as the Series 567) is shown in Fig. 6.

In these engines a fabricated-steel crankcase supports the cylinders and the main-bearing frames. Top decks, cooling-water, and lubricating-oil manifolds are formed in the crankcase. Each cylinder with its water jacket is a separate casting of alloy iron. The cylinder is secured to the cylinder head by eight studs and nuts, and the assembly is held in place on the crankcase by cylinder-head crabs. Head-retainer steel forgings in the top decks are counterbored to receive the cylinder assemblies. Horizontal plates and vertical stress plates hold the whole assembly in alignment. Two parallel plates of the crankcase, located at the bottom of the cylinder water jackets, form the water manifold and are bored to receive the cylinders, sealing rings of synthetic rubber being inserted between these plates and the cylinders. The oil manifold is formed where the two cylinder banks join at the center of the engine. The oil pan is a fabricated-steel base to which the crankcase is bolted, and its bottom slopes toward a sump at the middle. Main and connecting-rod bearings are inspected and serviced through hand holes in the oil pan, which are located directly below similar hand holes in the air box. Oil tubes leading to the various bearings extend up about 2 in. into the manifold, so that no sediment can get into them. The exhaust ports

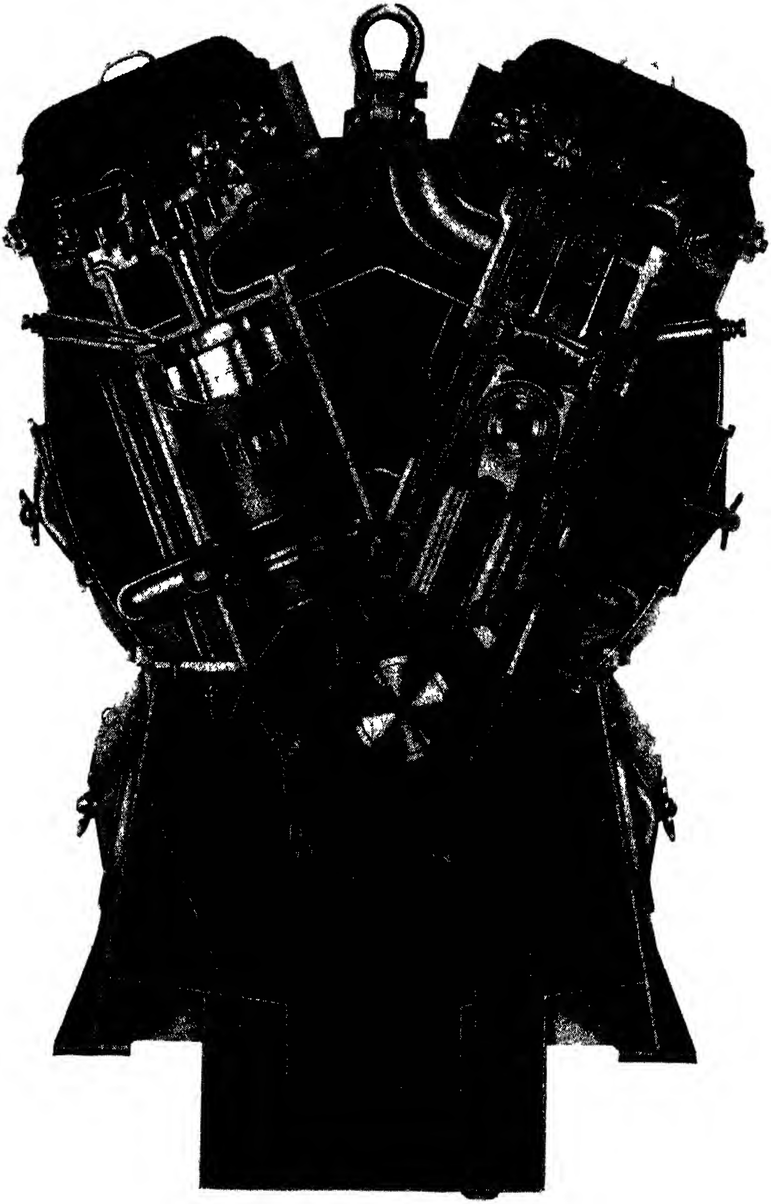


FIG 6.—CROSS SECTION OF GENERAL MOTORS (ELECTRO-MOTIVE DIVISION) V-TYPE, TWO-STROKE LOCOMOTIVE ENGINE.

in the cylinder heads line up with water-jacketed exhaust passages.

There are four exhaust valves in each cylinder head. As shown in Fig. 7, two valves are operated by a single rocker arm by means of a valve bridge. Hydraulic lash-adjusters are inserted between the valve bridge and the valve stems. Oil flows through a drilled passage in the valve bridge to the top of the lash adjuster. A coil spring resting on the cylinder head holds the valve bridge in contact with the rocker arm. The spring has a spherical seat on the cylinder head, and

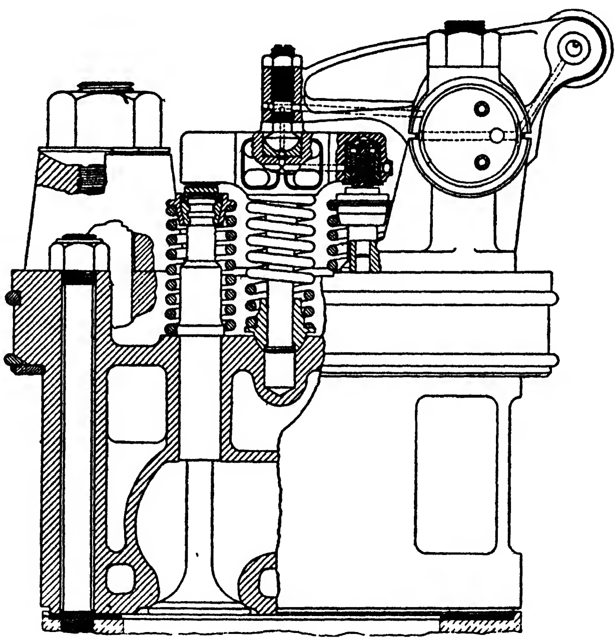


Fig. 7.—Valve-Rocker Mechanism.

there is a ball-and-socket contact also between the bridge and the rocker arm. Exhaust valves have a head diameter of 2½ in., 30-deg seats, and a lift of 0.686 in. The valve ports are venturi-shaped.

The pistons, of which two sectional views are shown in Fig. 8, are of alloy iron, in two parts. The body or main part is supported by a pin carrier held in position therein by a $\frac{3}{16}$-in. snap ring, with a bronze washer between the two parts. This so-called "floating piston" tends to prevent ring-belt cracking, as the temperature gradients in the piston body are less than in a piston with integral bosses and large supporting ribs. The interior of the piston forms a cooling chamber through which oil is circulated. Drill holes parallel

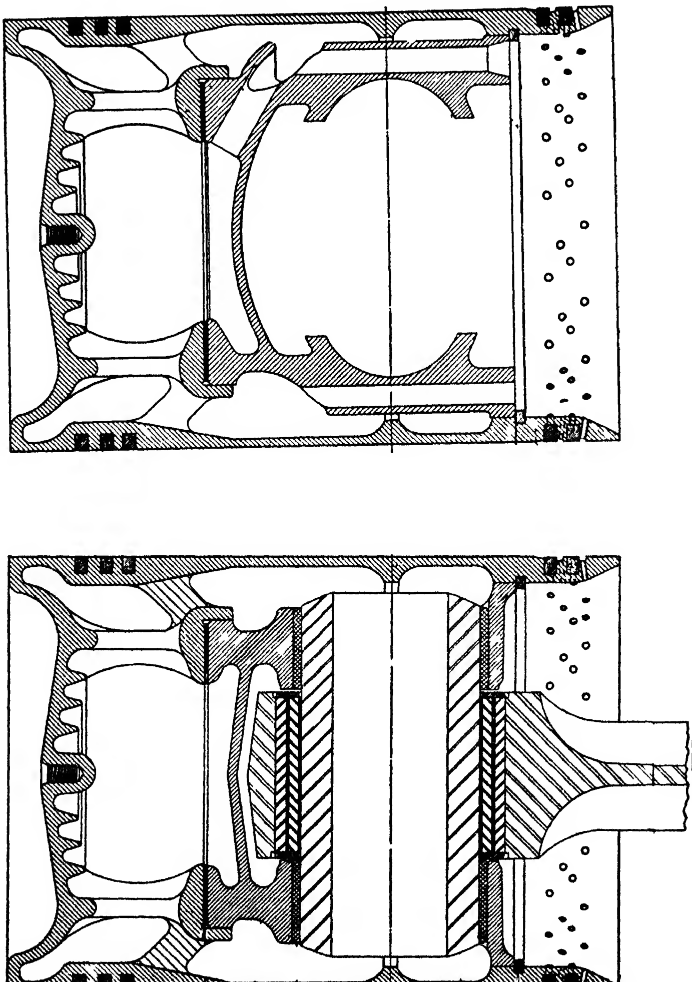


FIG 8—TWO SECTIONAL VIEWS OF THE TWO-PART PISTON

with the piston axis extend up through the pin carrier on opposite sides. Oil is injected into one of these holes—which has a tapering inlet—by a fixed nozzle directly below it, and returns to the sump through the other hole. The piston pin, of 3.5-in. diameter, is of the floating type, there being bronze bushings in the pin carrier and a silver-plated floating bushing in a fixed sleeve in the small end of the connecting rod.

Connecting rods are of the "blade-and-fork" type. The blade rod oscillates on the outside of the upper bearing shell and is held in place by a counterbore in the fork rod, as shown in Fig. 9. Serrations on the side of the fork rod match serrations on the two-piece, hinged "bearing basket." The outside of the upper shell in the fork rod provides the bearing surface for the blade rod.

The crankshaft, with main bearings of 7.5 and crankpin bearings of 6.5 in. diameter, is Tocco-hardened. In the six-

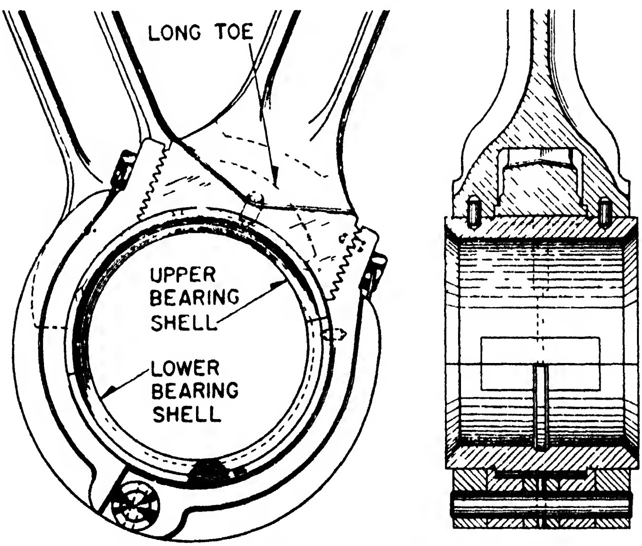


FIG. 9.—BIG ENDS OF THE "FORK-AND-BLADE TYPE" CONNECTING RODS.

teen-cylinder engine it is made in two sections and supported in ten bearings, including two center bearings. The generator armature serves as flywheel and is connected to the crankshaft by a flexible coupling, consisting of a relatively thin steel disc 36 in. in diameter (which comes with the generator), and a somewhat thicker disc welded to a counterbored ring, which latter is secured to the generator disc by bolts. A Harmonic balancer is located at the forward end of the crankshaft, and an accessory-drive gear mounted on the crankshaft immediately ahead of the balancer drives the water pumps, oil pumps, and governor. Camshafts are made in sections, each section carrying the cams for three cylinders in the six- and twelve-cylinder engines, and for four cylinders in the eight- and sixteen-cylinder models.

An overspeed trip mechanism, shown in Fig. 10, prevents injection of fuel if the engine reaches an excessive speed. In the latched position a notch in the reset lever engages the trip pawl, so that the pawl camshaft is held away from the rocker arm. Thus the engine camshaft is free to operate the injector rocker arms in the usual manner. When the engine reaches the speed for which the trip is set (910 rpm), centrifugal force on the flyweight mounted on the counterweight

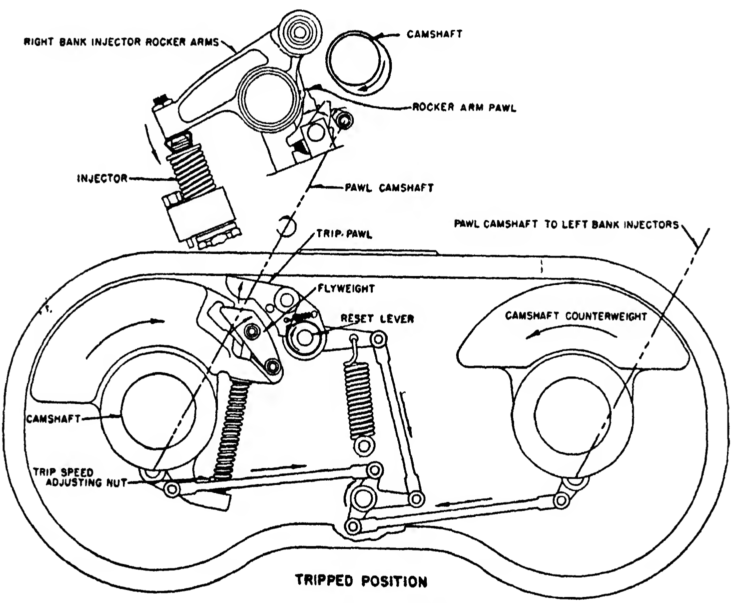


Fig. 10.—Overspeed Trip Mechanism.

of the right-hand camshaft overcomes the spring, and the flyweight moves out and engages the trip pawl. This pushes the shoulder of the trip pawl out of the notch in the reset lever, and the spring-loaded linkage rotates the pawl camshaft, which then engages the injector rocker-arm pawl. Then, as the engine camshaft lifts the injector rocker arm, the rocker-arm pawl snaps into a notch at the back of the rocker arm, holding the injector plunger in a depressed position and bringing the engine to a stop.

Each engine is fitted with two gear-type oil pumps combined in one, a plate separating the two pairs of gears. One pump delivers lubricating oil to the bearings, the other cooling oil to the pistons. A scavenging pump returns oil from the sump to the strainer chamber through an oil filter and an oil cooler. The pressure in the lines to the main bearings ranges between 45 and 60 psi, the piston-cooling pressure between 20 and 30 psi. An oil separator for the oil vapor and blow-by which the blower draws from the crankcase is installed at the rear end of the engine.

The engines are equipped with a low-oil-pressure alarm, which gives both visual and audible signals and slows the engine down to idling speed when the oil pressure drops to a predetermined limit. Another safety device gives a signal when the vacuum in the suction line becomes excessive as a result of clogging of the oil strainer. The oil pump is geared to turn at 1.415 times crankshaft speed. On the twelve-cylinder at 800 rpm the lubricating pump has a capacity of 86 gpm; the piston-cooling pump of 43 gpm, and the scavenging pump of 170 gpm.

Blowers are of the triple-lobed, helical type, and both the housing and the rotors are of aluminum. There are single blowers on the six and the eight, while the twelve and sixteen have two blowers each. Each rotor is pressed onto a tubular shaft. A flanged quill shaft connecting to one of the tubular shafts by a serrated joint, extends through it and transmits the blower drive. This light shaft protects the rotors and drive from shock, because of its flexibility. Blowers on the six and twelve have a capacity of 2000 cfm each at 800 rpm, while those on the eight and sixteen have a capacity of 2700 cfm each. At the speed mentioned the blower pressure ranges between 7 and 8 in. of mercury.

The blower discharges into an air box which is formed by the sides and ends of the crankcase and the top and bottom decks. Scavenging air enters the cylinders through ports in the cylinder wall 1.36 in. high. Inlets to these ports are strongly flared.

Weights of the four engines are approximately 15,000, 17,500, 23,500, and 30,000 lb.

Fairbanks-Morse Engine—Fairbanks, Morse & Co. in 1944 announced a ten-cylinder, two-stroke, double-piston engine for railroad and other applications. Development work on this type of engine (which is produced also in other cylinder numbers) was begun during the thirties, with railroad applications in view. During the war, engines of this type were supplied to the Navy for submarine and surface vessels. A switching locomotive with a six-cylinder engine was placed

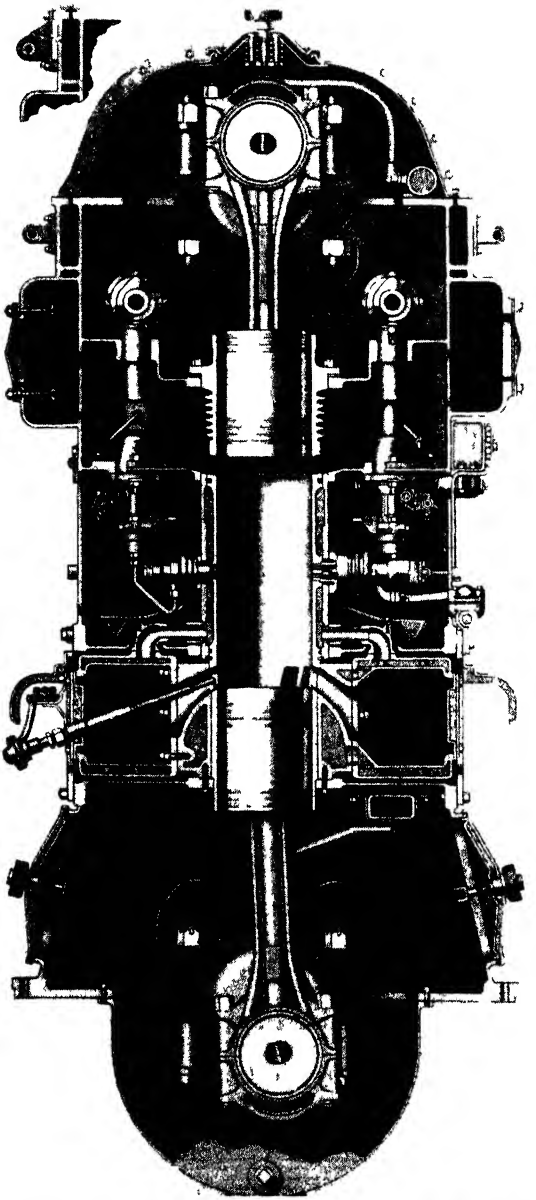


FIG. 11.—CROSS SECTION OF FAIRBANKS-MORSE LOCOMOTIVE ENGINE.

in service by the Chicago, Milwaukee & St. Paul Railroad in 1944.

With a bore of $8\frac{1}{8}$ and a stroke of 10 in., the ten-cylinder model has a piston displacement of 10,370 cu in. and is rated 1600 hp at 720 rpm, which corresponds to 84.9 psi bmep. At 850 rpm it develops a gross output of 2150 hp and a net of 2000. This is a direct-injection engine and it shares the characteristics of that type: The maximum combustion pressure is given as 1250 psi and the fuel consumption at the rated speed and load as 0.375 lb per hp-hr.

The cylinder block is built up of steel plates which are cut to shape, located in jigs, and welded together to form a light, rigid structure. Transverse vertical members together with horizontal decks form enclosures, bearing housings, and supports for the operating parts. The four horizontal decks are bored to receive the cylinder assemblies. An extension is provided at one end for the attachment of the Roots blower. Within the cylinder block are formed a number of compartments, which are clearly shown in the cross section, Fig. 11. First above the lower crankcase comes the exhaust-manifold and deck compartment, which extends lengthwise of the block on both sides. An exhaust deck and two exhaust manifolds are installed in this compartment. Next come the injection-nozzle compartments, which house the injection nozzles, injection pumps, air-start check valves, cylinder relief valves, and fuel control rods. Immediately above these is the air-receiver compartment, which forms a passage for the scavenging air to the inlet ports of the cylinders. Above this is the upper crankcase.

After being welded, the block is sand-blasted, and a Magnaflux test is made to check the welding in certain vital spots. It is then annealed to relieve the stresses set up by welding.

The two ten-throw crankshafts—upper and lower—are cast of alloy iron. They are set out of phase by 12 deg, the lower crankshaft leading. This not only makes it possible to keep the inlet ports open after the exhaust ports have closed, but also results in most of the power being transmitted directly to the lower crankshaft, which is the output shaft. It is figured that 72 per cent of the total power is imparted to the lower shaft directly, and only 28 per cent to the upper, and of this latter fraction a considerable amount is used in driving the blower, so that relatively little power has to be transmitted by the vertical shaft, its flexible coupling, and the two pairs of spiral bevel gears which connect the two crankshafts. Exhaust ports open 16 deg ahead of the scavenging ports. The exhaust extends over 112 deg of crankshaft rotation and the inlet over 104 deg. A torsional vibration damper of the dynamic type is fitted to the lower crankshaft.

Gears on the two crankshafts mesh with pinions whose shafts are mounted in roller and thrust bearings in the upper and lower drive housings. A flexible coil-spring coupling unit, an adjusting flange, and a flexible coupling hub connect the two coupling hubs together and complete the drive. The vertical flexibility allows for expansion of the cylinder block and also for a small amount of misalignment. Its torsional flexibility enables the coupling to absorb any torsional vibration. These and other details are clearly shown in the two part-sectional views Figs. 12 and 13.

The cylinder assembly, which consists of a liner, jacket, rubber rings and lock ring, is bolted into the cylinder block by means of lugs near the upper end of the liner. That part of the liner within the air receiver is provided with circumferential cooling ribs, while the part within the jacket has longitudinal ribs designed to guide the cooling water upward. The jacket is shrunk to the liner. Halfway up the liner there are openings for the two injection nozzles, an air-start check valve, and a cylinder relief valve. Tapped holes for lifting eye bolts are provided in the lugs by which the cylinder assembly is bolted to the block.

Each piston carries four compression rings near the closed end and an oil scraper ring and two oil drain rings near the open end. Piston-pin brackets bolted into the pistons, in addition to transmitting the gas pressure from the piston crown to the piston pin, form a chamber within the piston through which cooling oil circulates. Oil is admitted to this chamber through the drilled connecting rod. The eye at the small end of the connecting rod is provided with a bronze-lined steel bushing. Big-end bearings have steel backs lined with lead-base bearing alloy containing a special hardener. Main bearings are supported between saddles in the cylinder block and the bearing caps. The half shells are doweled together and consist of bronze backs with lead-base alloy linings. End thrust is taken on the bearings adjacent to the bevel gears. Limits on main-bearing clearance at assembly are 0.009 and 0.0115 in. In an engine of this type all bearing wear comes on the shell in the cap.

Two injection pumps and two nozzles are provided for each cylinder. Fuel is delivered to the injection pumps by a gear-type transfer pump under a pressure of 25 psi. The injection pumps are located within the cylinder block on opposite sides of the cylinders, and the connections to the nozzles are relatively short. These pumps are operated by camshafts driven from the upper crankshaft through a toothed chain at the control end. They are of the conventional jerk-pump type, with delivery control by racks. All of the racks on the same side of the block are adjustably connected to a control rod, and the two control rods are linked to a Woodward hydraulic-relay type of governor at the control end of the block. Fuel nozzles are of the differential type and have three orifices of 0.020-in. diameter, inclined 15 deg to the nozzle axis. The nozzles, which are set to open at 3000 psi, are cooled by the water in the cylinder jackets. To facilitate their removal, the camshafts are made in sections which are flange-bolted together.

The blower is of the Roots type, with triple-lobed helical rotors. It is driven from the upper crankshaft through helical gears and a flexible coupling, at twice crankshaft speed. At rated engine speed it has a capacity of 6000 cfm, which is about 39 per cent more than the rate of piston displacement. At this same speed the blower has a volumetric efficiency of 77 per cent and a mechanical efficiency which increases with the scavenging pressure and which at a pressure of 4 psi is equal to 58 per cent. Under these conditions the blower absorbs slightly more than 160 hp.

A number of different governing systems can be supplied, but all incorporate the same basic hydraulic-relay type of governor which was described in Chapter VII. The governor is driven from a coupling at the top of a flexible pump drive. To absorb any vibration which might pass through the flexible pump drive from the lower crankshaft to the governor, the coupling shaft is designed to "float" between the pump drive and a second intermediate drive shaft.

Lubrication of the engine is entirely by pressure, all bearings, including those of the vertical drive shaft, the camshafts, blower, pump and governor drives, being oiled automatically. As the oil is used not only for lubrication, but also for cooling purposes, it is circulated at a high rate, and the herringbone-gear-type oil pump has a delivery of 280 gal per minute and requires 18.5 hp to drive it at normal engine speed. The relief valve is set at 60 psi.

The engine is started by compressed air. The starting system includes air flasks, a reducing valve, a relief valve, gauges, and the necessary piping to the high-pressure header and the cylinders. The engine starting mechanism includes an air-start control valve, an air-start distributor, a header, pilot-air tubing, and air-start check valves at individual cylinders.

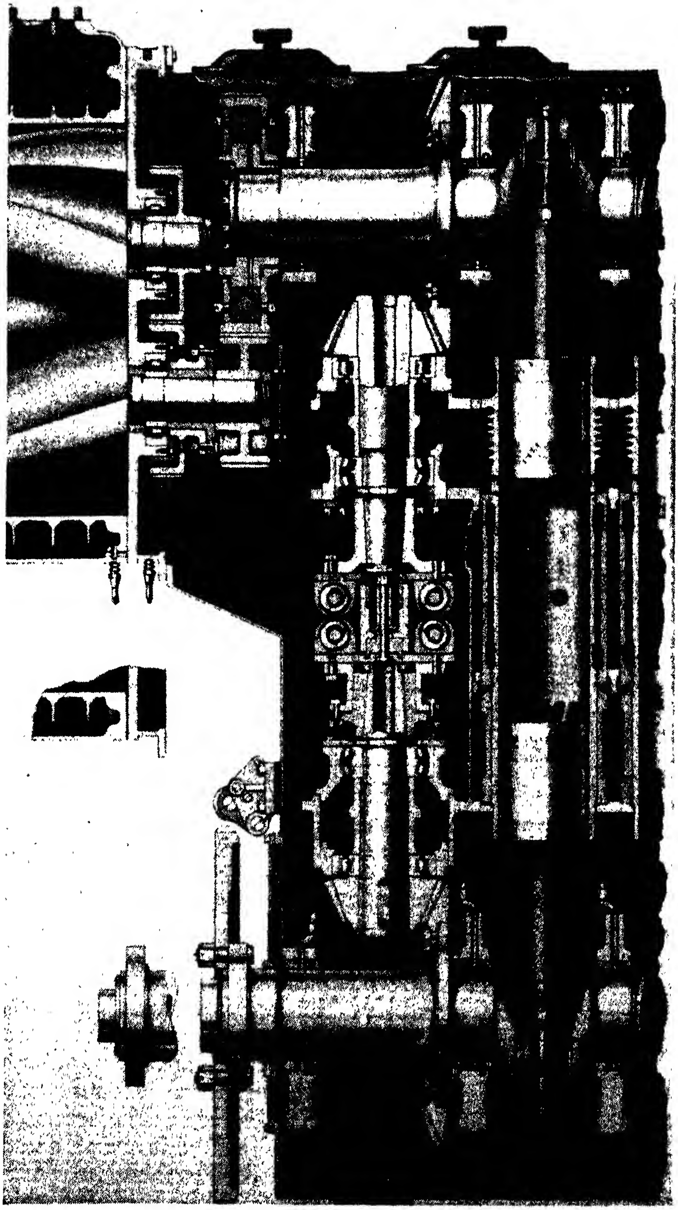


FIG. 12.—BLOWER END OF FAIRBANKS-MORSE ENGINE, SHOWING BEVEL-GEAR DRIVE AND PART OF BLOWER.

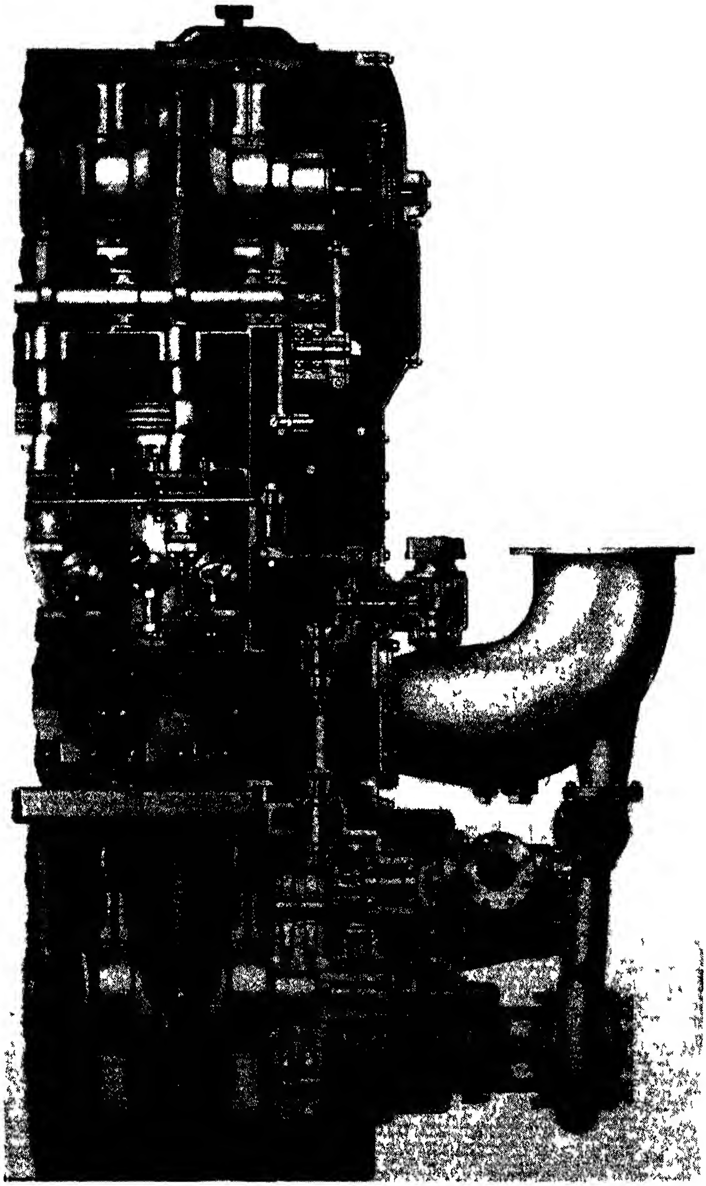


Fig. 13.—Control End of Fairbanks-Morse Engine, Showing Injection Pumps, Torsion Damper, and Other Details.

The starting mechanism operates at 250 psi, and a reducing valve with by-pass is installed in the supply line. From the header beyond the reducing valve a pipe leads to another pressure gauge on the instrument board. For the protection of the system a relief valve set at 275 psi is installed in the air-supply line between the reducing valve and the engine. The air-start control valve is mounted near the control end of the engine, on the side opposite the control quadrant. When the control-shaft lever is moved to the "Start" position, the air-start control valve is opened through a linkage. Compressed air then enters the header which leads to the air-start check valves of the individual cylinders. Air also passes into the pilot-air supply pipes connected to the air-start distributor. This latter has one pilot air valve for each cylinder. These valves are arranged radially in a circle around the air-start distributor camshaft, according to the firing order of the cylinders. A spring normally holds the valve out of contact with the cam, but when air enters the distributor from the control valve, the air pressure overcomes the spring and forces the valve plunger into contact with the cam. For any given position of the camshaft, one valve will be on the low point of the cam and therefore will be fully open, and two other valves, on opposite sides of this one, will be partly open. These three valves will admit air to three air-start check valves. The air pressure then opens the check valves and the air rushes into the cylinders, forcing the two pistons in the cylinder apart and causing the crankshafts to revolve. As soon as the engine begins to fire the control-shaft lever is moved to the "Run" position. This closes the air-start control valve and shuts off the air pressure from the distributor.

Maybach Engines—Maybach Motor Manufacturing Co. of Friedrichshafen, Germany, which in 1933 furnished the engines for the original "streamliner," the "Flying Hamburger," in 1950 brought out an entirely new line of Diesel railroad engines comprising four-, six-, eight-, and twelve-cylinder, four-stroke models. The early Maybach engine was a development of one built for lighter-than-air craft (Zeppelins), and was unusually light. In the design of the new engines the principal aim is said to have been to combine high specific output with long life (long periods between overhauls). There is nothing very unusual in this goal, which probably is aimed at by a majority of all engine designers, but in this case the quest for the result sought has led to some unusual design features.

In the following description the reference for the most part is to the supercharged twelve-cylinder model, of which a photograph is shown in Fig. 14 and part of a longitudinal section in Fig. 15. The cross section, Fig. 16, is of the six-cylinder model, this having been chosen because it can be shown to better advantage in the space available. In this view the oil sump, containing two gear pumps for circulating the

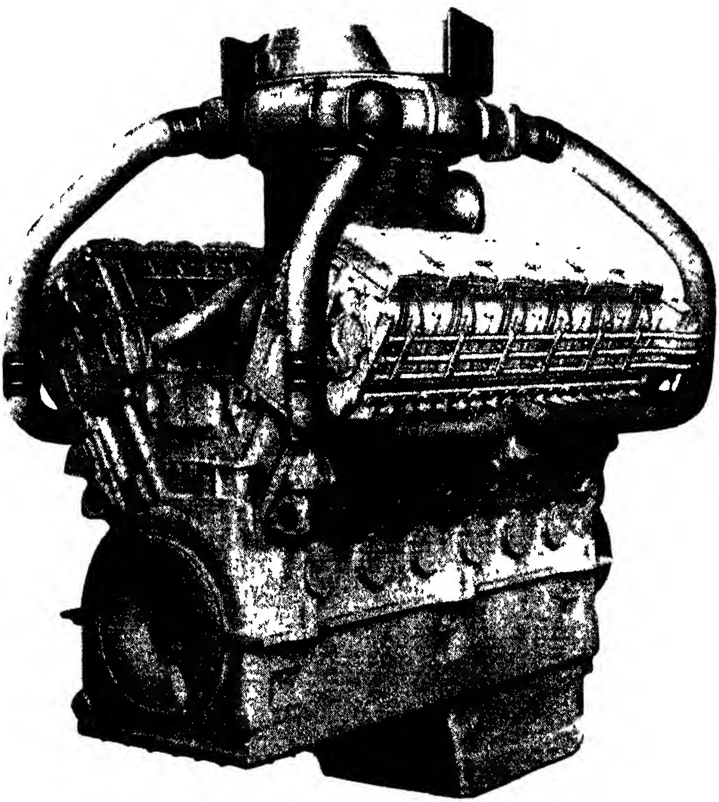


Fig 14—Maybach Twelve-Cylinder, Supercharged, Four-Stroke Rail-Car Engine.

oil through separate piston-cooling and lubricating circuits, has been omitted, for reasons of space. Such parts as pistons, cylinder heads, valves, rocker levers, injection units, connecting rods, and cylinder liners are interchangeable throughout the whole line, and crankshafts, camshafts, and "frames" are interchangeable between the four- and eight-cylinder and between the six- and twelve-cylinder models, respectively.

The engines have a bore of 7.28 and a stroke of 7.87 in., which makes the displacement of the individual cylinder 328 cu in. and that of the twelve-cylinder engine, 3936 cu in. The twelve-cylinder model shown is supercharged by a turbo-blower and has a continuous-output rating of 1200 hp at 1600 rpm. Making allowance for the fact that the American horse power is 1.42 per cent larger than the metric one, this corresponds to a bmep of 149 psi. However, as used by the German State Railways the output is limited to 1000 hp, which limit is said to be imposed by the capacity of the largest transmission available.

The engines are of the direct-injection type, having spherical combustion chambers in the individually-cast cylinder heads. To provide the necessary room for these combustion chambers, three inlet and three exhaust valves are used, arranged symmetrically around the combustion chamber. In continuous service, engine outputs usually are limited by overheating of pistons, which causes the piston rings to stick; by excessive loads on main and connecting-rod bearings, which induce fatigue in the linings, or by overheating of the exhaust valves. To prevent trouble from ring-sticking, the pistons of Maybach engines, which are of cast iron, have their crowns cooled by positive oil circulation. A cooling chamber is formed in the top of the piston, between the piston proper and a steel head secured to it by six cap screws. There is a cooling-oil circuit, including an outside oil-cooler or heat interchanger, which is entirely separate from the lubricating-oil circuit. Oil enters and leaves the cooling chamber in the piston crown through telescoping tubes, and the passage through the cooling chamber is said to be so designed that the greatest cooling effect is produced where it is most needed. The intense cooling makes possible the use of pistons of true cylindrical form (not tapered and not ovaled), fitted with very small clearances and carrying only three narrow compression rings. These latter are located in grooves in the detachable steel head. The narrow ring belt leaves a comparatively long effective bearing surface and makes possible a favorable location of the piston pin.

Valve troubles usually increase and decrease with the valve diameter. As there are three exhaust valves in each cylinder, the valve diameter is small and valve cooling presents no problem. Each set of valves is actuated from a separate overhead camshaft through small rocker levers. Direct actuation reduces the weight of the valve-reciprocating parts and the spring force required to close the valves. Valve life is increased further by causing the valves to rotate in opera-

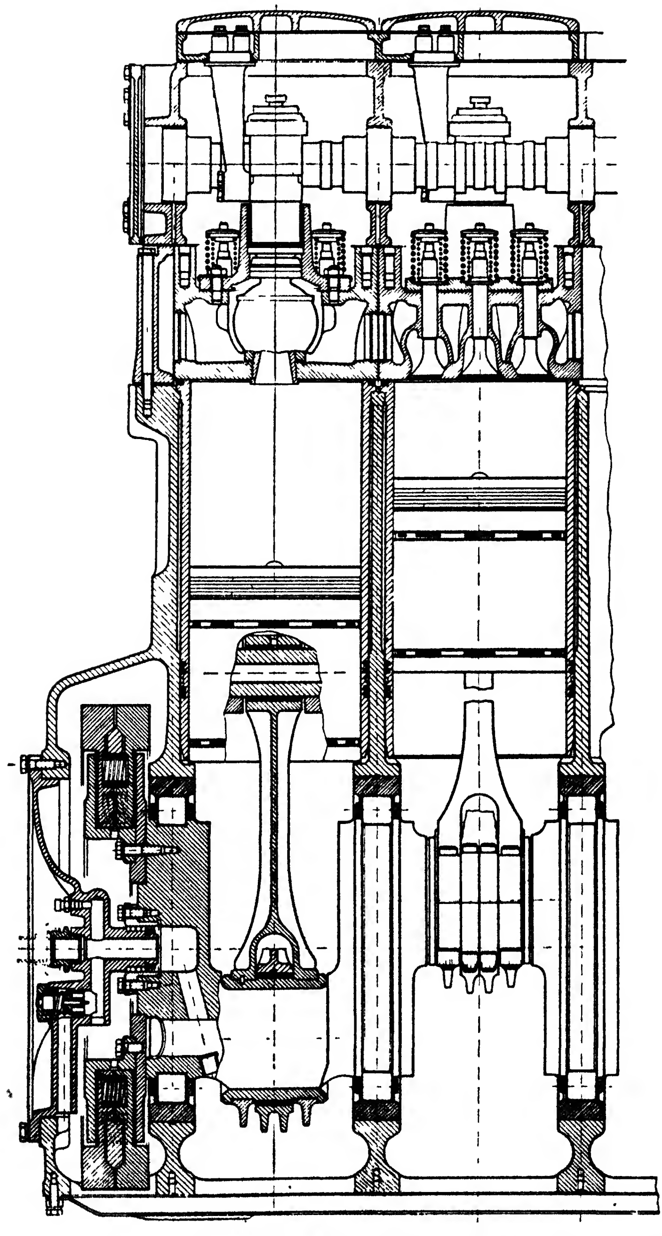


FIG. 15.—LONGITUDINAL SECTION OF FORWARD PART OF MAYBACH TWELVE-CYLINDER ENGINE.

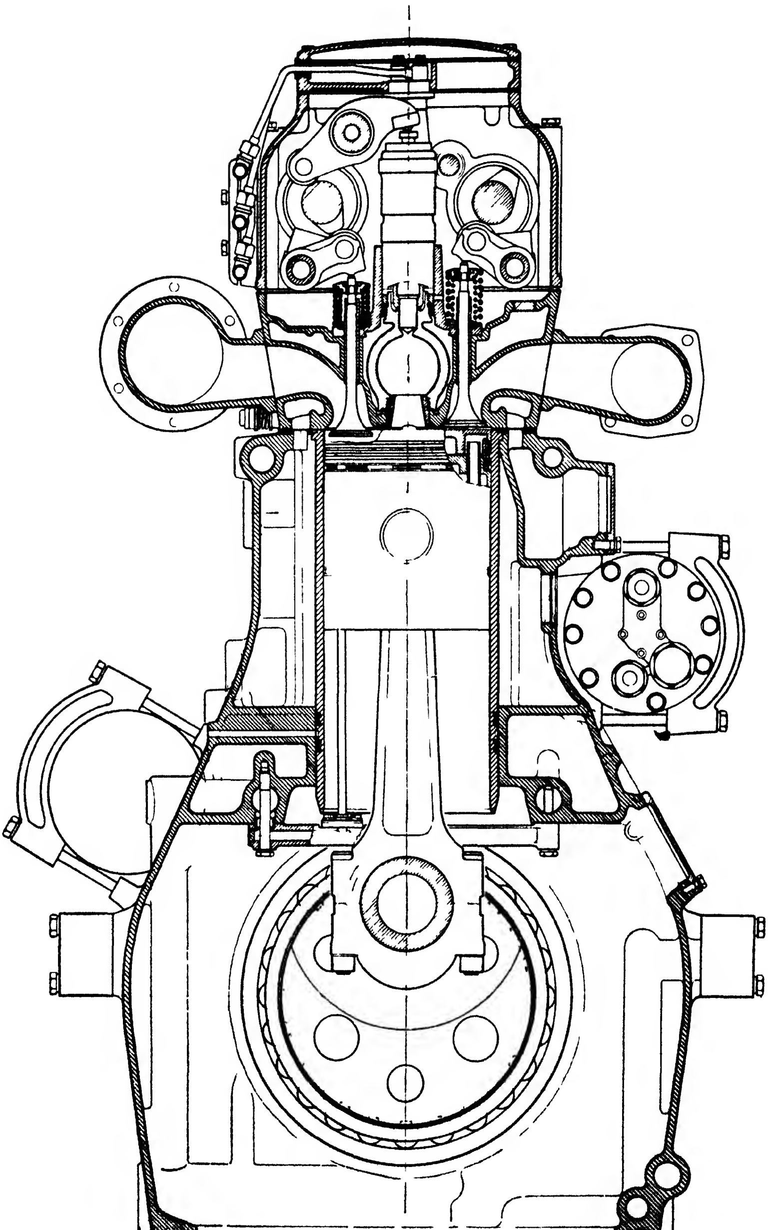


FIG. 16—CROSS SECTION OF MAYBACH SIX-CYLINDER ENGINE.

tion and by providing them with hydraulic clearance take-ups. Opening pressure is applied to the valve stems in such a manner that they produce no side thrust on their guides.

Fuel is injected into the spherical combustion chamber by a combined pump and injector, which was developed in collaboration with the L'Orange fuel-injection-equipment firm. Efficient combustion is possible only if the fuel is injected during a relatively short period. That calls for high injection pressures which, if produced in multi-unit pumps, would result in pressure waves in the high-pressure lines and operating difficulties connected therewith.

The most radical features of the engine are to be found in the crankcase and crankshaft design. The crankcase is similar to what in this country has been known as the barrel type (the Germans call it the tunnel type), and is cast integral with the two banks of cylinders, which certainly should make a rigid block. Instead of crank arms, the seven-bearing crankshaft has cylindrical discs which form the inner races for the large-diameter roller bearings supporting the shaft. This makes practically the whole length of the shaft available for bearing purposes, and both the main and crankpin bearings can be made of liberal size, even though the cylinders are placed as close together as possible. One forked and one plain connecting rod are carried on each crankpin. The bearing of the forked rod, which extends practically the whole length of the crankpin, has a diameter of approximately 4⅜ in. and a length of 4 in. It consists of a heavy-walled steel shell made in halves, with a lead-bronze lining to which a protective coating is applied by electro-deposition. The plain rod has a bearing on the outside of the shell, also lead-bronze lined. Since it has only a slight rocking motion on the shell, it does not require a great length. Serrations on the faces of the joint between the half-shells help to prevent deformation of the bearing in service.

The turbo-blower is located centrally on top of the engine, hence connections from it to the different cylinders are of nearly equal length. The weight of the supercharged twelve-cylinder model is given as 3600 kg or just under 8000 lb. That corresponds to a specific weight of 6.7 lb per bhp on the basis of the manufacturer's continuous-output rating, and to 8 lb on the basis of the maximum output of the engine on the German State Railways.

Sulzer Engine—What was claimed to be the largest railroad Diesel engine built up to that time was supplied by the Swiss firm of Sulzer Brothers to the French P.L.M. railroad in the late thirties. It is a twelve-cylinder with a bore and stroke of 12.2 by 15.35 in., is supercharged, and has ratings of 2200 hp at 700 rpm for one hour and 1900 hp at 600 rpm continuously. Unlike most twelve-cylinder engines, it is not a V type, but has two rows of parallel cylinders, all in the

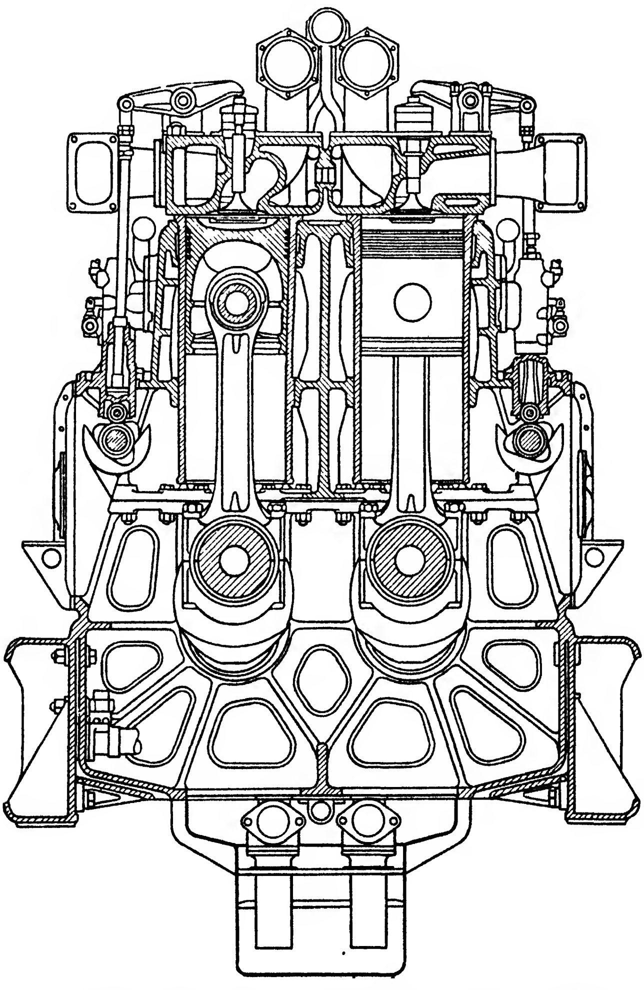


Fig. 17.—Sulzer Twin-Type Locomotive Engine of 2200 HP at 700 RPM.

same block. The pistons of each row work on a separate crankshaft and the two crankshafts drive the generator armature through gearing having a ratio of 5:6. This, of course, means that electric drive is being used. A cross section of the engine is shown in Fig. 17.

Air is supplied to the cylinders by two turbo-blowers. In the event one of the blowers should fail, it can be completely cut out and the engine half affected operated with atmospheric induction, with which it develops about one-half its normal power.

The crankcase lower half is made of steel castings welded together. It is substantially ribbed, and bolted to a common bedplate with the generator. Main bearings are formed in the lower half of the crankcase, which latter has an oil sump of welded sheet steel secured to its bottom. The cylinder block also is made of welded steel castings and is bolted to the crankcase. Cylinder liners of alloy iron are used, and are inserted in the block from above. At the top the joint is sealed by a copper gasket under a flange on the liner, at the bottom with three rubber rings.

Each of the seven-bearing crankshafts carries a torsion damper at the front and a drive gear at the rear end. There is a separate head on each cylinder. The injection pressure is 3900 psi (direct injection). Each injector is supplied by its own pump. A consumption of 0.41 lb per hp-hr is guaranteed, but in acceptance tests the consumption is said to have been as low as 0.37 lb per hp-hr.

There are two oil pumps in the lubrication system, one supplying oil under pressure to the bearings, the other passing it through an oil cooler. Each has a capacity of 16 gpm at full engine speed. The engine weighs approximately 20 lb per hp.

CHAPTER XIV

Supercharging

Most four-stroke engines of both the spark-ignition and compression-ignition types operate with natural induction; that is to say, the charge of combustible mixture or air is drawn into the cylinder by the pumping action of the cylinder itself. With natural induction the amount of air drawn in at low speeds is slightly less than the piston-displacement volume of air at atmospheric density, while at the speed corresponding to the peak of the horse-power curve it drops to about 70 per cent of that volume. The ratio of the mass of air actually drawn in to that of the cylinder-displacement volume at the prevailing atmospheric density is called the volumetric efficiency of the engine. By using other charging means than the working cylinder itself, more air can be forced in per cycle, and this greater air charge will burn more fuel and develop more power. As pointed out above, the volumetric efficiency drops with increase in speed, and sometimes the extraneous charging means is used only to secure at high speeds the same cylinder charge as that obtained with natural induction at low speeds or under the most favorable conditions. In that case the process may properly be called "charge restoration." But if this extraneous charging means maintains in the inlet manifold a pressure materially higher than atmospheric, and the pressure in the cylinder at the time of inlet-port or inlet-valve closing is above atmospheric, then the process is known as "supercharging."

Supercharging Four-Stroke Engines—Blowers are commonly used to scavenge high-speed two-stroke engines, and the various types were described and illustrated in Chapter XII. But while these blowers often have a capacity about 50 per cent greater than the pumping capacity of the engine, most of the extra air delivered by them is blown right through the cylinders for internal cooling, and at the moment the inlet port closes the pressure in the cylinder is little, if any, above atmospheric. As each down-stroke or in-stroke is a power stroke, the heat stresses might be excessive if a material supercharge were used.

With four-stroke engines supercharging is now used to a considerable extent for large powers. It has been found that the output of an engine can be increased by as much as 50 per cent, and that without adding materially to the bearing loads and to heat stresses in such parts as pistons, rings, and valves. These results are obtained without intercooling the air between blower and engine, and it is believed that intercoolers—which may be warranted on quite large engines—would make it possible to double the output as compared with atmospheric induction. With the supercharging ratios currently employed, resulting in bmeps of from 130 to 150 psi, the specific consumption of the supercharged engine generally is less than that of the unsupercharged one in the upper half of the load range. For small load factors the supercharged engine consumes more fuel than the unsupercharged one, because within that range the power required to drive the blower—which in the case of a mechanical drive varies with the speed but not with the engine load—has greater influence on the efficiency of operation.

Valve Overlap—To secure the full advantage of supercharging in a four-stroke engine, the inlet and exhaust periods must be made to overlap considerably, so that the cylinder will be fully scavenged. Scavenging is an effective aid in cooling the piston crown and other high-temperature portions of the combustion-chamber wall. It can be readily effected in the case of direct-injection engines, in which the entire combustion space is in a single, compact volume, but involves difficulties in its application to other types, such as prechamber engines, in which part of the combustion chamber is separated from the remainder by a partition with narrow passages. The amount of valve overlap required for effective scavenging increases with the speed of the engine. Fig. 1 shows the variation of the exhaust temperature and that of the specific fuel consumption with load for two different overlaps of inlet and exhaust periods (95 and 149 degrees), as observed on a Westinghouse single-cylinder 9 by 12-in. engine operating with a supercharge of 9 in. of mercury. In the case of the long overlap the inlet opened about 80° ahead of, and the exhaust closed about 70° after, top center.

Effect on Output—Surprise has sometimes been expressed at the fact that the engine power apparently increases more rapidly than the charging pressure. For instance, it has been found that if a pressure gauge on the inlet manifold shows 5 psi, the power will be 50 per cent greater than with atmospheric induction. If atmospheric pressure at sea level is taken to be equal to 15 psi, the gauge pressure of 5 psi corresponds to an absolute pressure one-third greater, and a gain in output of that order might be expected. However, atmospheric pressure at sea level is only 14.7 psi, and owing to the altitude of the engine above sea level and a drop of pressure in the air cleaner the inlet-manifold pressure with atmospheric induction may be no greater than 14 psi. That would account for a gain of 40 per cent with 5 psi gauge manifold pressure. The remainder of the gain evidently is accounted for by an increase in the combustion efficiency, and especially an increase in the mechanical efficiency.

The compression pressure increases substantially in the same proportion as the manifold pressure, but the maximum combustion pressure does not increase nearly so much, be-

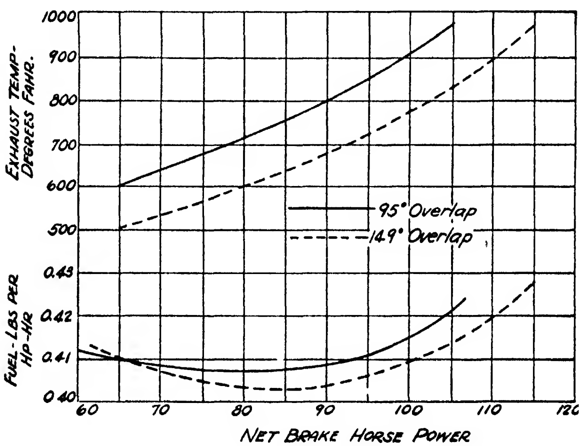
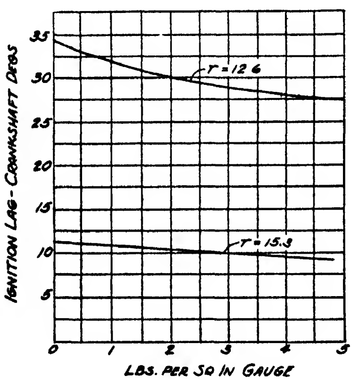


Fig. 1.—Variation of Fuel Consumption and Exhaust Temperature with Load and Valve-Period Overlap.

cause the ignition lag is reduced by the greater density of the charge. A diagram showing how the ignition lag varies with the charge density, based on results obtained by Professor Bird, was given on page 56. Fig. 2, which is taken from an N.A.C.A. report, shows how the ignition lag varies with the supercharge ratio for two different compression ratios. Harte Cooke, of the American Locomotive Company, in an S.A.E. paper made the statement that when using practically the normal compression ratio and a supercharge of 5 psi, he found no appreciable increase in the maximum combustion pressure. O. Thornicroft in England found that for an increase in the inlet-manifold pressure from 0.9 to 1.2 atmospheres (33 per cent) the maximum combustion pressure increased from 890 to 985 psi, or only 10.7 per cent.

British Experimental Results—Some experimental results obtained with a supercharger by O. Thornicroft in England are given in Fig. 3. As already mentioned, raising the inlet pressure from 0.9 to 1.2 atmospheres increased the maximum combustion pressure from 890 to 985 psi. The bmep and the temperature of the air at the inlet increased with the inlet pressure as shown by the graphs. It will be seen that at 1.2 atmospheres inlet pressure the bmep curve has become practically horizontal, indicating that a further increase in the degree of supercharge would bring little further gain in output. Thornicroft's experiments were made with an engine of 6-in. bore by 6.5-in. stroke running at 2000 rpm. Compression of the air by the blower naturally raises its temperature,



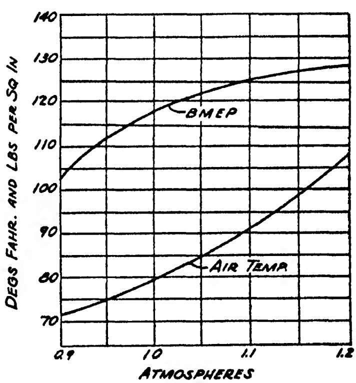


FIG. 2 (*Left*).—VARIATION OF IGNITION LAG WITH SUPERCHARGE PRESSURE AND COMPRESSION RATIO.

FIG. 3 (*Right*).—VARIATION OF INLET TEMPERATURE AND BMEP WITH INLET PRESSURE (SUPERCHARGE).

and in this case the inlet temperature rose from 72 F at 0.9 atmosphere inlet pressure to 108 F at 1.2 atmospheres. Thornicroft's experiments date back to the pioneer days of supercharging in the high-speed Diesel field. At present charging pressures of about 1.40 atm are frequently used, and the gain in output is greater than that obtained by him.

Types of Blower—There are essentially four types of blower available for supercharging Diesel engines, as follows:

1. Reciprocating pumps as used by Attendu, Junkers, Sulzer, and Petter in two-stroke engines.
2. Rotating so-called positive-displacement pumps, such as the Roots blower.

3. Mechanically-driven centrifugal blowers.
4. Turbo-blowers, that is, centrifugal blowers driven by exhaust turbines.

The reciprocating pump has the most desirable delivery characteristics for application to road-vehicle engines, as its delivery-speed curve is very similar to the volumetric-efficiency curve of an engine, the delivery being substantially constant over a considerable speed range and dropping off moderately in the higher speed range. This is the only type of pump or blower which maintains (and even increases) its delivery per cycle when the engine is pulled down in speed by a heavy load, and which, therefore, increases the "lugging"

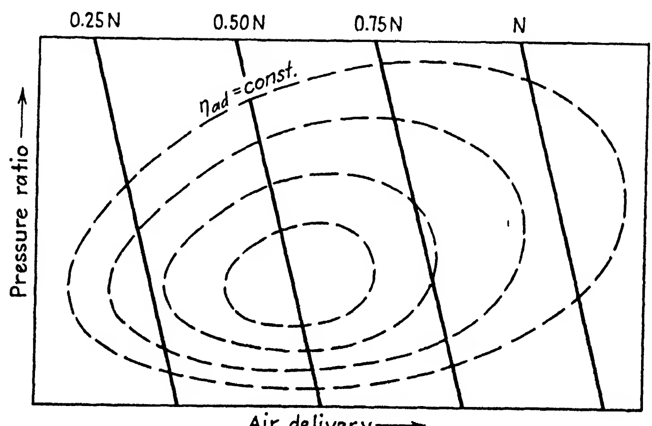


FIG. 4.—RELATION BETWEEN PRESSURE RATIO, SPEED, AND AIR DELIVERY OF ROOTS-TYPE BLOWER.

power. Some experimental work on a unit comprising two power cylinders with a compressor cylinder between them has been done by Ricardo. As there must be one compressor cylinder for every two power cylinders, this type of supercharger would add materially to the weight and bulk of the engine, and it probably will be limited to engines with few cylinders.

As regards the second type, positive-displacement rotary blowers, these may be divided into two subclasses, some having internal compression while others have not. The former are more efficient and probably are to be preferred where it is intended to use relatively high charging pressures, say more than 8 psi.

Characteristics of Rotary Blowers—These blowers have speed-delivery characteristics rather different from those of piston-type pumps. Measurements made on a two-stroke engine equipped with a Roots scavenging blower showed that the charging pressure varies nearly as the square of the speed, so that the engine acts almost as a fixed orifice. That, however, does not apply to four-stroke engines. The leakage paths in the blower also act as a fixed orifice.

Relations between the delivery of the blower on the one hand, and its speed and the pressure head against which it delivers on the other, are generally shown in graphs similar to that of Fig. 4. The ordinates of that graph represent the ratio of the blower delivery pressure (absolute) to its intake pressure. As the intake pressure usually is equal to one

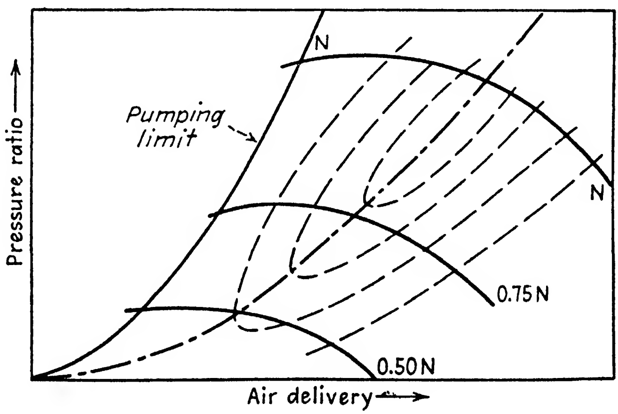


FIG. 5.—RELATION BETWEEN PRESSURE RATIO, SPEED, AND AIR DELIVERY OF CENTRIFUGAL BLOWER.

atmosphere, the pressure ratio is equal to the delivery pressure expressed in atmospheres. Abscissas represent the air delivery, which is usually expressed in cu ft of air at the prevailing atmospheric density, per minute. At any given speed the delivery decreases as the pressure ratio increases, the proportional decrease being greatest at low speeds.

Characteristics of Centrifugal Blower—Characteristic curves of a centrifugal blower with radial vanes are shown in Fig. 5. Such a blower does not operate satisfactorily at low deliveries against high pressures, as under these conditions there are likely to be interruptions in the flow, and the operation then becomes erratic. In Fig. 5 this field of unstable operation is bounded by the axis of ordinates and a parabolic "pumping-limit" line. At constant speed, within the field of stable operation, the pressure ratio at first increases slightly with increase in the delivery, and then begins to drop. When operating at points on a parabolic line from the origin through the point of maximum adiabatic efficiency, the delivery increases substantially in direct proportion to the speed, while the pressure increases as the square of the speed. The oblong loops in Fig. 5 are curves of constant adiabatic efficiency.

Intake Lines of Engine—German engineers * have devised a graphic method of determining the results which may be expected when a blower of certain characteristics is fitted to

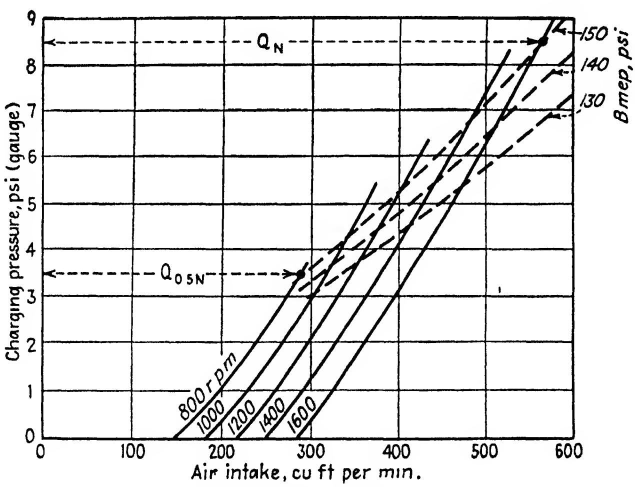


FIG 6—VARIATION OF AIR INTAKE AND BMEP WITH CHARGING PRESSURE AND SPEED OF SUPERCHARGED ENGINE.

an engine of a given design. They plot what may be called "intake lines" of the engine for different operating speeds on coordinate diagrams in which the abscissas represent the volume of air taken in by the engine in unit time, and the ordinates the charging pressure. At any given speed the engine will take in a definite quantity of air, depending on its displacement, the intake pressure, the temperature of the entering air, the sizes of intake and exhaust valves and passages, and the valve overlap. Fig. 6 shows such intake lines for an engine equipped with a turbocharger. From the diagram it can be seen that at one-half the normal speed the air intake per unit of time is one-half that at full speed,

* Supercharging of Four-Stroke Diesel Engines, by Dr. Ing. Karl Zinner, Motortechnische Zeitschrift for May-June, 1950.

but the pressure required to get the air into the engine is only about 40 per cent that required at full speed. As the amounts of air taken in by the engine are directly proportional to the speeds, the cylinder charge is the same under both conditions, and the bmeps therefore also are the same. The dashed inclined lines in the diagram are lines of constant bmep.

To determine the air intake and the charging pressure of the supercharged engine at different speeds, we superpose the characteristic curves of the blower on the intake curves of the engine, both sets of curves having the same coordinates. Such sets of superposed curves are shown in Fig. 7. There the

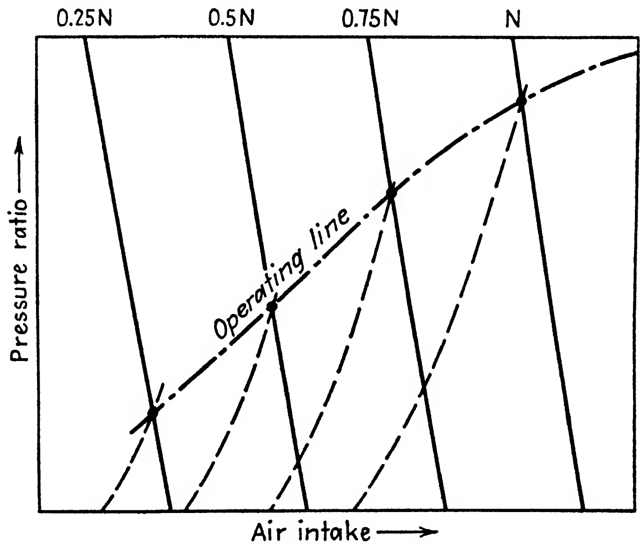


FIG. 7.—DELIVERY CURVES OF BLOWER SUPERPOSED ON "AIR-INTAKE" OR "GULP" LINES OF ENGINE.

deliveries of a centrifugal blower with straight blades are plotted for 25, 50, 75 and 100 per cent of normal speed, together with intake lines (dashed) of the engine for the same proportional speeds. Coordinates of a point of intersection of a blower-delivery curve with the engine-intake curve for the same speed represent the air intake and the charging pressure for that particular speed. The dash-dotted line through the various points of intersection, which may be called the "operating line," shows the variation of intake volume with intake pressure as the speed of the blower varies. Results obtained from a mechanically-driven blower on a given engine can be changed by altering the gear ratio between engine and blower, and the effects of such a change can be studied by means of the superposed charts of engine intake and blower delivery.

Turbochargers—A turbocharger or turbo-blower is a machine combining a gas turbine with a centrifugal blower in a common housing, with connections to both the exhaust and intake ports of the engine. Since the exhaust gases drive the turbine, use is made of what would otherwise be waste energy. However, insertion of the turbine in the exhaust line increases the exhaust back pressure slightly, and therefore reduces the shaft power of the engine. Another effect of the turbocharger is that it silences the exhaust, and in some cases obviates the need for a muffler.

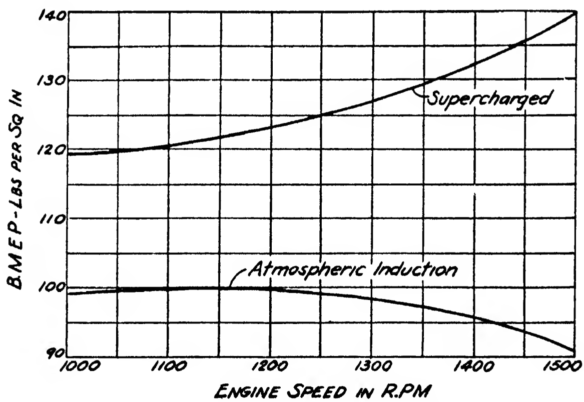


FIG. 8.—TORQUE CURVES OF NORMAL AND SUPERCHARGED ENGINES.

Characteristics of Turbochargers—With turbochargers and mechanically driven centrifugal blowers the engine torque increases with increase in engine speed, as shown in Fig. 8. An engine with such a characteristic is well suited to propulsion by means of a propeller (airplanes and motorboats) but is not suitable for use on motor vehicles with stepped mechanical transmissions. Where the gear ratio is fixed, as it is with a stepped transmission in any particular gear, the engine torque should increase when the vehicle speed is reduced by an increase in the traction resistance, as otherwise there will be a constant tendency to stall, and the driving conditions will be very awkward. The situation is different, however, with either electric or hydraulic transmission, where the effective transmission ratio changes continuously and automatically.

Among the disadvantages of the turbo-blower, aside from its high cost, are the fact that under full load the turbine blades operate at a dark-red heat and are then exposed to the corrosive influences of the exhaust gases; and that the bearing adjacent to the turbine rotor, owing to the high speed of the rotor and the high temperature to which it is exposed, is likely to be troublesome. The blades of the centrifugal blower sometimes become coated with a layer of incrustation composed of dust and oil, which may appreciably reduce the size of the air passages and the output of the engine. This trouble may be remedied by running the blower with open delivery and feeding Oakite plater's cleaner in with the air for a few minutes.

Büchi System—A system of charging by turbo-blowers developed by Dr. Alfred J. Büchi of Winterthur, Switzerland, has come into wide use in Europe, and has been adopted also in this country, chiefly in connection with engines of rather large output. In four-stroke engines thorough scavenging of the combustion chambers is combined with supercharging, to permit of high specific outputs at moderate combustion temperatures, to reduce heating effects, and to ensure a high mechanical efficiency.

Owing to the rather low efficiency of the turbo-blower, the mean pressure required to drive the turbine is approximately the same as the blower pressure, consequently it is difficult to obtain the excess pressure in the charging air that is necessary for efficient scavenging. In the Büchi system scavenging is made possible by creating strong pressure pulsations in the exhaust system. In an engine with four or more cylinders, only cylinders whose exhaust periods do not overlap are allowed to exhaust through the same manifold, and in a twelve-cylinder engine, for instance, four distinct exhaust lines, with an equal number of nozzle-ring sectors, are employed. Fig. 9 shows the pressure variations in the exhaust manifold of three cylinders of a six-cylinder engine when operating at a bmep of 146 psi. The wavy, nearly-horizontal line represents the blower pressure, while the straight vertical lines on the diagram limit periods when the exhaust and inlet valves of the same cylinder are open simultaneously and when only the exhaust valve is open. It will be seen that during the overlap of inlet and exhaust periods the exhaust pressure is much lower than the blower pressure, so that the cylinder can be thoroughly scavenged. Because of the scavenging, the cylinders are internally cooled and the volumetric efficiency is increased. Besides, as the burnt gases are completely swept from the engine, there will be more air in the cylinder and more fuel can be burned. Fig. 10 shows an Alco (American Locomotive) engine equipped with a blower of Brown-Boveri make operating on the Büchi principle.

Turbocharger for Truck Engines—Bus and truck engines equipped with turbochargers of Brown-Boveri make were placed in production in 1945 by Adolphe Saurer of Arbon, Switzerland. Such chargers had been used on large stationary and railway engines for some time, but their application to the engines of road vehicles called for the solution of a number of difficult engineering problems. At normal engine speed the speed of the charger exceeds 40,000 rpm, and the dimensions of the unit are necessarily quite small. It is fig-

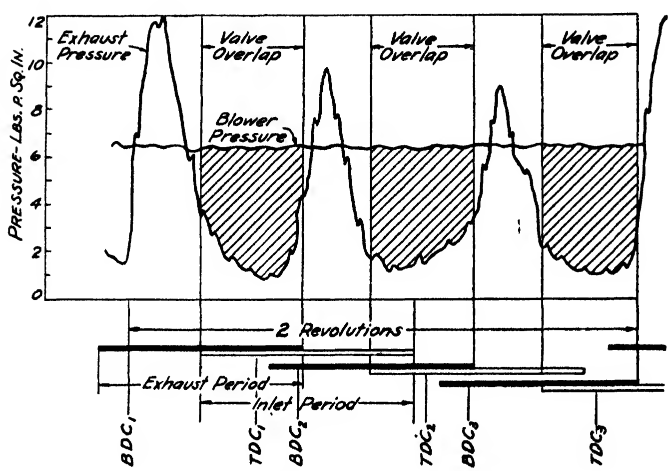


FIG. 9.—PRESSURE DIAGRAM FOR SIX-CYLINDER SUPERCHARGED ENGINE.

ured that the gain in output must be at least 30 per cent, else the cost of the installation is not warranted. To obtain this—and preferably a somewhat larger—increase in output, the leakage losses must be held down by making the clearances extremely small. Another problem concerns the torque characteristic of the supercharged engine. To ensure good "lugging" power, the torque must increase fairly rapidly as the speed decreases. Now, even with natural induction the torque of a Diesel engine usually rises rather slowly with decrease in speed, and with a supercharger there is a tendency to increase the torque more at high than at low speeds, thus further reducing the "lugging" power. This tendency must be controlled, and to that end the masked inlet valves are rotated around their axes by a relay-type governor to increase the turbulence at low and to decrease it at high speeds.

Large supercharged four-stroke engines usually are designed to have considerable valve overlap, as this facilitates the scavenging action of the blower In the Saurer engine, in which the compression chamber is in the piston and the piston crown comes within 0 040 in of the cylinder head at the end of the stroke, a large valve overlap was impossible It was undesirable also for the reason that in Switzerland most commercial vehicles are equipped with the Oetiker ex-

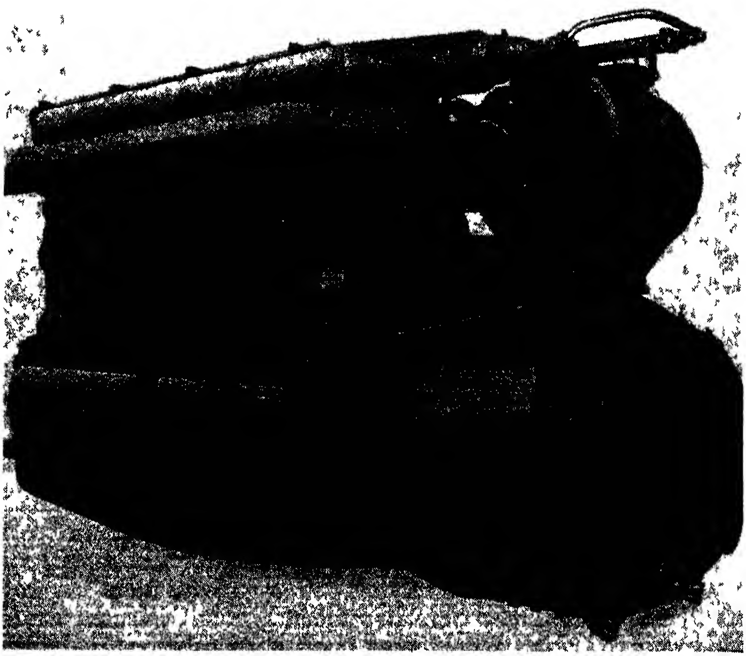


FIG 10—ALCO ENGINE EQUIPPED WITH A BROWN-BOVERI TURBOCHARGER

haust brake, which converts the engine into an air compressor while descending long, steep grades, and the effectiveness of the engine as a brake would be materially reduced by a large valve overlap.

Brown-Boveri Turbocharger—Fig. 11 is a sectional assembly view of the Brown-Boveri turbocharger. The bearing problem of such a device naturally is a difficult one, on account of both the high shaft speed and the great heat from the exhaust gases. In this turbocharger the rotor shaft is mounted in ball bearings, and the bearings are located near the ends of the shaft, instead of between the two rotors, as in some earlier designs. Provision is made for flood lubrication of the bearings, and operators are advised to renew the oil supply and thoroughly clean the oil wells every 3000 miles or every 250 hours. In the drawing *A* is the combined air cleaner and inlet silencer; *B*, the blower outlet; *C*, the impeller; *D*, the turbine runner; *E*, the turbine inlet; *F*, the oil gauge; *G*, an oil ring; *H*, the water-jacketed turbine housing; *J*, the turbine exhaust, and *K*, a heat-insulating disc.

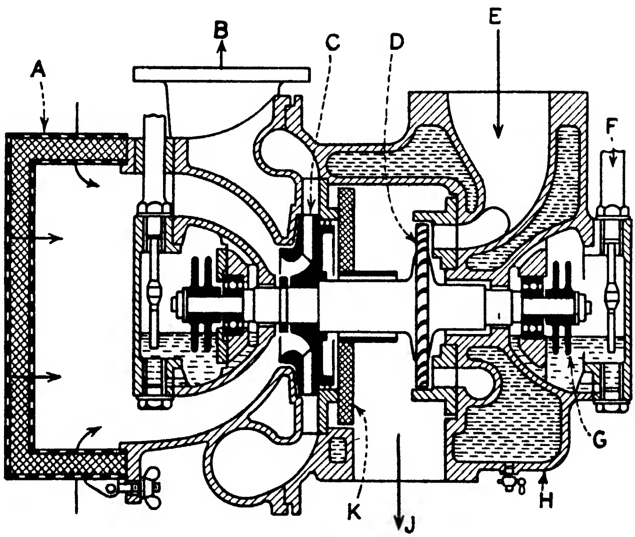


FIG. 11.—SECTIONAL VIEW OF BROWN-BOVERI TURBOCHARGER.

Comparison of Blower Types—A number of high-powered supercharged engines were developed for the German State Railways following World War II. Both mechanically-driven centrifugal blowers and turbochargers were tried on these engines, and the State Railways decided in favor of the turbocharger. Fig. 12 gives results obtained with one of these engines, a Daimler-Benz 12-cylinder 800 hp, when equipped with a mechanically-driven blower and a turbocharger, respectively. In connection with the curves of Fig. 12 it should be explained that they were obtained, not with the fuel-control lever permanently set in the "maximum" position, but adjusted to give outputs corresponding approximately to those which would be absorbed by a propeller at the different speeds. The bmep curve shows how the engine has to be "throttled" to reduce the speed from the maximum of 1400 to the minimum of 1000 rpm. These engines drive through hydraulic or electric transmissions, and they therefore operate within a comparatively narrow speed range.

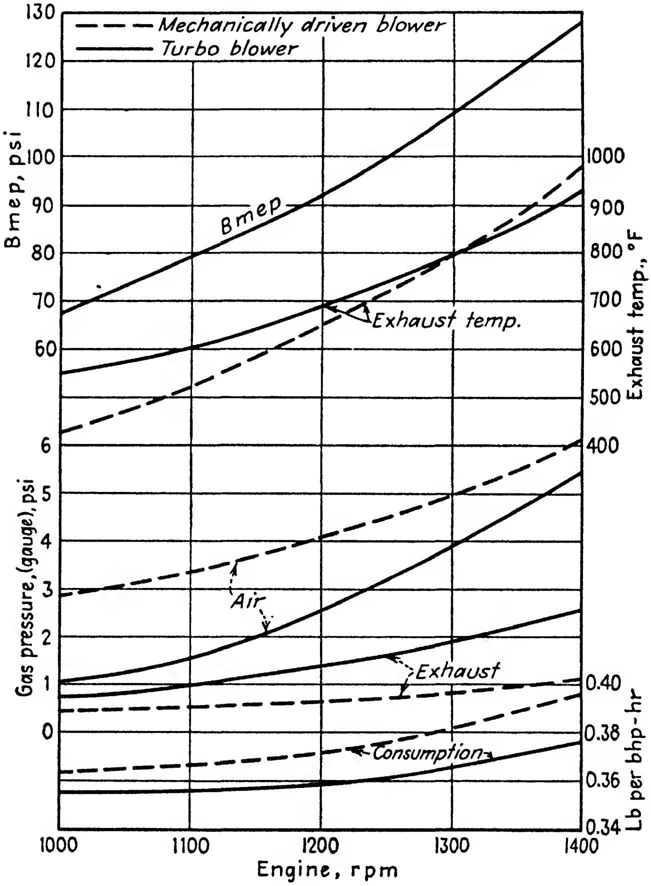


FIG. 12.—PERFORMANCE CURVES OF FOUR-STROKE ENGINE WHEN EQUIPPED WITH A MECHANICALLY DRIVEN BLOWER AND A TURBO-BLOWER, RESPECTIVELY.

With the mechanical blower the exhaust back pressure is lower throughout the speed range, while the exhaust temperature is slightly higher only at and near maximum speed, this being due to the rapid increase with speed of the power consumed by the blower. Another advantage of the mechanical blower is its far smaller weight, 110 lb as compared with 660 lb in this case. One difference between the turbocharger and the mechanically-driven blower is that they respond differently to sudden changes in engine load. The turbocharger naturally runs much faster at full engine load than when the engine is idling, because at full load both the exhaust pressure and the rate of production of exhaust gases are higher. But it takes time for the rotors to accelerate from the idling to the full-load speed, and if the load is applied suddenly the excess-air factor will be momentarily reduced and the engine may smoke during the short interval till stable conditions are reached. It has been shown, however, that if the rotors are made of light alloy, to reduce their moment of inertia, and are well supported, the drop in engine speed and the decrease in the excess-air factor persist for only a few seconds, and have no really objectionable effects. If a mechanically-driven blower is so designed that it delivers just the right amount of air at full load, it will deliver more than sufficient air when the load is suddenly reduced, but with a Diesel engine an over-supply of air is never as harmful as an undersupply.

The turbocharger shows up to best advantage in a comparison based on fuel economy. In the Daimler-Benz 800 hp engine, with the two types of supercharger proportioned to give about the same excess air, the specific consumption with the turbocharger is 5 per cent less at full load (1400 rpm) and 3 per cent less at 1000 rpm and the corresponding "propeller" load. On the average the fuel consumption is about 4 per cent less with the turbocharger, and that seems to have induced the German State Railways to adopt it.

Heat Balances for Supercharged and Other Engines—The table below gives heat-balance data for an M.A.N. six-cylinder engine of $11\frac{13}{16}$-in. bore and 15-in. stroke, which develops 690 hp at 700 rpm with atmospheric induction, and 950 hp at 700 rpm when supercharged with a turbo-blower, the specific fuel consumptions in the two cases being 0.387 and 0.376 lb per hp-hr respectively.

	Atmospheric Induction		*With Turbo-Blower*	
	Btu per Hr	*Per Cent*	*Btu per Hr*	*Per Cent*
Shaft output...	1,760,000	36.0	2,420,000	37.0
To radiator....	1,220,000	25.0	1,020,000	15.5
To oil cooler...	160,000	3.3	180,000	3.0
In exhaust.....	1,620,000	33.5	2,760,000	42.5
Radiation......	100,000	2.2	120,000	2.0

These heat balances reflect the internal cooling of the supercharged engine due to scavenging. A much larger proportion of the waste heat goes into the exhaust and a much smaller one into the cooling water.

Bearing-Load and Vibration Problems—Two problems that must be studied when contemplating the installation of a supercharger on an existing engine model are those of the effect on bearing loads and on critical speeds and torsional vibration. Both of these effects are dependent on the higher maximum combustion pressure in the supercharged engine. An investigation of the effect of supercharging on bearing loads was made by Russell Pyles of Clark Brothers Co., who found that the principal effect is a material increase in the load on connecting-rod bearings. If the resulting loading should be excessive for the standard type of bearing, the possible remedies would be an increase in the reciprocating weights, an increase in the operating speeds, and change to a bearing material of higher load-carrying capacity. The increase in the load on the center main bearing is negligible. The mean bearing load *per horse power* is less in the supercharged than in the conventional engine, and this conclusion is borne out by the finding of John Dickson of the Westinghouse company that the heat lost to the oil increased only 5 per cent when the bmep was increased 50 per cent by supercharging.

If the blower is positively driven from the forward end of the engine crankshaft, it adds materially to the polar moment of inertia of the crankshaft assembly, thereby lowering the critical speeds and increasing the chances of trouble from torsional vibration. It is therefore advisable to drive it from near the nodal point of the assembly, at the rear end of the crankshaft. If this is inconvenient, the crankshaft can be made stiffer, or the damping force of the vibration damper can be increased. Inertia characteristics of the crankshaft assembly are not affected by a turbo-blower, which has no mechanical connection with it. Where other types of blowers are used the effect on the crankshaft inertia can be eliminated by using either a magnetic or a hydraulic-drag-type coupling.

Highly Supercharged Two-Stroke Engines—Some modern medium-speed two-stroke Diesel engines are supercharged, but the charging pressure usually is quite moderate, from 3 to 8 psi gauge. The reason the charging pressure is not carried higher is that then too much of the engine power would be required to drive the blower, besides which too much energy would be thrown away in the exhaust, since the exhaust pressure increases rapidly with the charging pressure. When a two-stroke engine is supercharged, the final compression pressure remains substantially unchanged, the compression ratio of the engine being decreased as the compression effected in the blower is increased. This, of course, results in a lower expansion ratio and a lower thermal efficiency. The natural remedy would seem to be to use a turbocharger and thus recover some of the energy in the exhaust; but, unfortunately, the conventional turbocharger cannot be used with a two-stroke engine, which must be charged by outside means under all conditions, and the turbocharger cannot perform this function while the engine is being started.

Sulzer Brothers of Winterthur, Switzerland, have worked out an arrangement designed to overcome this handicap of the turbocharger. It consists in providing the engine with a positively-driven blower and discharging the exhaust through a gas turbine which is in driving connection with the engine crankshaft, so that the power of the turbine is added to that of the engine. Some kind of flexible or safety driving coupling must be employed to prevent excessive stresses due to inertia effects. An attractive aspect of this plan is that as the supercharge ratio is increased, the engine displacement required to handle a certain mass of air per minute is decreased (and there is, of course, an inclination to consider the power output proportional to the mass of air passing through the engine in unit time).

With an increase in the supercharge ratio the fraction of the engine power required to drive the blower increases, and with a charging pressure of about five atmospheres (75 psi) practically all of the engine power is consumed by the blower. In that case only the power of the turbine is available power, and the turbine, therefore, need have no driving connection with the engine. The engine and blower then constitute a gas-generating plant producing "power gas," and it would be advantageous to build the engine and blower as a free-piston unit without crankshaft. In Fig. 13 is shown a diagram of a powerplant comprising a power-gas generator of the free-piston type. The two Diesel pistons *A* in the same cylinder have two compressor pistons *B* formed integral with them and are interconnected by a synchronizing mechanism. The Diesel pistons are forced toward each other by the compressed air remaining in the compressor cylinders at the end of the out-stroke, whereby the air in the Diesel cylinder is compressed. At the end of the in-stroke, fuel is injected through nozzle *E* and burned in the highly-compressed air, and the resulting increase in pressure forces the pistons outward and compresses the air drawn into the compressor cylinders during the previous stroke. This air serves to scavenge and charge the Diesel cylinder. The exhaust gas from the

Diesel engine—which is referred to as "power gas"—passes through pipe *H* to exhaust turbine *I* which drives the electric generator *K*. This plant is designed for a Diesel charging pressure of 6 atmospheres, and as it takes more power than is generated in the Diesel cylinder to compress the air from one to six atmospheres pressure, a precompressor *M* is provided

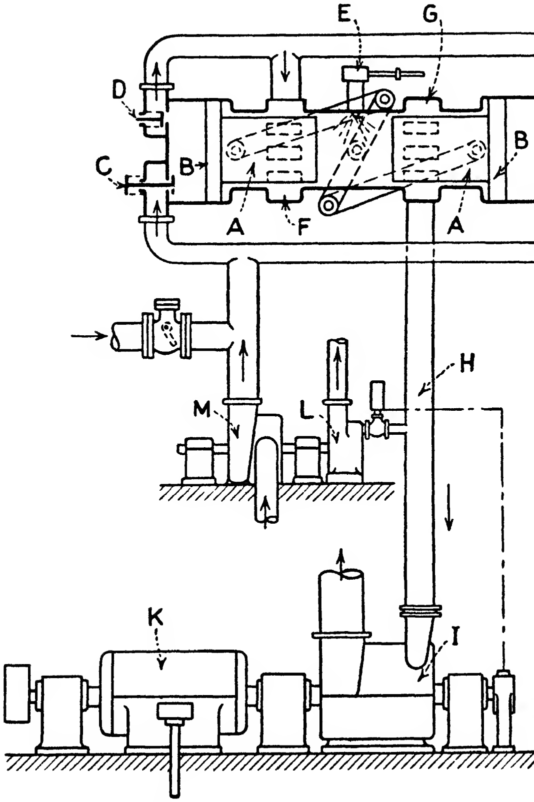


FIG. 13.—DIAGRAM OF SULZER "POWER-GAS" SYSTEM, COMPRISING A FREE-PISTON-TYPE AIR COMPRESSOR, A GAS TURBINE, A GENERATOR, AN AUXILIARY GAS TURBINE, AND A BLOWER.

and is driven by an auxiliary gas turbine *L*, which also is supplied with "power gas" from the Diesel engine.

Sulzer Brothers have built a number of experimental engines using a high supercharge ratio. These engines are of the double-piston type (similar to the Junkers). A four-cylinder engine of 7.48-in. bore and 2 x 11.81-in. stroke, operating at a charging pressure of 28 psi, developed 1370 bhp at 750 rpm (one-hour rating which corresponds to a bmep of 174 psi). A six-cylinder engine of 7.09-in. bore and 2 x 8.87-in. stroke, also charged to 28 psi, has a one-hour rating of 1560 bhp at 850 rpm.

When an engine is highly supercharged, the mean pressure and temperature during the power stroke are materially greater than in an engine without supercharge, and trouble from excessive heat stresses might be anticipated. This is guarded against by limiting the amount of fuel injected. Experience with various experimental engines is said to have shown that with a charging pressure of 28 psi it is safe to go to a bmep of 170 psi; with a charging pressure of 42.5 psi, to 210 psi, and with a charging pressure of 85 psi, to 225 psi bmep. It is claimed that when supercharging to 28 psi, the specific fuel consumption was below 0.352 lb per bhp-hr.

CHAPTER XV

Some Details of Engine Design

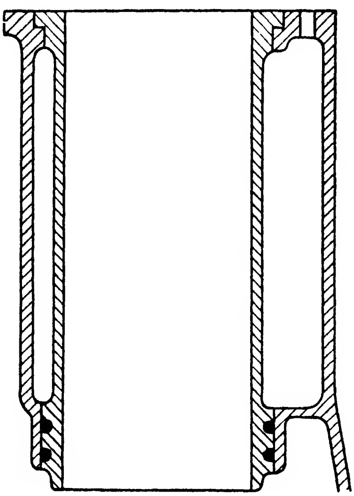
In a general way the design of parts for high-speed Diesel engines follows closely that of the corresponding parts for heavy-duty carburetor-type engines, which is dealt with in the author's book on High-Speed Combustion Engines. For a study of the general principles underlying the design of cylinder blocks, pistons, connecting rods, crankshafts, valves and valve gearing, and of lubricating and cooling systems, the reader is referred to that book. However, owing to the considerably higher peak pressures in the cylinders of Diesel engines, there is a tendency to deviate from established gasoline-engine practice in certain particulars, and in this Chapter will be discussed some design practices which are either peculiar to the Diesel engine or else much more common in the Diesel than in the gasoline-engine field.

Cylinder Liners—Removable cylinder liners have long been used in tractor engines of the carburetor type, owing to the rapid wear of cylinder bores sometimes caused by the entrance of abrasive dust into the cylinders in field operations; more recently they have come into use also in truck and bus engines, but the use of such removable liners is most common in high-speed Diesels. Under similar service conditions, the rate of wear on the cylinder bore is always greater in Diesel engines. This is generally ascribed to the considerably higher peak pressures of combustion. The gases of combustion get behind the topmost piston ring and force it outward against the cylinder wall with great pressure, and this causes rather rapid wear at the upper end of that portion of the cylinder bore which the topmost piston ring slides over in its reciprocations.

Fig. 1 shows a cylinder with a "wet" liner; that is, a liner in direct contact with the cooling water. The joint between the block and the lower end of the liner is sealed by means of synthetic rubber rings, while that at the upper end is sealed by the cylinder-head gasket. Where "wet" liners are used, the walls of the block—which correspond to the jacket walls of a block without liners—are cast of heavier section. Fig. 2

shows a cylinder with a "dry" liner. In 1946 "wet" and "dry" liners were used in approximately the same numbers of models. By 1952 the proportion had changed to 60 per cent with "wet" liners and 36 per cent with "dry" liners, the remaining 4 per cent having no liners. With two-stroke engines with ports in the cylinder walls "wet" liners present a difficult problem. A Graef & Stift (Austrian) eight-cylinder V engine features this construction, the liner being provided with a flange at the port level which is a close fit in a bore of the block.

The liners generally are cast of nickel-chromium iron containing 1.80-2.20 per cent of nickel and 0.55-0.75 per cent of chromium. While ordinary gray iron as used in cylinder



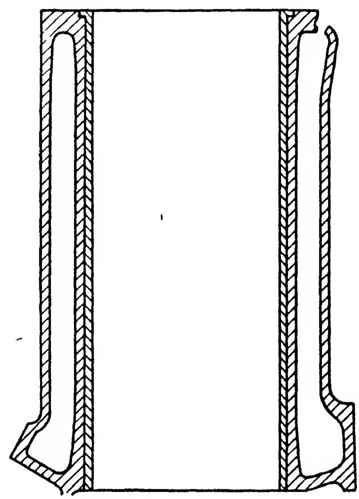


FIG. 1 (*Left*).—CYLINDER WITH "WET" LINER.
FIG. 2 (*Right*).—CYLINDER WITH "DRY" LINER.

castings has a Brinell hardness of 230-250, these alloy iron liners when properly heat-treated have a Brinell hardness of slightly over 500, and their life between reconditionings is substantially three times as great as that of an ordinary gray-iron cylinder wall. With these alloy cast-iron liners it is not wear of the liner but loss of sealing effect of the piston rings, due either to wear of the rings and their grooves or to loss of tension of the rings, that first calls for reconditioning. Therefore, when such engines first begin to consume oil at an excessive rate, or to show excessive blow-by, the trouble usually can be cured by installing new, standard-size piston rings.

Heat treatment of the alloy cast iron liners increases not only the hardness of the liner, but also the tensile strength of the material. Thus a cast iron containing 3.10-3.40 per cent total carbon, 0.75-0.90 per cent combined carbon, 0.55-0.75 per cent manganese, 0.20 per cent max. phosphorus, 0.10 per cent max. sulphur, 1.90-2.10 per cent silicon and the proportions of nickel and chromium given above, will show a tensile strength of 45,000-56,000 psi when hardened and then tempered for one hour at 600 F. The hardening operation consists in heating in a furnace to 1540 F-1560 F for 30 to 40 minutes and then quenching in still oil.

These liners are machined for an interference fit in the bores of the cylinder block, the interference allowance being of the order of 0.0005 in. per inch of bore. Selective assembly is generally found advisable. Liners are forced into place in the bores of the block in a press, with the block previously heated in water at near the boiling point and the liner at atmospheric temperature.

One difference between engines with hard cylinder liners and those having wearing surfaces of ordinary gray iron is that with the former a much longer time is required for the piston rings to wear in, or to properly seat themselves. This makes the oil consumption rather high at first, and the mileage per quart of oil consumed will increase for a considerable time. On the other hand, the engines can be operated under normal load right from the beginning, and need not be "nursed" through a running-in period.

"Through" Bolts—In the conventional internal-combustion engine the stresses due to the gas pressure are taken by the cylinder block and crankcase, these stresses being largely in the form of tension. As these parts ordinarily are castings with unfinished surfaces, one can never be sure regarding the extent of the areas over which the gas-pressure loads are distributed, and to be on the safe side, wall thicknesses therefore have to be made relatively heavy. For this reason the use of "through" bolts or "through" studs, tieing together the cylinder head, cylinder block, crankcase (if separate) and main-bearing caps has found some favor with Diesel-engine designers. The gas pressure exerts itself against the cylinder head in one direction, and—through the piston, connecting rod and crankshaft—on the main-bearing caps in the opposite direction, and if cylinder head and bearing caps are tied together by bolts or studs extending through both, the engine structure is relieved of tensile stresses due to these pressures. Bolts or studs and nuts of alloy steel are generally employed for this purpose. This practice, of course, is particularly applicable to engines with removable cylinder liners, in which the tensile stresses due to gas pressures otherwise would have to be taken entirely on the water-jacket walls, and it is practically imperative in engines with cylinder blocks cast of light alloys. Fig. 3 shows an arrangement of "through" studs which has been used in large Hercules engines. The studs are provided with eccentric heads that enter corresponding recesses at the top of the separate crankcase. These eccentric heads prevent the studs from turning and permit of removing the cylinder head without disturbing the main bearings. Also, tightening of the cylinder-head nuts does not affect the adjustment of the main bearings.

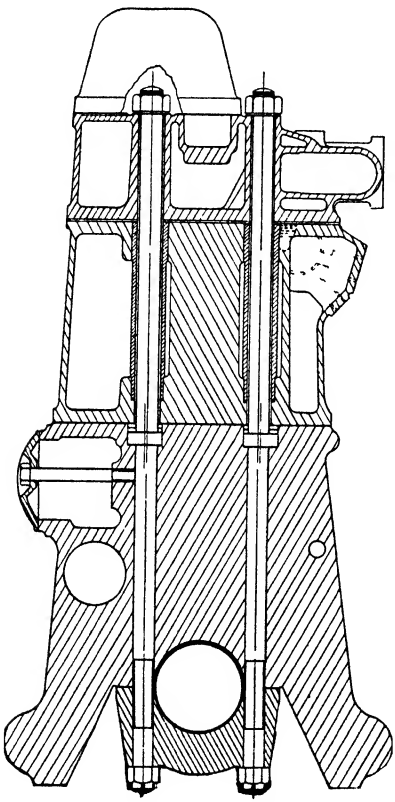


FIG. 3.—"THROUGH" BOLTS

Welded Steel Engine Blocks—In order to keep down the specific weight of high-speed engines of large output, as used mainly for marine and railroad work, recourse has been had to the use of alloy steel as a structural material, members of the engine block or frame being cut from plate steel by means of the oxygen cutting torch and welded together with electric welders. Such welded steel-frame engines have been built in this country, England and Germany. In this country engine blocks of this type are being produced by Lukenweld, Inc., of Coatesville, Pa.

In the fabrication of these welded engine blocks, use is made of a low-carbon alloy steel having an endurance limit of 50,000 psi. (The endurance limit is the maximum stress to which a part subjected to alternating load can be carried an infinite number of times without failure.) In order to eliminate any injurious effects of the welding heat on the physical properties of the base metal adjacent to the weld, and also to remove residual stresses set up by the cooling of the unequally-heated structure, the whole structure is annealed after the welding operations on it have been completed. It has been found that without a thorough annealing of the completed block, it is entirely impossible to keep it from warping and twisting during machining operations. There is a tendency to stress concentration at the edges of the welds, and considerable skill on the part of the welder is said to be necessary to prevent weak spots in the structure at these points.

Fig. 4 shows one of the main-bearing girders of a twelve-cylinder V engine, cut from 4-in. plate steel. To utilize the metal to best advantage, it would be necessary to subject these

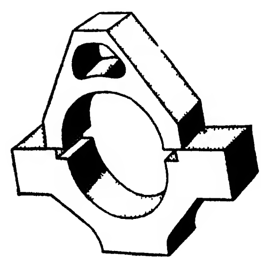


FIG. 5.—FLAME-CUT BEARING AND CAP.

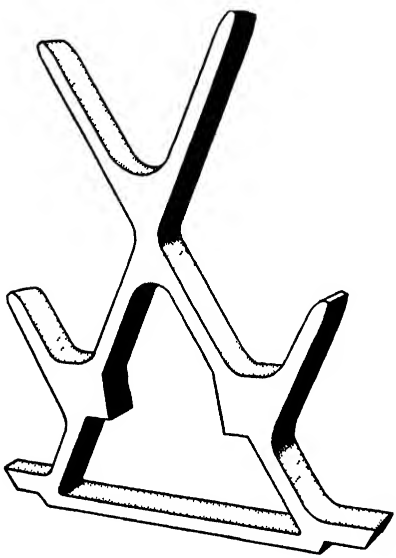


FIG. 4.—FLAME-CUT MAIN-BEARING GIRDER FOR V ENGINE.

members to simple tension, and in an in-line engine this could be achieved by merely extending the upper arms of the bearing girder all the way to the top deck. In a V engine this is impossible, on account of the offset between oppositely located cylinders, which in this particular case amounts to 3 in. So "transition" plates are used which extend the whole length of the engine. Each top deck is butt-welded to two such transition plates, and the transition plates in turn are welded to the main-bearing girders. These transition joints between the girders and plates are of rather peculiar shape, and to guard against imperfections in the welds, all of them are inspected by means of X-rays, and the welds are also carefully examined for undercuts and discontinuities. Fig. 6 shows the block in the state of assembly at which these inspections are made. All of the welds subjected to gas-pressure forces are completed, but considerable welding still remains to be done.

Next the inner deck, which supports the lower ends of the cylinders, is welded in place, and side plates and stiffening ribs are added. It will be noted that there are large hand holes in the lower side plates, which give access to the connecting-rod bearings. There are similar smaller hand holes in the upper side plates, through which the piston rings can be inspected as they pass the transfer ports (this being a two-stroke engine). To prevent these side plates from drumming in operation, the edges of the hand holes are flanged, which

FIG. 6.—ENGINE FRAME AT STAGE WHERE WELDS ARE INSPECTED.

is said to stiffen them materially. Fig. 7 shows the engine block completed and ready to leave the welding shop. The oil pan, which is shown in place, also is of welded construction and is provided with a heavy top plate forming a tie for the bottom legs of the main-bearing girders and ensuring the requisite lateral stiffness close to the level of the crankshaft axis, where lateral reciprocating forces due to the angularity of the connecting rods must be taken care of. The main-bearing caps also are cut from steel plate and are of the form shown in Fig. 5.

The engine block represented by Fig. 7, which is for a twelve-cylinder 1000-hp engine, weighs about 2.6 lb per hp, and the entire engine as mounted on the test block weighed 10 lb per hp.

Steel-Barrel Company's Design—In England, welded steel engine frames have been built by the Steel-Barrel Co., Ltd., of Uxbridge, from designs of C. H. Stevens, and chiefly used in marine engines. The features of the Stevens design may be seen from Figs. 8 and 9. All gas-pressure stresses are taken up by a series of main-frame plates, of which there is one on each side of each cylinder. They are so arranged that they act as a sling around the cylinder-head seats and the crankshaft. An elevation of one of these frame plates is shown at *A* in Fig. 8. It is cut from plate steel and is of such

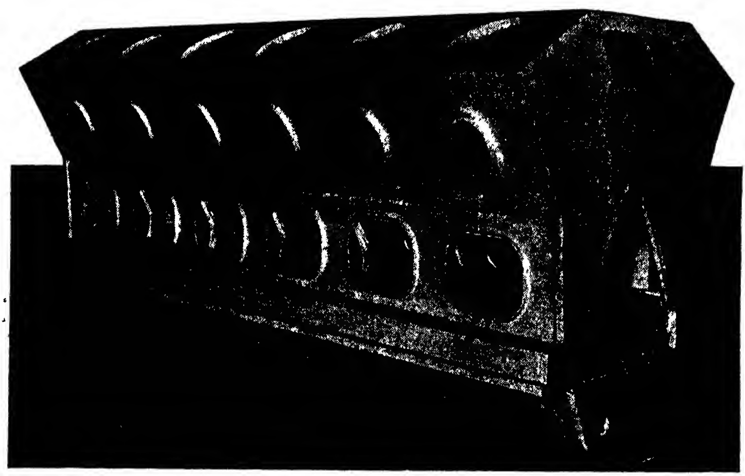


Fig 7—Welded Engine Frame Completed.

shape that its contour conforms to the end elevation of the engine. A rectangular opening is cut in the upper part of the plate, to permit of the insertion of two horizontal plates extending the whole length of the engine, one at the top and the other at the bottom of the opening. These two plates serve as top and bottom supports for the cylinder liners. *B* in Fig. 8 illustrates the way in which these horizontal plates are inserted. They have slots cut in their sides, as indicated at *C* in Fig. 8, to accommodate the tension members.

There is a second large opening in the lower part of the tension members, for the insertion of the crankshaft bearings. The series of vertical tension members are connected by what is referred to as a wrapper plate, which extends the whole length of the engine on both sides and may be provided with hand holes as required by the particular design. A short annulus is welded to the lower deck to give a fairly long guide for the cylinder liner, which latter is provided with a flange at its upper end that is sunk in a counterbore in the top plate, and the wrapper plate then serves as the water-jacket wall. If the cylinders are cast with integral water jackets, the lower horizontal plate or lower deck is unnecessary, and the tension member can then be given the form shown at *D* in Fig. 8.

All of the various members of the whole structure are united by electric welding, and an interesting feature of the design is that none of the welds have to take care of gas-

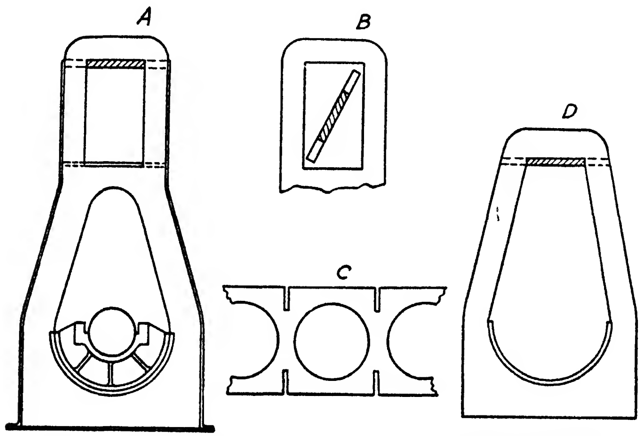


FIG. 8.—ELEMENTS OF STEVENS WELDED ENGINE FRAME.

pressure forces. These forces are taken directly on the top and bottom girders of the unwelded tension members.

A six-cylinder engine block of the general design described in the foregoing paragraphs is shown in Fig. 9. Bosses for the cylinder-head studs are welded to the under side of the top deck, and seats for the main bearings, of circular-arc shape, are welded in place on the tension members and suitably supported by stiffening ribs. In the case of the engine represented by the illustration, the specific weight was reduced from 48 to 27 lb per hp, by the use of steel in place of cast-iron construction.

Cylinder-Head Seals—In Diesel engines with their high combustion pressures the problem of producing a dependable seal between the cylinder head and block always has been a difficult one. Some early engines with individual cylinders and heads even had tongued-and-grooved joints between these parts, which must have been rather expensive to produce. The ordinary laminated gaskets used in most gasoline engines to seal the cylinders, water passages, and oil holes, are not entirely satisfactory in Diesel engines, and gaskets of sheet copper or soft iron are extensively used in the latter. Detroit Diesel Engine Division of General Motors Corporation in 1951 introduced on its Type 71 engines a new method of "individual" or "metal-to-metal head-and-block" sealing, which is illustrated in Fig. 10. The compression gaskets, consisting of laminated terne-plate rings, take the major part of the pressure produced by drawing up the cylinder-head nuts.

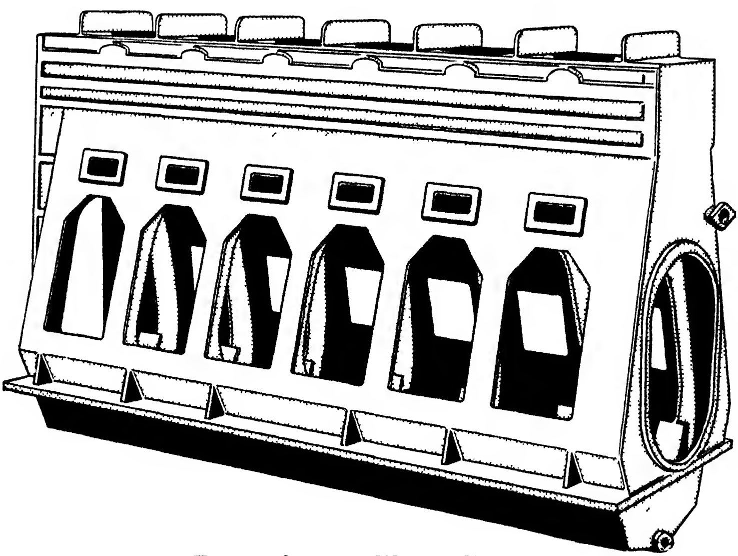


FIG. 9.—STEVENS WELDED ENGINE.

By tightening these nuts to the recommended wrench torque, these gaskets are compressed from the original 0.085-0.100 in. thickness to 0.057-0.061 in. A small skirt on the gasket is made to enter a recess between liner and cylinder bore, to ensure correct positioning of the gasket relative to the liner. Immediately adjacent to the compression gasket there is a clearance of 0.007-0.014 in., and when the cylinder-head nuts are drawn up in accordance with instructions, positive metal-to-metal contact between head, liner, and gasket is assured.

Water and oil passages are sealed by individual sealing rings of synthetic rubber. Around each drilled water or oil passage a recess of liberal dimensions is counterbored into the block. The washer-like rubber seals are placed in these counterbores and are compressed when the cylinder-head nuts are tightened. As only light pressures have to be contained, the compression of these seals is not critical. A long endless seal fits into a rectangular, milled groove near the edge of the block and seals the camshaft cavities from the outside.

Pistons—Pistons in general are made with heavier sections at the top end than those of gasoline engines of similar bore, to keep down the piston-head temperature. This is the more necessary the larger the bore and the higher the engine speed, and, of course, the heat problem is more difficult in two-stroke than in four-stroke engines, on account of the

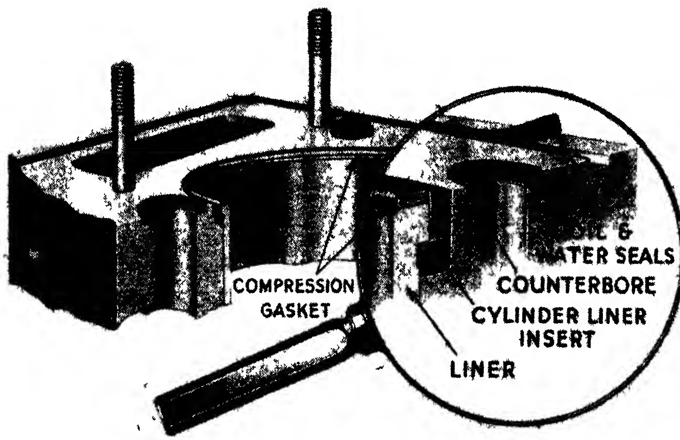


FIG 10—CYLINDER-HEAD GASKET OIL AND WATER SEALS OF G M SERIES 71 ENGINE.

greater frequency of explosions in the case of the former. Excessive piston-head temperatures result in sticking of the piston rings in their grooves, due to carbonizing of the oil in the grooves. A stuck piston ring is of no value as a seal for the piston, and if more than one ring are stuck, excessive blow-by takes place, and the entire piston is likely to be ruined by overheating in short order. Thickening the walls of the piston increases its weight and inertia, thereby increasing the bearing loads, but this cannot well be avoided. Diesel engine pistons of aluminum-silicon alloy (Lo-Ex), together with the piston pin and rings, average in weight as follows for different diameters: 4-in. bore, 4 lb; 4.5 in., 5.5 lb; 5 in., 7 lb; 5.5 in., 8.5 lb.

In designing pistons for high-speed Diesel engines it is the prevailing practice to make the top land quite wide, which tends to reduce the temperature of the top ring by moving it further from the surface directly exposed to the heat of combustion, and at the same time makes it more difficult for the gases of combustion to get behind this ring. A practice that has been found useful by some manufacturers as a means to improve the seal, consists in leaving the top ring groove empty, or, in other words, providing a double land above the topmost piston ring, separated by a groove approximating a ring groove in shape. This, apparently, has an effect similar to that of the so-called labyrinth packing widely used in steam engineering. When cast-iron pistons are used in engines of fairly large bore, it is usual to form a closed chamber

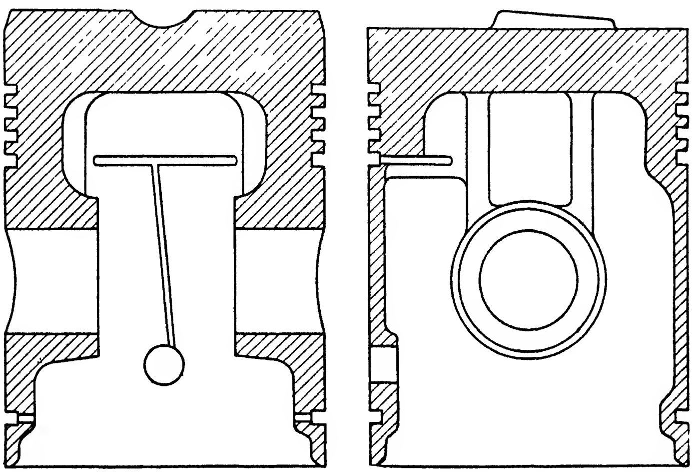


FIG. 11.—TWO SECTIONS OF AN ALUMINUM-ALLOY DIESEL PISTON.

at the upper end, so that oil thrown off the crank arms cannot come in contact with the under side of the piston head, where it would be likely to carbonize or "coke," on account of the high temperature of the head.

Because of the higher gas pressures in Diesel engines, a greater number of piston rings is generally used. Gasoline engines, as a rule, have either three or four rings, of which either one or two are scraper rings, but in high-speed Diesel engines the number of rings per piston is usually either five or six. Three or four are compression rings and one or two oil rings.

Fig. 11 shows two sectional views of the aluminum-alloy piston of a Lanova-type engine. This piston is provided with a T-slot to control its heat expansion. To make it possible to reduce the compression space to the volume required, small ''displacers'' are cast on the piston crown, which enter the lobes of the combustion chamber when the piston completes its up-stroke.

Piston-pin diameters for Diesel engines are made larger in proportion to the bore than in carburetor engines, to allow for the higher combustion pressures. As the combustion pressure determines the maximum bearing load, the ratio of pin diameter to piston diameter should vary with it. Piston pins always are of tubular form, and it is important that the wall thickness be adequate, as otherwise the pin will deform under the shock load to which it is subjected. The following proportions have given satisfactory results in service:

Max. Comb. Press., Psi	*Ratio of Pin O.D. to Bore*	*Ratio of Pin I.D. to Bore*
1000	0.41	0.250
900	0.37	0.225
800	0.34	0.200
700	0.29	0.175

Needle Bearings for Piston Pins—In some cases needle bearings have been used on the piston pins. These are essentially roller bearings employing rollers of quite small diameter and without a cage holding adjacent rollers apart. The action of these needle bearings is claimed to be quite different from that of conventional roller bearings, however, in that whereas in the roller bearing the rollers have a continuous rolling motion, in the needle bearing the needles turn around their axes only when the load is so high as to squeeze out the oil film between the needles and their races. Under other conditions the rollers, while partaking in the relative motion between their inner and outer races, do not turn around their axes.

These needle bearings naturally require hardened inner and outer races. As the piston pins are always case-hardened and ground, they will serve as inner races, but for an outer race a case-hardened or oil-hardened bushing must be forced into the small-end boss. A design of needle-type piston-pin bearing due to The Bantam Ball Bearing Co. is shown in Fig. 12. The outer race is somewhat shorter than the boss of the small end, so as to leave room at each end for a hardened thrust washer. In another design the thrust washers are placed between the small end of the connecting rod and the piston bosses. This makes the entire width of the small-end boss available for bearing purposes, and two series of needles are used, separated by a spacer ring at the center.

Owing to the extreme gas pressures and the high inertia loads in Diesel engines, it is necessary to make all parts subjected to these forces as rigid as possible, considering weight limitations. The eyes at opposite ends of the connecting rods should be made of such form that their distortion, or their deviation from the true cylindrical form, under the effects of the forces acting on them will be a minimum. The small end is materially stiffened on its under side by its junction with the shank, and as there is no corresponding stiffening effect on the upper side, it has been suggested that the bore

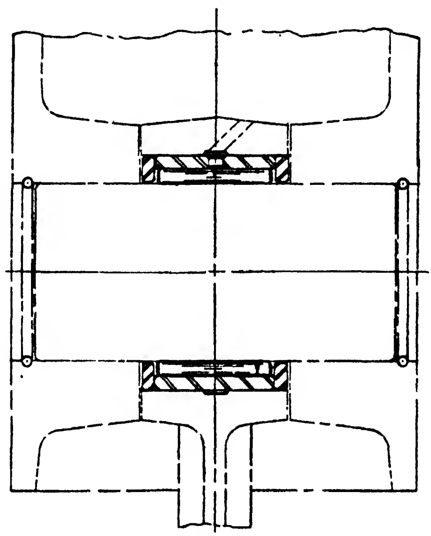


FIG. 12.—NEEDLE BEARING ON PISTON PIN.

of the small end be made slightly eccentric, so that the wall will be thickest on top or on the side opposite the junction with the shank.

Big-End Bearings—This same reasoning applies to the big end. Here the cap is provided with a substantial stiffening rib between the bolt lugs. The bearing pressures are greatest in the upper half of the big-end bearing, which takes the load due to gas pressure, and during the first few years of high-speed Diesel engines a good deal of trouble was experienced from the babbitt on this part of the big-end bearing cracking and eventually flaking off. This was evidently due to excessive pressure on this part of the bearing surface, directly underneath the junction of the shank with the connecting-rod head. To reduce stress concentration at this point, it is advisable to increase the section of the shank as it approaches the head to the full width and length of the head. Many manufacturers have overcome trouble at this point by the use of copper-lead instead of babbitt big-end bearings, while one uses a duralumin half bushing for the part of the big-end bearing which takes loads due to gas pressures. Both of these materials retain their hardness at much higher temperatures than babbitt, and therefore are able to stand much higher peak loads. However, both are harder than babbitt also at normal temperatures, and to prevent excessive wear on journals where these bearing metals are used, the former must be hardened, which adds to the cost.

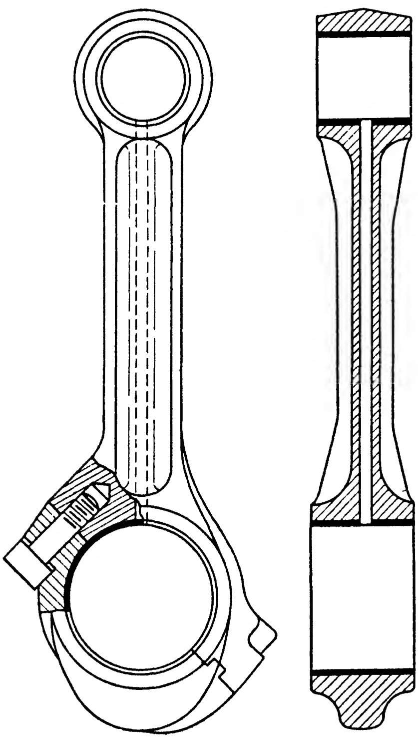


FIG. 13.—DIESEL CONNECTING ROD.

Owing to the very rugged crankshafts used in Diesel engines it is usually difficult to so design the connecting-rod big end that the rod can be withdrawn through the cylinder. Fig. 13 shows a rod which is split at an angle of 45 deg to make this possible. The joint between rod and cap is tongued and grooved to hold the two parts rigidly in alignment. Similar results are obtained by fitting dowel bushings around the two connecting-rod bolts. Center-to-center lengths of connecting rods for high-speed Diesel engines average close to twice the length of stroke.

Because of the high combustion pressures in Diesel engines and the risk of trouble from torsional vibration, crankshafts are made considerably more rugged than those for carburetor engines of the same cylinder dimensions. In six-cylinder engines the main bearings usually have a diameter equal to 90 per cent of the cylinder bore, and in one eight-cylinder engine they are even equal to the bore. Crankpins are made somewhat smaller in diameter than main bearings, in order to keep down the centrifugal forces on the connecting-rod big end (and resulting bearing loads). On the average, crankpin diameters are equal to about two-thirds of the cylinder bore.

The risk of injury to big-end and main bearings can be lessened by keeping down the temperature of the oil in the crankcase under severe operating conditions. One means to this end consists in the provision of an oil cooler, a heat exchanger through which the oil is passed on its way from the pressure pump to the bearings and the water on its way from the circulating pump to the jackets. A number of American high-speed Diesel engines are regularly equipped with such oil coolers or heat interchangers, which have the additional advantage that, when starting up from cold, they raise the temperature of the lubricating oil more rapidly, thus tending to prevent injury to the bearing surfaces before an adequate oil film can form on them.

Valves and Timing—All four-stroke Diesel engines have the valves in the cylinder head, and in proportioning the valves the best possible use must be made of the available space. Inlet valves always are given a larger diameter than the exhaust valves. The largest possible inlet-valve capacity is needed (in engines with natural induction) for maximum engine output. Smaller exhaust valves may slightly increase the exhaust back pressure, but that does not materially affect the engine performance, because at the end of the exhaust stroke the combustion-chamber volume is very small. In Hercules Diesel engines the exhaust-port diameter is only about 70 per cent of the inlet-port diameter. Engines with cylinders of large size sometimes are provided with four and even six valves in the cylinder head, arranged symmetrically around the injection unit.

An average valve timing for a four-stroke high-speed Diesel engine with natural induction is as follows:

Inlet opens 5° ahead of top center
Inlet closes 35° past bottom center
Exhaust opens 45° ahead of bottom center
Exhaust closes 10° past top center

CHAPTER XVI

Methods of Starting Diesel Engines

Starting of Diesels involves greater difficulty than starting gasoline engines. If the atmospheric temperature is very low, it is hard to generate enough heat by compression to ignite the fuel when injected into the cold engine. The maximum temperature produced in the cylinder when the engine is being cranked or motored over depends not only on the temperature of the air as it enters the engine, and the temperature of the cylinder walls, but also on the speed at which the engine is being cranked. The torque required to turn a Diesel engine over against compression is about twice as great as that required by a gasoline engine of equal displacement. To make sure that the compression temperature will be sufficiently high to produce ignition even under the most unfavorable conditions, the engine must be cranked at a comparatively high speed. As both the torque and the cranking speed must be high, the starter must develop high power, and if an electric starter of the type used for automobile engines is employed, its capacity and that of the battery from which it draws its current must be several times as great as in the case of a gasoline engine of equal displacement. This naturally makes electrical starting equipment rather expensive, the cost being particularly onerous in applications where electrical generating equipment is not needed for other purposes.

Compression Pressures and Temperatures—In Fig. 1 are shown curves of compression pressure based upon tests made on an engine having $4\frac{1}{2}$ by $6\frac{3}{4}$-in. cylinders. Of course, ignition is most directly dependent on the temperature of compression, which is related to the compression pressure. The temperature at the end of compression for any initial air temperature (in the cylinder at the beginning of compression), any compression ratio, and any engine speed can be calculated from the chart with a fair degree of accuracy. Use is made of two thermodynamic equations which express the relation between the compression ratio on the one hand and the ratios between the final and initial absolute pressures and temperatures, respectively, on the other hand. These equations are

$$P_2/P_1 = r^n$$

and

$$T_2/T_1 = r^{n-1},$$

where P_1 and P_2 are the pressures and T_1 and T_2 the temperatures at the beginning and end of compression, respec-

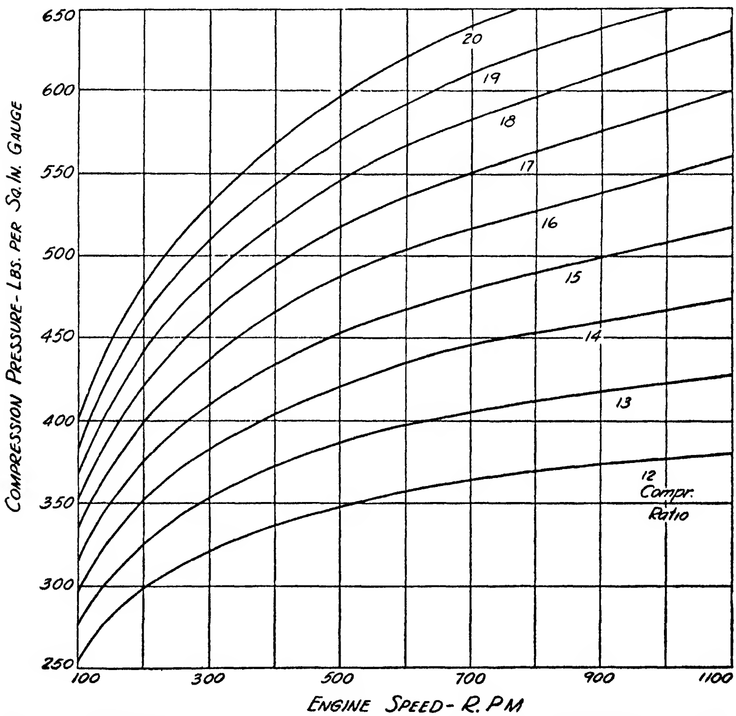


FIG. 1.—VARIATION OF COMPRESSION PRESSURE WITH COMPRESSION RATIO AND ENGINE SPEED.

tively; r is the compression ratio and n, the so-called polytropic exponent.

From the chart we find that with a compression ratio of 16, if the engine is being cranked at 200 rpm, the compression pressure will be 400 psi gauge or 415 psi abs. Substituting in the first of the above equations,

$$415/14.5 = 16^n.$$

Taking logarithms,

$$\log 415 - \log 14.5 = n \log 16,$$

from which it follows that

$$n = \frac{2.6180 - 1.1614}{1.204} = 1.21.$$

When a cold engine is being cranked the initial temperature is equal to the ambient temperature, and we will assume this to be 32 F or 492 deg abs. Then, by making use of the second equation, we find that

$$T_2 = 492 \times 16^{0.21} = 882 \text{ deg abs or } 422 \text{ F}.$$

This temperature is too low to ignite ordinary Diesel fuels. However, cranking of the engine generates heat, both by compressing the air and by overcoming the friction of the moving parts, and a good deal of this heat is absorbed by the cylinder walls. As the temperature of the walls rises, some heat is given up to the entering air, hence the initial temperature of each succeeding charge is slightly higher. Compression temperatures rise correspondingly, and eventually ignition may occur.

When the engine is operating under normal conditions the compression temperature, of course, is much higher. The engine considered in the foregoing, when running at 1000 rpm will develop a compression pressure of 550 psi gauge or 565 psi abs. The polytropic exponent then is

$$n = \frac{\log 565 - \log 14}{\log 16} = 1.34.$$

In normal operation the incoming charge is heated by contact with the intake passages and the cylinder walls, and under full load its temperature at the beginning of compression will be of the order of 200 F or 660 deg abs. The temperature of compression then will be

$$T_2 = (660 \times 16^{0.34}) - 460 = 1234 \text{ F}.$$

Starting Torque Required—For engines of a given number of cylinders and generally similar design, the cranking torque under given operating conditions varies substantially as the piston displacement. Much of the starting torque, especially in cold weather, is required to shear oil films. The shearing force needed is directly proportional to the area of the films, and it also varies with the cranking speed. It is greatest at the very start, and is then known as the breakaway torque. With increase in speed the cranking torque at first decreases, then reaches a minimum value, and thereafter increases again. The point of minimum cranking torque evidently corresponds to the change-over from boundary to full fluid lubrication, for which latter the friction coefficient increases with the speed.

The actual viscosity of the oil in the film depends on both the viscosity grade of the lubricant used and on the temperature of the oil. How the cranking torque varies with the ambient temperature 5 seconds after the breakaway is shown in Fig. 2, which is taken from an S.A.E. paper by H. L. Knudsen. The data were obtained with a Cummins six-cyl-

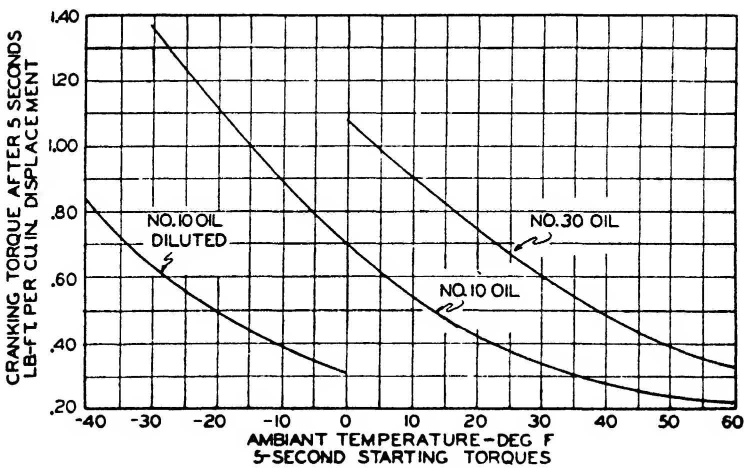


Fig. 2.—Variation of Five-Second Cranking Torque with Ambient Temperature.

inder super-charged engine of 672 cu in. displacement. Oils of three viscosity grades were used in the experiments, namely, an S.A.E. 30, an S.A.E. 10, and an S.A.E. 10 diluted with 30 parts of kerosene to 70 parts of oil. Mr. Knudsen found that when starting a cold engine only a small fraction of the total cranking power is consumed in overcoming compression. For instance, to crank the engine referred to at 200 rpm with an ambient temperature of minus 20 F when lubricated with No. 10 oil required about 12.5 hp with the compression relieved, and only 2 hp more with the compression on.

When an engine is being cranked, the torque, which is at its maximum during the breakaway, gradually decreases and usually reaches a steady value after about 5 minutes. Fig. 3, which also is from Mr. Knudsen's paper, shows the relation between the breakaway torque and the torque after 5 minutes of cranking, for two different grades of oil. From Figs. 2 and 3 it can be seen that a very thin oil in the crankcase would be of great help in starting a Diesel engine in extremely cold weather. However, a lubricant such as the diluted oil could not be used with safety in normal operation even in cold weather. If such an oil were to be used for starting, it would have to be introduced into the engine just before it was shut down after a run, and immediately drained

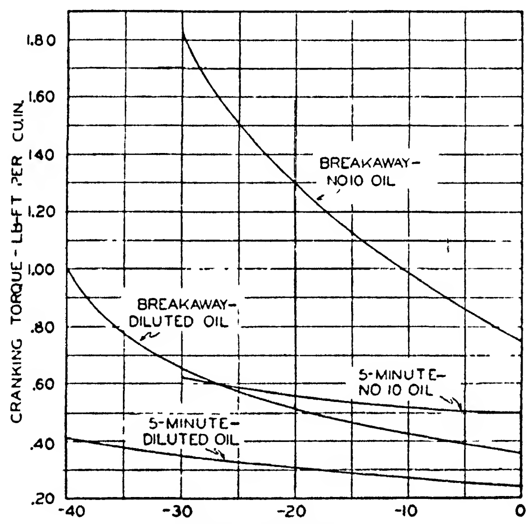


FIG. 3.—VARIATION OF BREAKAWAY AND FIVE-MINUTE TORQUES WITH TEMPERATURE.

off and replaced with the regular lubricating oil after the engine had been warmed up. This procedure evidently is too complicated to be practical. The charts, however, show that under cold weather conditions an oil of relatively low-viscosity must be used to facilitate starting. It should be pointed out here that the cranking torques plotted in Figs. 2 and 3 were obtained with the engine "decompressed."

Large Diesel engines sometimes are started by compressed air, which is admitted to the cylinders by means of special valve gear. This system is not adapted to small multi-cylinder engines, on account of its cost and complication. Separate air motors, supplied from a storage tank (air bottle) and driving through a Bendix gear, have been used for marine engines of smaller size, particularly in England.

Electric Starting—Wherever electric current is needed for other purposes, such as lighting in the case of road vehicles, motor boats, etc., electric starting offers many advantages. In this country electric starters are produced on an enormous scale for motor vehicles of all kinds. For starting the engines of passenger cars, six-volt systems are used exclusively here, but this low voltage is impractical where such large starting torques are required as for Diesel engines of 100 hp and over. For this service batteries and starters of

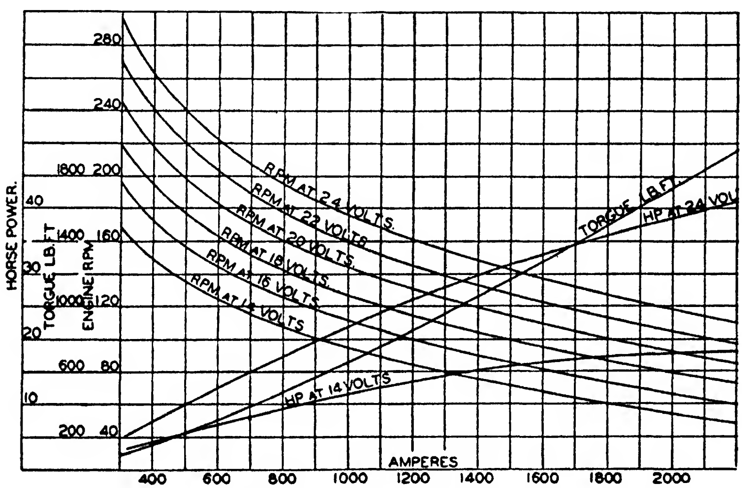


FIG. 4.—CHARACTERISTICS OF 6-IN., 24-VOLT ELECTRIC STARTER.

12 and 24 volts have come into use. Fig. 4 shows the characteristics of a 24-volt starting motor of about 6-in. diameter and 12-in. length. The engine rpm and torque in the scales at the left are based on a reduction ratio of 10 to 1 between starter armature and crankshaft. This starter will turn the engine over at 160 rpm, for instance, when supplied with current at 24 volts, and will then draw about 1200 amperes. The starting torque at the engine crankshaft is then about 1040 lb-ft, which corresponds to the torque required by an engine of about 1000 cu in. displacement at around zero F. At normal temperatures the starting torque required by this engine would be far less, and the starter would then turn the engine at relatively higher speed.

Starting-Battery Characteristics—When an engine is being started in very cold weather, the battery always must furnish a very large current, and it can maintain such a current for a short length of time only. Fig. 5 shows how the cranking times or discharge periods vary with the ambiant temperature for four different discharge rates, corresponding, respectively, to 0.5, 1.0, 1.5 and 2.0 times the 20-hr-discharge ampere-hour rating of the battery. It will be seen that at minus 40 F, for instance, the battery will furnish a given current for only about one-fifth the length of time as at plus

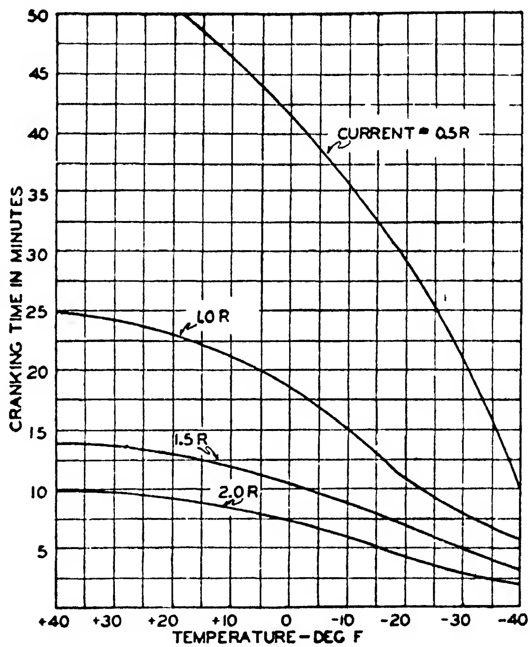


FIG. 5.—VARIATION OF BATTERY-DISCHARGE PERIOD WITH AMBIENT TEMPERATURE.

40 F. At very low temperatures it is therefore necessary to restore the capacity of the battery by heating it, and there are several methods of doing this. At moderately low temperatures the battery can be kept warm (overnight, for instance) by circulating hot water through a pipe coil at the bottom of a box surrounding the battery. But when the battery has been parked for several days at minus 40 F, this does not suffice. Under these conditions a gasoline burner, such as the Coleman, can be used to advantage. This burner,

which develops 5000 Btu per hr, is provided with a flue rising to a height of about 5 in. above it, and is "bayonetted" to the battery hanger by three tabs. Gases of combustion from the burner pass into the battery box through a 3.5-in. hole in the bottom, and escape through the 1-in. clearance holes for the cables and through an additional ¾-in. hole in the top of the casing. With this heater the temperature of the battery can be raised from minus 40 to minus 10 F in about 40 minutes. However, only batteries with hard-rubber cells can be subjected to this rapid warming-up process, which is too severe for ordinary composition cells.

Delco-Remy Heavy-Duty Starter—Fig. 6 is a longitudinal section of an electric starter developed for use on Diesel and large industrial engines by the Delco-Remy Division of General Motors Corporation. It has a frame of 5$\frac{9}{16}$-in. diameter and 9$\frac{1}{16}$-in. length, and it develops a maximum output of 12.2 hp with a current draw of 1000 amperes from a 24-volt battery. The starter is of robust design throughout, and steel banding is applied over the armature core and coils to prevent "throwing" of windings at high speed. There are six poles and six field windings in the starter. Twelve brushes can be used on the commutator, in six pairs, each pair acting as a single brush, but giving increased current-carrying capacity and steadier contact with the commutator. The armature is supported in three bronze bearings provided with wick oilers in oil wells.

These cranking motors are equipped with the Dyer drive for transmitting the power to the engine flywheel. Component parts of this drive are shown in Fig. 7. The motor shaft has helical splines cut on it and is surrounded by the starter pinion and a pinion guide, which latter is provided with internal helical splines engaging with the shaft splines, and with two driving lugs on its circumference which engage into longitudinal slots in an extension of the pinion hub. A coil spring surrounding the hub extension tends to force the pinion guide away from the pinion. The various parts mentioned are surrounded by a shift sleeve with a helical slot in it, into which latter engages a pin in the end of the shift lever, the opposite arm of which has link connection to the plunger or core of a solenoid mounted on top of the motor. However, instead of by a solenoid with remote control, the shift lever also may be controlled by a direct link connection to a starter pedal. In addition to the spring surrounding the pinion-hub extension, there is a torsion-type return spring on the shift-lever shaft. The springs aid in locking the starter pinion in the disen-

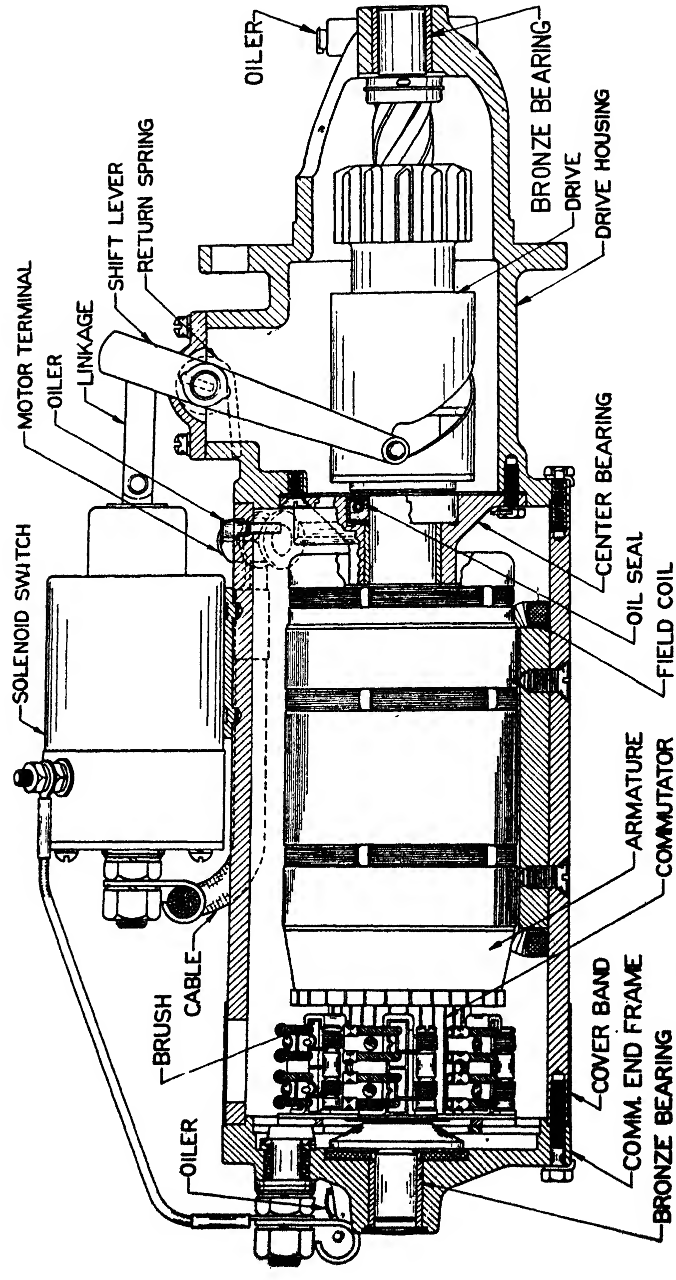


Fig. 6.—Longitudinal Section of Delco-Remy Electric Starter for Diesel Engines.

gaged position, away from the flywheel gear, and also in the engagement action.

Action of Starter Drive—With the Dyer drive the starter pinion is positively meshed with the flywheel gear before the starter circuit is closed, so there can be no clashing of gear teeth. When the starter is at rest, its pinion is held away from the flywheel ring gear by a locking action of the pinion guide and pinion spring in combination with a milled section of the helically-splined shaft. The first motion of the shift lever, moving the shift sleeve toward the flywheel, unlocks the pinion guide from the splined shaft, while farther motion of the lever causes the shifting motion of the shift sleeve to be transmitted to the pinion through the pinion guide and the coiled spring on the pinion-hub extension, and the pinion to be meshed with the flywheel gear. Axial motion of the pinion is limited by a pinion stop near the end of the shaft, and by thrust washers on the shaft near the intermediate bearing.

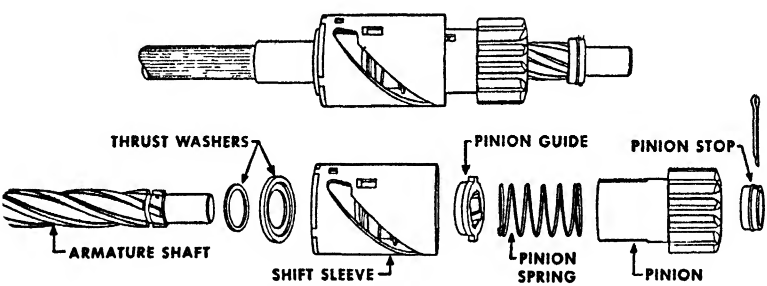


Fig. 7.—Parts of Dyer Starter Drive.

When the pinion abuts against the stop, the shift lever closes the motor switch which, as shown in Fig. 5, is incorporated with the starter solenoid. Cranking then begins, and the shift sleeve at the same time is carried back to its original position by rotation of the starter shaft and compression of the coil spring. This axial return motion of the shift sleeve is accompanied by a rotary motion due to the presence of the pin at the end of the shift lever in the helical slot in the sleeve.

As soon as the engine begins to fire, there is a reversal of the torque on the starter pinion, which forces it out of mesh with the flywheel ring gear and back to its locked position. A safety feature embodied in the design makes it impossible to close the starter circuit while the engine is running, for as soon as the pinion teeth contact the teeth of the flywheel gear, the shift sleeve is rotated and the pinion follows the armature-shaft spline back to the locked position.

Electric starting motors generally are so designed that they can safely carry the heavy currents required for cranking for a short time only, and the manufacturers warn operators against cranking for longer than a specified time, usually 30 seconds, if the engine refuses to pick up its cycle.

Starting on Gasoline—Diesel engines can be so designed that they can be started as carburetor engines, by supplying them with gasoline-air mixtures during the starting period. To this end it is necessary that either the compression chamber be enlarged for starting, so that the temperature of com-

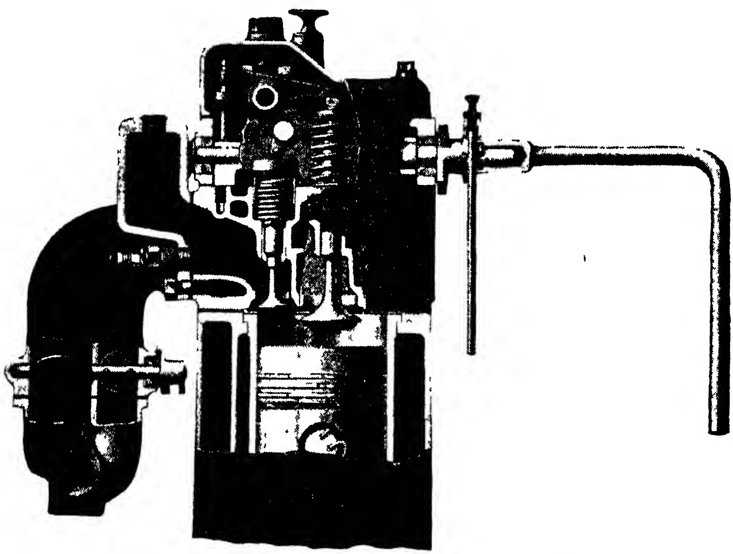


Fig. 8.—International Harvester Tractor Engine Which Starts as a Gasoline Engine.

pression will not be sufficiently high to cause ignition of the mixture even if a full charge is drawn in; or that the amount of charge drawn in be limited in some way, so that even with the small compression space of the Diesel the heat of compression will not be sufficiently high to cause self-ignition. The former seems to be the more practical method. It is employed in the International Harvester engine, the cylinder and head of which are shown partly in section in Fig. 8. This is a four-cylinder engine with 4¾ by 6½-in. cylinders and operates on the precombustion-chamber principle. The air inlet and exhaust valves are located in the cylinder head, slightly to one side of the longitudinal center plane of the engine, and to the left of these valves in the drawing there is an additional valve which places the engine cylinder in communication with an auxiliary combustion chamber used during the starting period only. The poppet valve is opened by means of a linkage connecting to a lever on the dash of the tractor, and the same operation closes butterfly valves in the main air passage in the manifold, opens the gasoline line to the carburetor, and turns on the ignition. A magneto furnishes current for ignition. The spark plug is screwed into the wall of the auxiliary compression chamber, so that it is isolated and not subjected to deterioration when the engine is operating normally.

Starting is effected by means of a hand crank, and since the compression is reduced, the engine is no harder to crank than a gasoline engine of the same size. After the engine has run for about one minute on gasoline, the change-over to Diesel operation is made by a reverse movement of the lever on the dash, which, through the linkage referred to above, closes the valve controlling the passage to the auxiliary combustion chamber, opens the butterfly valves in the main air inlet passage in the manifold, shuts off the carburetor so that no more gasoline is drawn in, grounds the magneto, and starts fuel injection.

Starting by Separate Gasoline Engine—Instead of an electric motor, a small gasoline engine may be used to start the Diesel. This plan has particular merit in the case of farm and industrial tractors, where no lighting is required. It is used on the chain-track tractors of the Caterpillar Tractor Co. Among the advantages of the system are that the engine can be cranked for a long time without exhausting the source of power; it can be started with the compression relieved, which reduces the stresses on the starter engine and gear, and in extremely cold weather the exhaust from the gasoline engine can be made use of to heat the intake manifold of the Diesel before the starter gear is engaged, which facilitates ignition in the cylinders of the Diesel.

Fig. 9 illustrates the drive from the two-cylinder gasoline starter engine on the Caterpillar tractor engine. The starter engine is mounted at the side of the Diesel and drives to the flywheel of the latter. It is started by means of a hand crank from the front of the tractor in the usual way. In very cold weather a heat-exchange valve is then opened, which causes the intake manifold of the Diesel to be heated by the exhaust from the starter engine. When this has been accomplished, the heat-exchange valve is closed again, the decompression device is set, and the starting gear is engaged.

Referring to Fig. 9, the pinion at the right, which is shown in the engaged position, engages the ring gear on the flywheel of the Diesel engine directly, and is splined to the shaft which extends from the clutch through the transmission housing. The centrifugal weights are pivoted to a case which is held to this pinion by small cap screws. In starting the Diesel, the pinion at the right is first thrown into engagement with the ring gear on the flywheel of the Diesel, by means of a hand lever which operates the small lever shown in dotted lines at the right of the assembly. This lever is then released and the clutch is thrown into engagement by another hand lever.

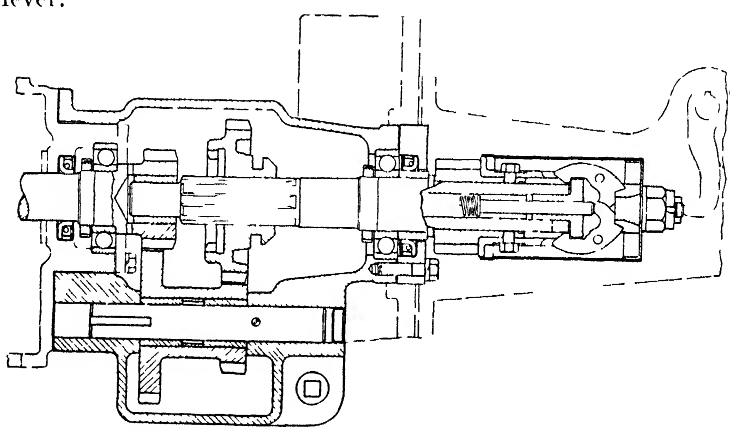


FIG 9—STARTER GEAR OF CATERPILLAR ENGINE.

As soon as the engine is turning over at the proper speed, the compression release is thrown off and the fuel-pump control lever turned on, whereupon the engine begins to operate on its own cycle. When it has attained a certain speed, the pivoted weights fly out of contact with the plug in the end of the shaft, thereby allowing the spring and plunger inside the shaft to push the entire gear and bracket carrying the pivoted weights out of engagement with the flywheel ring gear.

The cooling system of the starter engine is connected in parallel with that of the Diesel, which simplifies the pipe connections.

Starting Aids—There are numerous expedients that may be resorted to in order to facilitate starting of engines. Small engines may be spun by means of a hand crank while decompressed; in that way enough energy is stored up in a heavy flywheel so that when the decompressing device is released, sufficient heat will be generated in the cylinder to ignite the fuel when it is injected. Small stationary and marine engines can be started in this way.

One method of starting Diesel engines in cold weather consists in spraying a volatile fuel into the inlet pipe and igniting it either by means of a spark plug connected to an induction coil of the magnetic vibrator type, or by a blow torch. If the engine is being cranked while the starting fuel is burned in the inlet pipe, the interior of the cylinders quickly reaches a temperature at which compression-ignition of the regular fuel becomes possible. Even gas oil can be used as starting fuel, provided it is sufficiently finely atomized, and that makes it unnecessary to carry or keep on hand a special, volatile fuel.

Ether as Starting Fuel—Instead of burning a combustible mixture in the intake pipe, the engine may be caused to draw in a small quantity of fuel of low ignition temperature with the air. This fuel will ignite at a lower temperature than the regular Diesel fuel, and the heat of its combustion will ignite the Diesel fuel upon its injection. Of a number of different chemicals with low ignition temperatures tried, best results were obtained with diethyl ether. It was found that at the start the ether-air mixture should be rather rich, to make sure that ignition will occur and that sufficient heat will be generated by the heat of combustion of the ether to ignite the Diesel fuel. Once ignition has set in, the ether charge should be gradually reduced. In Germany extensive use has been made of a so-called "starting fuel" consisting of substantially equal parts of ether, kerosene, and gas oil. That helps to solve the problem of loss of ether, which is exceedingly volatile and hard to confine.

Capsule Starter—A device for feeding diethyl ether into the inlet pipe for starting has been developed by C.A.V. Ltd., in England, and is marketed under the trade name "Zero-start" (Fig. 10). A threaded, drilled plug at the top of the device is screwed into the manifold on the downstream side of a butterfly valve in the latter. The ether is contained in a small gelatine capsule *A* which is inserted into bowl *B*. The latter screws with a coarse thread into the body *C* of the device, and is held in position therein (against unscrewing and loss) by the wire bail *D*. As the rim of bowl *B* reaches the flexible collar *E*, the bowl is closed and ether is forced up through the axial hole in prong *F*—which has pierced the capsule—into the annular chamber *G*. As the rim passes the flexible collar, the bowl is again opened to atmospheric pressure, and finally the lower joint is sealed.

During the first few revolutions of the engine a rich ether mixture is drawn into the manifold from chamber *G*, designed to produce an immediate start. The supply of ether in chamber *G* is quickly exhausted, however. Thereafter the remaining ether is drawn through the hole in prong *F* into chamber *G*, where it mixes with air drawn in through felt filter *H* at the bottom of the device, and through passages leading from it to chamber *G*. In this way a weaker ether mixture is formed, which assists ignition until the engine has reached a

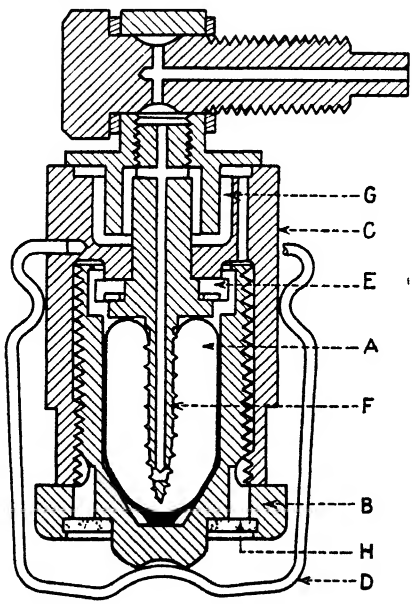


FIG. 10.—C.A.V. CAPSULE STARTER IN SECTION.

temperature sufficiently high to ignite the regular fuel positively.

Excess Fuel Device—British manufacturers of injection pumps generally provide these pumps with a device which makes it possible to inject at starting a quantity of fuel larger than that which would be injected with the control rack in the normal full-load position. An automatic trip incorporated in the device ensures that no more than the regular full-load charge can be injected in normal operation. Fig. 11 is a perspective view of the excess-fuel device (with cover removed) of the C.A.V. Type N injection pump. Rigidly secured to the fuel rack *K* at the end opposite the governor is a steel block *H*, which is threaded to receive the stop screw *B*. Mounted on the vertical spindle *L* in housing *J* is a stop plate *D*, on which is pivoted the trigger *C*. When the stop plate is pressed down by means of the external knob *E*, trigger *C* enters a groove *P* on spindle *L*, as shown in the view in the upper right-hand corner. This holds the plate in the "down"

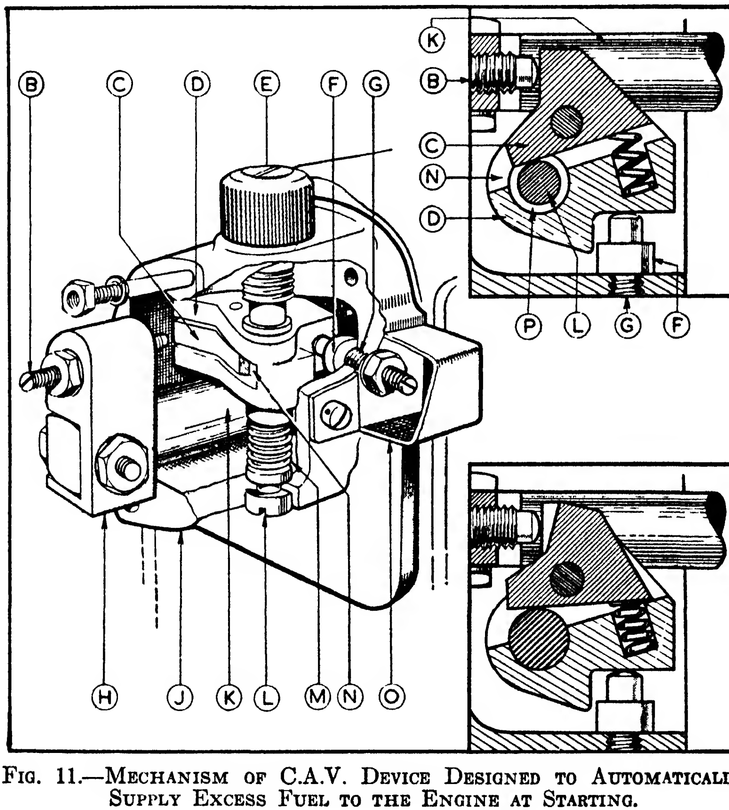


FIG. 11.—MECHANISM OF C.A.V. DEVICE DESIGNED TO AUTOMATICALLY SUPPLY EXCESS FUEL TO THE ENGINE AT STARTING.

position. If now the control rack is moved toward the maximum-fuel position by the governor at starting, stop screw *B* enters slot *N* in stop plate *D*, releasing trigger *C*, which is thereupon forced upward by return spring *M* till the clearance between slot *N* and the stop screw is taken up. This movement is sufficient to prevent trigger *C* from reentering the groove in the spindle.

As the speed of the engine increases, the governor retracts the control rack to reduce the fuel quantity, and stop screw *B* is retracted from slot *N* in stop plate *D*, thereby allowing the stop plate to return to its normal position. Thereafter the maximum "out" movement of the control rack is limited by stop screw *B* abutting against stop plate *D*. Rotation of the stop plate around the vertical spindle under the pressure of the control rack is resisted by adjustable screw stop *G*, which is accessible from the outside. The drawing in the lower right-hand corner of Fig. 11 shows the parts in their relative positions in normal operation.

Another excess-fuel device was shown in Fig. 25 of Chapter VII.

Electric Heater Plugs—Engines of the precombustion-chamber and turbulence-chamber types sometimes are provided with electric heater plugs. In precombustion chambers the wall area is so large in proportion to the volume of the chamber, and so close to the center thereof, that at low speed it is practically impossible to produce a high enough temperature by compression to induce ignition, unless an exceptionally high compression ratio is used. These heating plugs usually operate on 2 volts and consume between 40 and 50 watts each. They constitute an additional load on the starter battery just when the latter is subjected to a very heavy drain by the starter. However, engines fitted with heater plugs usually need not be cranked faster than 150 rpm, and the difference between the power consumption at this cranking speed and that at 200 or even 250 rpm more than compensates for the energy consumption of the plugs.

Electric heater plugs or glow plugs are made of nickel-chromium heat-resisting wire about $\frac{1}{16}$ in. thick. In the earlier plugs the coil of wire was grounded to the shell of the plug at one end, which necessitated connecting the plugs of individual cylinders in parallel. It takes from 25 to 30 amperes to keep the coil at a temperature of from 1500 F to 1600 F, and a storage cell of large size was required to supply the plugs with current. Such a cell, moreover, is quite inconvenient to recharge. In later models both ends of the heater-plug coil are insulated, and the plug is provided with two terminals. Such plugs are connected in series, and in the case of six-cylinder engines—which are the type most widely used on trucks and buses—they can be connected directly across the 12-volt starting battery. In the case of four-cylinder engines a resistance equal to that of two plugs must be connected in series with the four plugs.

If the heater plugs are to show a satisfactory life, they must not be placed directly in the path of the spray, yet they must be in a location where they will be surrounded with combustible vapors at the time of injection. Fuels with a high sulphur content are particularly injurious to the heater coils. To protect the coil as much as possible from heat and corrosive vapors, it is usually placed in a recess in the wall of the combustion chamber, with its tip substantially flush with the chamber wall.

Air Heaters—To reduce the work which must be done by the starter under low-temperature conditions, it is a good plan to supply heat from an outside source to raise the compression temperature. Heat can be applied to the air on entering the cylinder, to the water jacket, or to the entire engine, but it is generally most practical to supply it to the entering air. Electric heater units may be used for the purpose, and can be conveniently inserted between the inlet

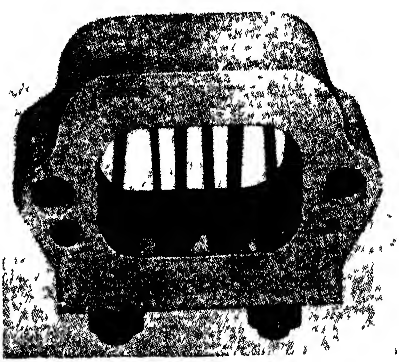


FIG 12—ELECTRIC AIR HEATER FOR INSERTION BETWEEN ENGINE AND INLET MANIFOLD

manifold and the cylinder block. Air heaters comprising ribbon-type Nichrome resistance coils in insulated blocks, as shown in Fig. 12, have been used by Mack. The ends of the Nichrome ribbon are fastened to binding posts, and copper jumpers connect the series of heater elements together. The elements are energized by means of solenoid switches controlled by a push button located convenient to the operator.

The heater elements are turned on before the starter switch is closed, in order to preheat the coils and the metal adjacent to them. They may be either disconnected when the starter switch is closed or they may be left on, in which latter case the current through them decreases about 25 per cent, due to the drop in applied voltage caused by the heavy starter current. It seems that an input of 600-700 watts per 100 cu in.

piston displacement is about the optimum. Larger inputs further reduce the time required to start, but not sufficiently to make the greater expenditure of electrical energy worth while. Sometimes it is found preferable to preheat only, opening the heater circuit as the starter switch is closed. This results in a slightly higher cranking speed and reduced cranking time. An attractive feature of these heater elements is that they are of simple construction and require little maintenance.

Flame Primer—Instead of being heated electrically, the intake air may be heated by burning fuel in it by means of a

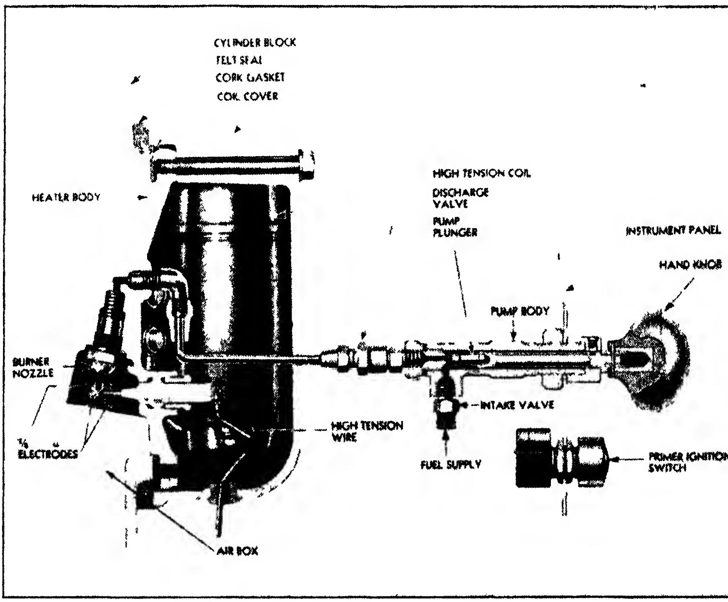


FIG. 13.—ASSEMBLY DRAWING OF GENERAL MOTORS FLAME PRIMER.

special device known as a flame primer. Only part of the intake air is consumed in the combustion of this fuel, and the remainder is available for the combustion of fuel injected into the engine cylinders. This type of preheater is particularly applicable to two-stroke engines, in which the scavenging pump furnishes a large excess of air, and in Fig. 13 it is illustrated as applied to the General Motors Type 71 engine. It is essentially a small pressure oil burner with electric ignition, similar in its operating principle to the oil burners of domestic heating plants. A diagram of the wiring and fuel connections is shown in Fig. 14. The burner proper is mounted in the engine air box, the air necessary for combustion being furnished by the engine blower, and the products of combustion together with the direct-heated air being discharged into the engine cylinders. Very little of the heat generated by the burner is lost, and the engine is said to respond immediately, as a rule.

The device consists of two assemblies, one comprising the pressure pump and ignition switch, the other the burner nozzle, filter, ignition coil and ignition electrodes. The former is mounted so as to be convenient to the engine controls, while

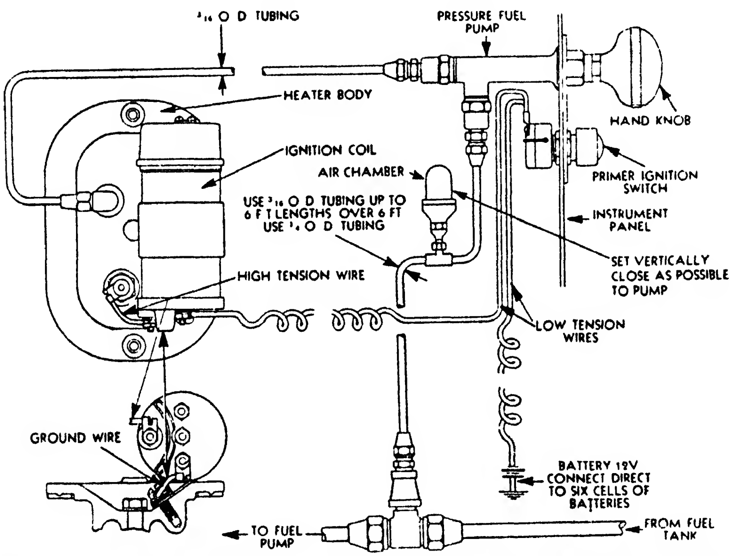


FIG. 14.—DIAGRAM OF WIRING AND FUEL CONNECTIONS OF FLAME PRIMER.

the latter takes the place of one of the hand-hole cover plates near the center of the engine air box. To prevent fuel dribble, a suitable air chamber is provided in the feed line near the pump.

When a cold engine is to be started, the primer ignition switch is turned on, the engine throttle opened wide, and the engine starter engaged. The pump plunger is then released by turning its handle from one-quarter to one-half turn in the counter-clockwise direction, and the primer pump is operated with firm, even strokes, with a pressure of up to 10 lb on the deivery stroke. After the engine has picked up its cycle, the throttle is regulated and the plunger is pushed in all the way, and then turned clockwise until the spring catch engages. Finally, the primer ignition switch is turned off.

Immersion Heaters—As already pointed out, heat from an outside source may be applied to the cooling water or coolant to facilitate starting. This can be done by means of electric immersion heaters. These, however, consume so much energy that it cannot well be supplied by a battery carried on the vehicle, but must be drawn from service mains. This particular starting aid therefore is limited in its applications.

Immersion heaters can be used in two different ways. They can be switched on when the truck or other vehicle returns from a trip, and then serve to retard cooling of the power unit and to cause it to reach an equilibrium temperature well above the ambient temperature and sufficiently high to permit of an easy start when the vehicle is to be taken out again. Alternately, the engine may be allowed to cool to the ambient temperature and the immersion heater switched on in time to allow it to approach the equilibrium temperature by the time the vehicle is to be taken out.

In some experiments with such an immersion heater, with an ambient temperature of 0 F, an engine with a coolant capacity of 6 gal cooled from 170 F to the ambient temperature in 12 hr when no heat was supplied. With an input of 500 watts to the immersion heaters it nearly reached the equilibrium temperature of 34 F in 8 hr, and with an input of 1100 watts it nearly reached the equilibrium temperature of 75 F in 6 hr. The temperature drop, of course, was quite rapid at first, and gradually slowed down. When the engine was allowed to cool down to the ambient temperature and the heaters were then switched on, it took practically the same times with the different inputs to reach equilibrium temperatures, and these latter were the same as when the engine was allowed to cool down with the heaters on. Retarding cooling of the engine is preferable to warming it from the ambient temperature, in that with the former process the engine is ready to be started throughout the period the equilibrium temperature is being approached, while with the latter it is not. With an ambient temperature of 0 F, a 500-cu in. engine equipped with an immersion heater with 1100 watts input had its coolant maintained at 74 F equilibrium temperature and was started in 8.5 seconds.

Heating Coolant with Liquid-Fuel Stove—Results similar to those obtained with electric immersion heaters may be obtained with what is essentially a miniature hot-water boiler, which is connected by piping to the engine jacket at top and bottom, and heated by a burner combined with it. The principal part of the latter is an adjustable metering device which feeds liquid fuel into a vaporizing chamber, where it is ignited by a lighted match or taper. The flame in the fire box heats the coolant in the boiler and causes it to circulate through the engine jackets by thermo-siphon action. One particular unit of this type is so designed that its rate of heat generation can be varied between the limits of 1200 and 6000 Btu per hr. Of course, when heat is generated by such a burner the losses are greater than with an immersion heater. With this type of heater the time required to reach equilibrium temperature also is relatively long.

The German State Railways in their rail cars and Diesel-engined high-speed trains, in cold weather heat the engines by steam coils for about 30 minutes preliminary to starting. The coils pass through and around the crankcase. This not only facilitates starting, but protects the engine bearings against injury due to excessive loads during the starting period. Bearings are particularly liable to injury during this period because (1) the combustion pressures are likely to be highest at low speeds, when the amount of air drawn in is a maximum; (2) the bearing load due to the initial combustion pressure is not neutralized to any extent by the oppositely directed inertia force, which latter decreases as the square of the speed and is therefore negligible at cranking speeds, and (3) when starting from cold there is practically no oil in the bearings.

Miscellaneous Starting Aids—In the past, engines have been built in which the compression ratio could be increased for starting by shutting off a part of the compression space. However, this is only a partial solution of the problem, for even if the compression ratio were increased from the conventional 15 to, say, 19 or 20, in really cold weather the engine would still be hard to start, unless additional starting aids were resorted to. The original Lanova engine had a two-compartment auxiliary chamber of which one compartment could be closed off for starting by a valve operated by means of a handwheel on the outside of the engine. Similarly, a Lister engine of the turbulence type, manufactured in England, had, in addition to the turbulence chamber, a small chamber communicating with it, which could be closed off from it for starting. It seems that all manufacturers who used this starting aid at one time have now abandoned it.

What is known as the Joule-Thomson effect is made use of in the Ganz-Jendrassik engine built in Hungary, to facilitate cold-starting. This engine is of the precombustion-chamber type. While starting, the inlet valves are opened only toward the end of the suction stroke, and then only slightly. It is claimed that the resulting strong rush of air through the narrow openings causes the air to be heated to such a degree that the engine can be started from cold without the use of any other starting aid, although its compression ratio is only 12.4. The system, of course, calls for the use of double inlet-valve cams on a shaft which can be shifted axially. Also, the energy that heats the air while entering the cylinder must be supplied via the starter, and the engine undoubtedly requires a somewhat greater starting torque.

CHAPTER XVII

Lubrication

All moving parts of Diesel engines—the same as those of other high-speed machinery—must be well lubricated to ensure their smooth and efficient operation, and to minimize the wear on them. The lubricating systems of modern engines are entirely automatic, and about all the operator has to do is to see to it that the engine is filled with a suitable grade of lubricating oil, that the strainers, filters and oil coolers are properly serviced, and that the oil is changed at regular intervals.

Oil Feed to Bearing Surfaces—All modern Diesel engines have a pressure lubrication system similar to that of automobile engines. An oil sump is formed at the bottom of the crankcase, and oil is kept in this sump at a level intermediate between maximum and minimum marks. Most engines are provided with an oil gauge, usually of the dip-stick type, by means of which the oil level can be readily ascertained at any time. A gear-type oil pump, driven from the crankshaft or camshaft, is located in the sump. Its inlet or suction pipe is provided with a screen which prevents dirt or grit from getting into it.

In aircraft engines, and in certain other types which may be operated at times at a considerable inclination to their normal attitude, the dry-sump system of lubrication is used. In this system there are two pumps which are generally combined in a single unit. One of these is a scavenging pump that draws oil from the bottom of the sump and delivers it to an outside tank or reservoir from which the pressure pump draws its supply. In certain other engines in which the sump is underneath one end of the crankcase, oil collecting in the opposite end will normally drain toward the sump, but if the engine is steeply inclined, this drainage fails, and a scavenging pump then returns oil which collects in the crankcase at the end away from the sump, to the latter.

The pressure pump delivers oil to a main distributing artery, which may be either a tube extending the length of the crankcase below the main bearings, or a drill hole in the wall of the crankcase parallel with the crankshaft. In most of the later engines an "oil gallery" is drilled in the crankcase or engine block, together with radial holes therefrom to the crankshaft and camshaft bearings. This construction is preferred to the tubular distributor mainly because it makes it unnecessary to disturb the lubricating system when crankshaft bearings have to be refitted and the crankshaft re-

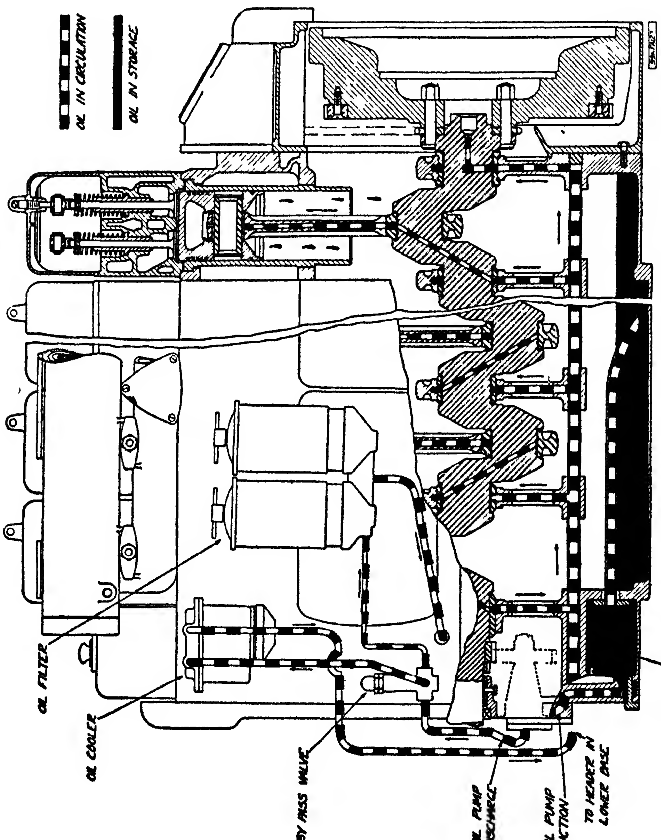


Fig. 1.—Lubrication Diagram of Fairbanks-Morse Marine Diesel Engine.

moved. A pressure-relief valve is incorporated in the pump or inserted in the delivery line. That line, as a rule, carries part of the oil to a filter and the remainder directly to the main bearings.

Crankshafts have oil holes drilled in them, which extend diagonally through the crank arm from the surface of a main bearing to the surface of an adjacent crankpin bearing. The inlet to this oil hole is so located that it registers with the oil inlet to the main bearing during part of a crankshaft revolution. Connecting rods are "rifle-drilled" and have oil holes extending through them lengthwise from the crankpin bearing to the piston-pin bearing. The inlet to this oil passage is so located that it is in registry with the outlet of the oil hole drilled in the crankshaft during part of a revolution, and oil under pump pressure therefore is supplied directly to all main, connecting-rod, and piston-pin bearings. The relative length of time an oil hole in one part remains in registry with that in another part rotating in or around it, can be varied by cutting a circumferential groove in the bearing of one of the parts, through the end of the oil hole therein, and changing the length of that groove. A lubrication diagram for a marine Diesel engine is shown in Fig. 1. In this engine the crankshaft bearings are carried by the lower half of the crankcase.

Cylinder Wall Lubrication—When a gear pump delivers oil through outlets of constant size (the bearing clearances), the pressure in the system increases with the speed, and to prevent excessive pressures from occurring in the system, the pump is provided with a relief valve—a spring-loaded ball or poppet valve which opens when the pressure in the system reaches a certain value, and then by-passes sufficient oil from the delivery to the suction side of the pump (or to the sump) to keep the pressure at the bearing inlets substantially constant. Oil working through the main and crankpin bearings is thrown off by the rapidly-revolving cranks in all directions. A certain proportion of this oil gets into the cylinders, and this oil spray is depended upon for cylinder and piston lubrication. More oil usually gets onto the cylinder walls in this way than is necessary for their proper lubrication, and the pistons therefore are provided with oil-control rings, whose function it is to scrape off excess oil from the cylinder walls and return it to the crankcase. In order that they may properly perform their function, these scraper rings must have a sharp lower edge and must exert a high unit pressure against the cylinder wall. Oil rings usually have a circumferential central groove in their outer surface, with slots at its bottom extending all the way through the ring. Any oil collecting in the groove in the ring passes through the slots into the ring groove, from which it returns to the crankcase through oil-return holes drilled through the piston skirt.

Valve-Gear Lubrication—Similar provisions are made for the lubrication of the valve gear. Oil is fed directly from the main distributing passage or from the main bearings to the camshaft bearings, and the camshaft sometimes is drilled out and forms an oil tube, with an inlet at one of its bearings and outlets at all of the other. The valve rocker shaft on top of the engine also is a tubular shaft and serves its part in the lubrication system. An oil tube or a drilled passage extends from the main distributing passage in the crankcase to the valve chamber, where it connects to one of the supports or pedestals for the valve rocker shaft, or it may connect to this shaft directly at one end. The rocker shaft does not turn, but the rocker levers rock upon it, and a radial drill hole is provided in the wall of the shaft at each rocker bearing. Sometimes oil holes are drilled in the rocker levers themselves, their inlets registering with the radial hole in the wall of the tubular shaft, and their outlet being located at the surface where the rocker contacts with the valve.

Oil from the pressure system is supplied also to the housing for the camshaft and accessories drive, and to the bearings in that housing.

Lubrication System for Heavy-Duty Engines—Reliable lubrication is particularly important in the case of large railroad and marine engines, which operate at a high load factor for long periods. Fig. 2 is a lubricating-system diagram of the Daimler-Benz 800-hp railroad engine. There are two separate oil circuits, each with its own gear-type pump in the engine sump. Pump *A* delivers oil through a "foot valve" *B* and a filter *C* to the main oil gallery, and through it to the bearings. After passing through the bearings the oil drains back to the sump. The filter is of the two-stage type, comprising a primary filter with three ribbon-type filtering elements or inserts, and a secondary fine filter with two inserts. Connected to the line from the filter to the oil gallery are the pressure-relief valve *D*, the pressure gauge *G*, and the hydraulic engine governor *F*.

The second pump, *I*, delivers oil through another "foot valve" *B* to the heat exchanger (or oil cooler) *K*, from which it returns to the sump. Heat exchanger *K* has a cooling circuit separate from that of the engine, including a radiator and fan *M*. There is only a single water pump, *L*, but it has two impellers. One of these forces water through the engine cooling circuit, the other through the heat-exchanger circuit. By keeping the two circuits separate it is possible to maintain the oil in the sump at 167 F, while the water enters the engine jackets at 185 F.

To protect the engine from injury during cold starts, an auxiliary electric pump *N* is provided. This pump starts operating the instant the engine starting device is operated, while the engine cannot start until a predetermined pressure has been built up in the lubricating system. The auxiliary pump forces oil into the engine lubricating system through check valve *O*.

Cause of Bearing Friction and Wear—In order to operate with a minimum amount of friction and wear, bearing surfaces must be kept apart by a film of oil. A bearing surface, even though it may appear perfectly smooth to the naked eye, consists of microscopic "hills" and "valleys," and when there is no oil film between the surfaces, the load on the bearing forces the "hills" of one surface into the "valleys" of

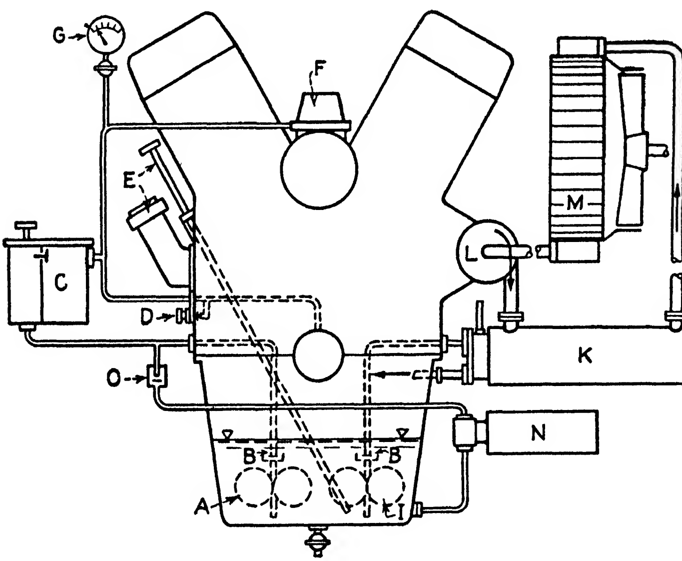


FIG. 2.—DIAGRAM OF DAIMLER-BENZ LUBRICATION SYSTEM FOR RAILROAD ENGINES.

the mating surface. Then, when the journal is rotated in the bearing, the "hills" on the former must be lifted out of the "valleys" on the latter and over the "hills" on same, which entails high frictional resistance. There is, of course, also a tendency for the hills on one surface to bend over or break off the "hills" on the other surface, and if such deformation of the bearing surfaces actually occurs, it means wear. To minimize both friction and wear, the opposing bearing surfaces must be kept sufficiently far apart so that the high spots on one will not come into metallic contact with those on the other. The journal then floats on a cushion of oil.

As long as there is a complete oil film there is practically no wear at all, but when the engine is stopped, the oil is squeezed out or drains off from the bearing surfaces, and most wear undoubtedly occurs during the first few minutes of operation after a period of rest, before the oil film has had a chance to form. Maintenance of the oil film probably is more difficult in the case of Diesel engines than in that of any other type of machinery, because of the high pressures and the high temperatures in the engine. Peak pressures in the cylinders of Diesel engines may exceed 1000 psi, and when starting, while the speed is still low, the unit bearing pressures are correspondingly high. Later on, when the engine is up to speed, the inertia of reciprocating parts acts in opposition to the gas pressure on the piston, and this reduces the bearing loads during periods of peak pressure in the cylinders. But the inertia forces themselves create bearing loads, which in a four-stroke engine reach peak values four times as often as the gas pressure, and at high speeds they are the chief cause of bearing loads.

Bearing Metals—Bearings of automotive-type gasoline engines have long been lined with tin-base babbitt metal, which has the desirable property that it is comparatively soft and can be used with a shaft or journal not specially hardened, without causing excessive wear on the latter. An objectionable feature of tin-base babbitt is that its hardness and strength decrease rapidly as the maximum temperature reached by crankcase oil is approached, and the load-carrying capacity of the bearing is then greatly reduced. In the case of many heavy-duty Diesel engines it has been found necessary to use bearing metals which are not so sensitive to temperature, such as lead-base babbitt, cadmium-silver, and copper lead. Such bearings will carry materially greater loads than tin-base babbitt, but the last two are considerably harder, and to prevent excessive wear on shaft journals, the latter must be hardened. Crankshafts of Diesel engines now are generally hardened on their bearing surfaces by the induction-hardening process; that is, their bearing surfaces are rapidly heated by subjecting them to the inductive effect of high-frequency electric currents in coils surrounding them, and are immediately quenched by a water spray. Aside from their greater wearing effect on shafts and journals, most of these substitute bearing metals are attacked by certain engine lubricants, such as the running-in oils, and also by straight mineral oils that may have undergone a chemical change due to exposure to excessive temperatures. No lubricating oils should be used in Diesel engines which have a corrosive effect on the metals with which their bearings are lined.

Piston Materials—Since the adoption of superior bearing metals, bearing lubrication has been less troublesome than cylinder lubrication. In severe service the crown of the piston reaches a very high temperature, and oil coming in contact with either the under side of the piston crown or with the lands of the ring belt is apt to reach a temperature that will cause it to undergo a chemical change. In service, the hottest point of the crown is usually the center. Most of the heat absorbed from the burning gases by the piston is transmitted by the piston rings to the cylinder wall, and thence into the cylinder jacket. Since the heat absorbed by the piston must flow radially outward to reach the rings, there must be a temperature gradient from the center of the crown to its edge, and this temperature gradient is much greater in cast iron than in aluminum pistons, primarily because cast iron has a much lower heat conductivity, but also because, on account of its higher specific gravity, it cannot be used in such thick sections. The crowns of cast-iron pistons often become so hot in service that the oil accumulating on their under surface will carbonize and form a thick layer of coke thereon. In pistons of large diameter this is sometimes guarded against by providing them with a diaphragm immediately above the piston bosses, which keeps crankcase oil away from the crown. The tendency in high-speed Diesel engines is toward the use of aluminum pistons, and the crown is usually made quite heavy.

Piston Temperatures—Lubrication trouble with Diesel engines is largely due to the occurrence of excessive piston temperatures. Exposure of the oil to excessive temperature results in deposits on combustion-chamber and crankcase walls, accumulation of varnish-like material on pistons and in ring grooves, formation of sludge in the oil sump, coating and etching of bearing surfaces, and deterioration of the oil as a lubricant. A considerable amount of research work therefore has been devoted to piston temperatures and the factors which control them. While the center of the crown is the point of maximum temperature—except in cases where some other part of the crown is struck by the fuel spray—the temperature that matters most is that adjacent to the top piston ring. This ring gets hotter than any of the others, and consequently it is most likely to stick as the result of accumulation of gum and carbon in the ring groove. When the top ring sticks, serious trouble usually is not far off, because if nothing is done about it, the other rings usually will follow suit in short order. The piston seal is then seriously impaired, blow-by will increase rapidly, and scuffing and scoring of the rings and cylinder wall will soon begin. Sticking of piston rings is indicated by a loss in torque or power for a given control-rod setting, and by an increase in the discharge of vapor or smoke from the breather pipe.

The temperature of the piston adjacent to the top ring increases almost in direct proportion with the load, and under very heavy loads, when the injection quantity is larger than the engine can burn properly, and the combustion period is therefore prolonged, the temperature rises even faster than the load. For a given fuel setting the temperature usually rises with the speed, though a case has been reported where the temperature was at its maximum at an intermediate speed and dropped off at both higher and lower speeds. A change in the jacket temperature within the range possible with water cooling results in a substantially equal change in piston temperatures. When some of the rings are removed from the piston, the temperature at the top ring increases, which may be accounted for by the fact that the path for heat flow to the cylinder wall is then restricted, but it may be due also to increased blow-by.

Oil Consumption—While the engine must be properly lubricated, it should not consume more oil than is necessary, because any excess consumption not only is a waste, but is likely to be attended by undesirable operating conditions, such as a smoky exhaust and the formation of carbon deposits in the combustion chamber. Excessive oil consumption often is due to cylinder distortion. Such distortion may be caused by unequal heat expansion of the cylinder head, which is usually a rather irregular casting and is rigidly secured to the cylinder. Distortion may be caused also by irregular flow of the cooling water through the jacket. Cylinder distortion can be measured only with the engine at rest and substantially at atmospheric temperature, and what it may be under operating conditions can only be conjectured. It has been claimed that a very smooth cylinder bore (less than 6 microinches irregularities) has the effect of increasing the oil consumption.

Oil consumption generally increases rapidly with engine speed. A. T. Stahl of Mack Manufacturing Co. at a National Oil and Gas Power Conference presented figures indicating that an increase in engine speed from 2000 to 2400 rpm roughly quadruples the hourly oil consumption. If the consumption is excessive, the usual remedy lies in a change in piston-ring equipment. How strongly the ring equipment can influence the rate of consumption was brought out by an experience related by Mr. Stahl at the same meeting. A production engine of about 500 cu in. displacement showed a wide variation in the consumption rate, and it was thought that this might be due to slight out-of-squareness of the compression rings. Therefore, following the usual "break-in" run, all of the compression rings were replaced so that any signs of hard bearing were at the lower edge. An oil-consumption test was then run, and following this all of the compression rings were reversed in their grooves, so that the hard bearing was at the upper edge. It was found that with the hard-bearing edge down the consumption was 0.130 lb per hour, whereas with that edge up it was 0.694 lb per hour. The out-of-squareness of the rings was very small as compared with the ¾ to 1 deg used in conventional tapered compression rings.

Lubrication of Two-Stroke Engines—The two-stroke engine presents some special problems in connection with cylinder lubrication, because of the presence of ports at the bottom of the combustion chamber. This part of the combustion chamber usually is surrounded by an air chamber in which the air is maintained at a super-atmospheric pressure by the scavenging blower. Care must be taken that not too much oil gets to the air ports, for if it does, carbon deposits are formed around these ports and interfere with the port action (the charging operation). To prevent an excess of oil from reaching the ports, an oil control ring is placed on the piston close to its open end, and the piston is made sufficiently long so that this ring is still below the port at the end of the up-stroke. Owing to the over-pressure in the air chamber, there is a constant blow-by from it into the crankcase. This not only reduces the amount of oil which the oil ring at the bottom of the piston must scrape off the cylinder walls, but it also helps to keep the passages in the control ring clear. In the General Motors two-stroke engine there are two oil-control rings at the open end of the piston, and these are of three-piece construction, comprising two ring sections and a steel expander spring. This is said to have the advantage that it affords a large oil-return passage, in addition to which the scraping of one section against the other is thought to give a self-cleaning effect. The effectiveness of these three-piece rings can be changed by placing both ring sections in the groove with the scraper hook down—in which case the ring is most effective—or with one up and the other down.

Results of Faulty Lubrication—Faults in the lubrication system or in the lubricant may result in (1) burning out or melting out of bearings, (2) corrosion of bearings, (3) sticking of piston rings, (4) scuffing and scoring of rings, pistons, and cylinder walls, (5) excessive cylinder wear.

A new engine or one which has been reground or has had new liners fitted must be "run in" or "broken in" to wear down the high spots on the contact surfaces of the cylinders, pistons and rings. During this "running-in" process there always is considerable risk of injury to the wearing surfaces by scuffing or scoring. Scoring results when high spots in the contacting surfaces are forced together with such pressure that the lubricant on them is squeezed out or scraped off. There is then "dry friction" at this point, and so much heat may be generated that the metal reaches the welding temperature; a small particle in one surface welds to the other and is carried along by it, and the scraping of this particle over the surface from which it has been torn does the scoring. Certain special compounded lubricants have been developed which have the property of lessening the scoring tendency during the running-in process. These "running-in" oils usually contain fatty acids, but such acids attack lead and cadmium, and the fatty-acid type of oil cannot be used in engines in regular operation.

Diesel Lubricants—The bulk of the oil used for lubricating combustion engines is straight mineral oil of either paraffinic or naphthenic base. Where the service is not unduly severe and where the engines, in consequence, reach only moderate temperatures, these oils serve quite satisfactorily. Oils are marketed in different viscosity grades, and the viscosity grade should be selected in accordance with the manufacturer's recommendation. A somewhat heavier grade may be used in summer than in winter, and in a worn engine than in a new one. Cold starting is easier with a low-viscosity oil, because with such an oil the starter will crank the engine at a higher speed, and a higher temperature will be attained in the combustion chamber during compression. Another factor besides viscosity to which considerable importance is attached by some lubricating engineers is the viscosity index. This refers to the change in viscosity with temperature. Two oils may have the same viscosity at one of the standard temperatures for making viscosity tests (130 F and 210 F) and yet their viscosities may be quite different at the maximum temperature of the oil film in the bearing. Oils whose viscosity changes least for a given change in temperature are said to have a high viscosity index and are generally considered superior engine lubricants.

Compounded Oils—All oils deteriorate if subjected to sufficient heat and agitation with air. The only difference between oils is that some of them can stand more heat than others. Like many other chemical processes, the oxidation and decomposition of oil increase in rate very markedly as the temperature rises. Oils do differ, however, in the type of undesirable products formed. Some of them oxidize to form sludge varying from a light brown, varnish-like material to a jet-black, asphaltic type. Varnishes or sludges may accumulate and become deposited on piston skirts, valve stems, rocker arms, in oil lines and crankcases, on the under side of pistons, and in the ring grooves. If they accumulate in the ring grooves, the heat tends to bake them into hard carbon deposits which hold the rings, preventing their proper functioning, thus causing engine failure.

Other oils tend to oxidize to form petroleum acids without concurrent sludge formation. These acids do not have a harmful effect on iron, aluminum, copper or babbitt when used in engines. On the other hand, they do attack lead and cadmium as used in the newer bearing metals.

Most oils marketed before 1935 were of the first type, in that they tended to form sludge, provided the engine conditions were severe. The oil industry eliminated most of this trouble by adopting solvent-refining processes. The solvent-refining process removes most of the sludge-forming constituents of the oil, but also some of the natural inhibitors which may reduce acid formation. These solvent-refined oils proved satisfactory until the newer types of bearings were introduced in heavy-duty or severe-service engines. For these engines it was necessary to add inhibitors to the oil to prevent bearing corrosion.

Detergent Oils—As operating conditions became more severe, ring sticking and piston coating became more evident, and it was found that it was necessary to use detergent oils for satisfactory operation. The detergent oils contain metal compounds or soaps which do not permit the solid oxidation products to coalesce and settle. They keep the sludge in a finely-divided form in suspension, so that it is washed out when the oil is drained. The early detergent-type oils corroded the newer type bearings, but the more recent oils have been properly inhibited or contain other detergents which function satisfactorily under all conditions.

So much emphasis has been placed on oils lately that one is apt to blame all trouble on them. It must be remembered that engines vary just as much as oils. Some engines operate with very high crankcase oil temperatures, causing the oil to oxidize very rapidly. Other engines have unusually high ring-belt temperatures, which inevitably leads to ring sticking if operated at full load. Proper water jacketing and oil coolers reduce sludging and ring sticking, and lengthen engine life just as effectively as the more expensive oils. Operating conditions, such as load and speed, and frequency of oil change, also determine engine cleanliness.

Other additives have been discovered which not only have oxidation-inhibiting properties but reduce the tendency to score rings and cylinder walls. Some engines require such compounds, especially when operated under severe conditions Chlorine, sulphur and phosphorous compounds possess this anti-scoring or anti-welding property, and are used in some heavy-duty crankcase oils. These compounds are similar to the extreme-pressure additives used in gear oils, particularly for hypoid gears.

Heavy-duty compounded oils can be used even if the service is not severe, but the extra expense is not justified. It is important, however, that the operator use a properly-inhibited oil if he plans to use it in a Diesel engine with copper-lead or cadmium-silver bearings where crankcase temperatures are above 200 F.

CHAPTER XVIII

Operation and Maintenance

Directions for the operation and maintenance of engines can be given with much greater definiteness if intended for a single make or design. This applies particularly in the case of high-speed Diesel engines, which are still in a state of rapid development and therefore differ widely as regards features of design and items of equipment. Nevertheless, some hints and directions are given in this Chapter which are believed to be of fairly general application. Even where they are not directly applicable because the assumptions on which they are based do not hold in the particular case, they may be suggestive as to what can be done under the prevailing circumstances.

Starting—After an engine has been shut down by cutting off the fuel supply at the injection pump, the stop control is released and the quantity-control rack of the injection pump or its equivalent is automatically returned to the full-load position by the governor spring. The control rod should always be in this position when the engine is being started, because fuel leakage is greatest when the engine is being cranked over at low speed, and the amount of fuel injected therefore will be less than at normal speed with the same setting of the quantity control.

The majority of high-speed Diesel engines are equipped with electric starters, and the routine starting operation consists merely in pressing down on the starter button or its equivalent. The starter will then crank the engine, and the latter, as a rule, will immediately pick up its cycle, whereupon the starter button is released. However, if the engine has not been used for some time, or if the atmospheric temperature is close to or below the freezing point, the starting operation is not quite so simple.

During an extended period of non-use, practically all of the oil will drain off the cylinder walls, and as it is possible for a Diesel engine to pull practically a full load immediately after starting from cold, whereas a gasoline engine must be "nursed" through a more or less extended "warming up" period in cold weather, precautions must be taken with the former to prevent injury to the bearing surfaces by subjecting them to excessive loads before they are adequately lubricated. One manufacturer recommends that before an engine is started after an extended period of rest, the injection nozzles be removed from the cylinders, oil injected into the cylinders with a squirt can, and distributed over the cylinder walls by turning the engine over with the hand crank three or four times. An incidental advantage of this procedure is that a better seal is assured for the first few compressions, so that conditions affecting ignition are improved. The cranking also starts circulation of the oil by the regular lubricating system of the engine, and it gives assurance that the engine is free to turn and cannot be injured by having the power of the electric starter applied to it. Injury under such conditions might result from an accumulation of water in the combustion chamber, possibly due to a leak in the cylinder-head joint.

Diesel locomotive and rail-car engines used on the German State railroads are equipped with electrically driven auxiliary oil pumps which put the lubricating systems under pressure before the engines can be cranked. A switch or contactor operated by the oil pressure is connected into the starter circuit, and this circuit therefore cannot be closed and the engine turned over by the starter until the oil pressure has reached a predetermined value.

Limiting Bearing Loads—Considerable trouble was experienced—especially abroad—with early high-speed Diesel engines from cracking of the babbitt lining in the upper half of the crankpin bearing. The cracking occurred directly below the junction of the shank with the bearing hub, and was the result of too high a specific pressure on that part of the bearing. Such injury is most likely to occur when starting from cold, because the low temperature of the air in the combustion chamber increases the ignition delay, thereby increasing the maximum pressure of combustion, which is not relieved or countered to any extent by inertia forces, owing to the low speed, and the effect may be further aggravated by the absence of an adequate oil film on the bearing surfaces, which always acts as a cushion.

In most of the later engines, lead bronze is used instead of babbitt for the connecting-rod bearings, which is capable of withstanding much higher bearing loads, and, besides, greater care is now taken to so design the rod that the thrust transmitted by it is distributed more nearly uniformly over the whole bearing surface. But even where these precautions have been taken by the designer, it is well for the operator to guard against excessive combustion pressures at starting, which are always detrimental to engine life.

Starting in Cold Weather—Considerable difficulty is often experienced in starting Diesel engines from cold when the surroundings are at a temperature below the freezing point. Ignition is dependent upon a certain minimum temperature being produced in a part of the combustion chamber that is reached by the fuel spray. This temperature may be hard to attain, because the air entering the engine is at a very low temperature, and also because the combustion-chamber walls are at an equally low temperature, so that as soon as the temperature of the air begins to rise by reason of its compression, the air loses heat to the cylinder walls rapidly. This loss of heat is likely to be aggravated by the fact that the engine is being cranked at less than the normal cranking speed, because the oil on the bearing surfaces is nearly congealed and the engine therefore is very stiff, and also because the capacity of the battery to furnish starting current is reduced at low temperatures.

Under extreme conditions it is advisable to drain all of the oil and cooling water (or anti-freeze solution) from the engine when shutting it down, and to return these fluids to the engine only after they have been heated to near the boiling point. This will heat up the whole block and render the engine limber for starting. It is also advisable to remove the battery from the engine and maintain it at room temperature overnight. Starting can be facilitated also by heating the branches of the inlet manifold with a blow torch over the greater part of its length, or the flame from the blow torch may be directed into the inside of the manifold through a threaded hole provided for the purpose and which is normally closed by a screw plug. This is done for a moment just before the operator presses down on the starter button. Each of the expedients mentioned has a certain effect in facilitating starting, and in moderately severe weather the one which is least irksome can be applied, additional ones being resorted to as conditions may require. When the cooling fluid is heated for starting, if it consists of a solution of alcohol or some other solute giving off inflammable vapors when heated, care must be taken that these vapors cannot be ignited.

The minimum temperature at which an engine can be started depends on the ignition quality of the fuel, but even with the best commercial Diesel fuels, those of highest cetane number, starting difficulties usually begin if the temperature drops below 20 F. It has been found that a blend of diethyl ether with an equal quantity of Diesel fuel greatly lowers the minimum temperature at which starts can be effected, to around 0 F, and even more effective is the feeding of diethyl ether directly into the inlet manifold while the engine is being cranked. By this method engines have been started at temperatures as low as minus 40 F. Of course, feeding fuel directly with the air into a Diesel engine involves certain risks, and care must be taken that not too much can be supplied.

In extremely cold weather it is always advisable to crank the engine over by hand (if that is possible) before attempting to start it with the electric motor, as that will prevent needless exhaustion of the battery.

Most of the established manufacturers of Diesel engines have worked out suitable cold-weather starting procedures for their engines, which are described in their instruction books. They either supply the necessary starting aids, or furnish information as to where they can be obtained or how they can be constructed, and these instructions should be followed whenever possible.

Selection and Handling of Fuels—Fuels should be selected in accordance with the recommendations of the engine manufacturer as to pour point, viscosity, sulphur, water, sediment, and Conradson carbon limits, and as to cetane number (critical compression ratio or Diesel index). Occasions may arise when the engine manufacturer's specifications cannot be met, and in emergencies it may become necessary to use kerosene or light distillate. The only disadvantages of these fuels, aside from their higher prices as compared with the usual gas oil, are that they have less lubricating value and are therefore somewhat harder on the injection pump and nozzles. This may be remedied by adding 2 to 3 per cent of a light lubricating oil to the fuel when filling the tank.

The main thing to look out for when handling the fuel is to keep all dirt and water out of the system. Some large users of Diesel engines have found it advantageous to centrifuge all fuel purchased by them. Others buy fuel in bulk and store it in large tanks for several days, so that all water and solid particles in the fuel may settle to the bottom, whence they can be drawn off through drain cocks.

When filling the main fuel tank of the truck, etc., it is advisable to make use of a funnel with a wire-screen strainer of 200 mesh soldered into it. In spite of such precautions a certain amount of water will always accumulate in the fuel tank, mainly through condensation of atmospheric moisture, and the tank is normally provided with a settling chamber with drain cock at its lowest point. It is quite important that no water should get to the injection pump, because both the plunger and the barrel of the pump are usually made of ferrous materials (cast iron and steel), and if the engine were laid up for some time and there happened to be even a small amount of water in the pump, the plungers probably would be found rusted to the barrels when the engine was next to be started again. Another reason for trying to keep water out of the system is that in cold weather it would freeze in the line and close it up. Most engines therefore have another settling chamber with drain plug in the fuel line from the tank to the transfer pump or between the transfer pump and the injection pump. This should be drained every day.

Fuel Filters—Because of the very small clearance between the injection-pump plunger and barrel (of the order of 0.0001 in.) it is necessary to remove even the finest particles of grit and abrasive material, and this is the reason for the successive filtering operations to which the fuel is subjected on its way to the pump. In fact, the life of the injection system is dependent chiefly on the thoroughness with which the fuel passing through it is deprived of abrasive impurities. Large particles of grit, if not filtered out, will cause the working parts of the injection system to stick or score, while smaller particles will increase the rate of wear on them.

Fig. 1 shows a fuel-system installation recommended by American Bosch Corporation. It shows the transfer pump or fuel-supply pump combined with the injection pump. At the outlet from the fuel tank there is a shut-off cock, and in the line from this cock to the transfer pump there is a "preliminary," low-resistance filter. In some installations this consists merely of a fine-mesh wire-gauze screen which, after removal from its housing, can be cleaned by washing in fuel oil, kerosene or carbon tetrachloride, and then subjecting it to a blast of air. So-called edge-type and renewable-element filters also are used in this location. The preliminary filter, through which the fuel is drawn by the suction of the transfer pump, serves to catch any coarse particles of dirt there may be in the fuel, and thus to protect the transfer pump. Between the latter and the inlet to the injection pump in Fig. 1 there are two filters connected in series. While the first of these, the primary-stage filter, may be of a cleanable type, it is recommended that a renewable-element-type be used for the final-stage filter. Since the fuel has been filtered at least twice before it reaches the final-stage filter, the latter should have a relatively long life. An alternate arrangement, by which the two filters in the line to the injection pump are connected in parallel, is indicated in dash-dotted lines.

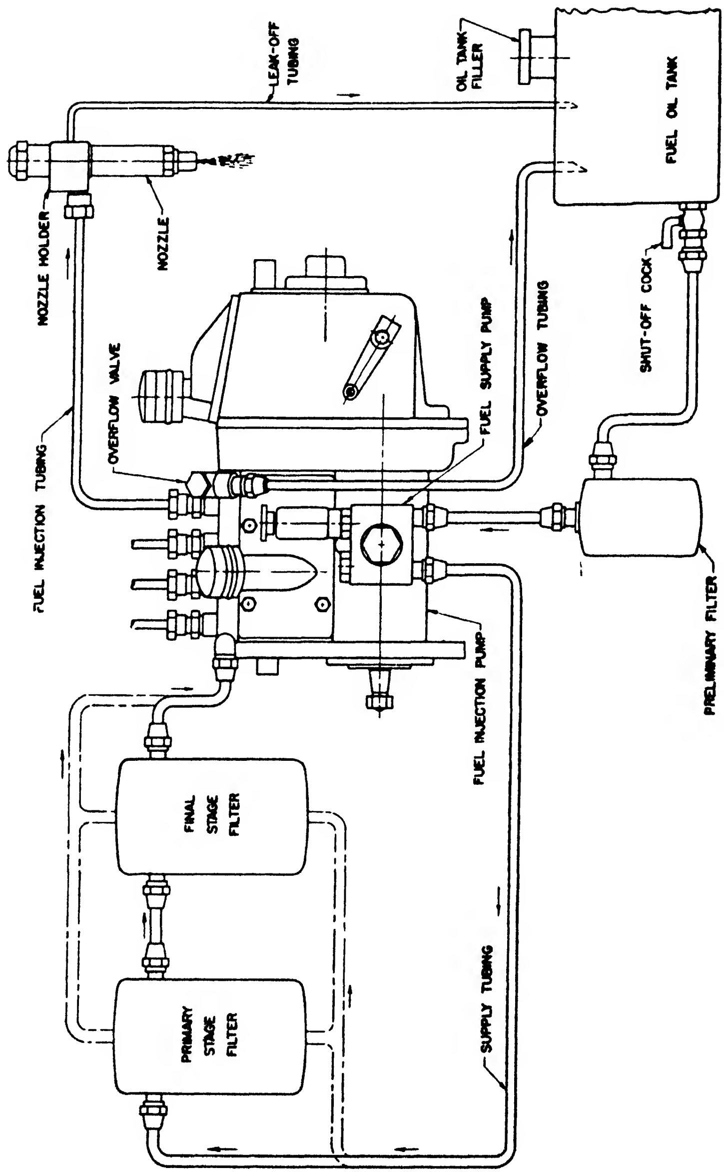
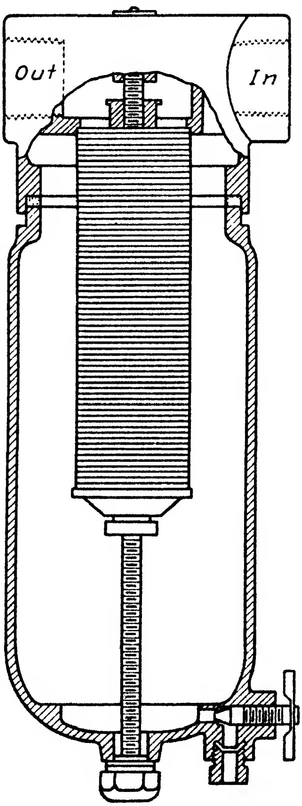


FIG. 1.—FUEL-SYSTEM INSTALLATION DIAGRAM.

Filter Types—The screen-type of filter, which is made of fine-mesh wire gauze, hardly calls for a description. The filtering element may be a disc clamped between flanges of its housing, but it is preferably made of conical shape, so it will provide a fairly large screening surface without being unduly bulky. The edge-type of filter, illustrated in Fig. 2, contains a filtering element consisting of a large number of metal discs stacked on top of one another. Risers on one side of the discs hold the latter a minute distance apart. Fuel enters the container on top at the right, is drawn or forced through the capillary spaces between discs, rises inside the stack of discs, and leaves the strainer through the outlet at the left. Dirt particles larger in diameter than the distance between discs are retained on the outside of the stack, and some may be shaken loose and drop to the bottom of the container, where any water in the fuel also will collect. The water can be drained off through a drain cock at the bottom, which should be opened at frequent intervals. Some filters of the edge type have provisions for cleaning the filtering element without removing it from the container. There may be a perforated sheet-metal cylinder inside the discs, with a piston which can be reciprocated in it by means of a handle at the top. Normally the piston is at the bottom of the cylinder, and a threaded extension of the hub of its handle is screwed into the head casting. By first giving the handle a few turns to free it from the head casting, and then pulling up on it, fuel is forced through the spaces between discs in the reverse direction, which tends to dislodge any dirt adhering to the outside of the stack. The only way in which the filtering element shown in Fig. 2 can be cleaned is by removing it from the container, loosening the discs by unscrewing the nut at the bottom of the stack, and then washing the element in gasoline. Sometimes, if the engine has been out of service for a considerable length of time, the spaces between discs may have become choked with gum formed in the gas oil. This can be removed by washing with lacquer thinner.

Filter Elements of Organic Material—Another type of fuel filter has a filtering element composed of felt, cotton, cellulose, or other fibrous material. One such filter is shown in section in Fig. 3. The container consists of a base, shell, and cover. The filtering element comprises inner and outer perforated shells and is slipped over a tube that screws into a threaded boss on the cover and at the bottom enters a hole at the center of the base, where it is engaged by a screw cap by means of which the parts are held together and through which the interior of the central tube communicates with the outlet in the base.

Some felt filtering elements can be cleaned by washing the felt discs in fuel oil until all traces of dirt have disappeared from the outside, rinsing in gasoline or other volatile cleaning fluid, and then allowing to dry in a dust-free atmosphere. This cleaning process can be expedited by providing one end



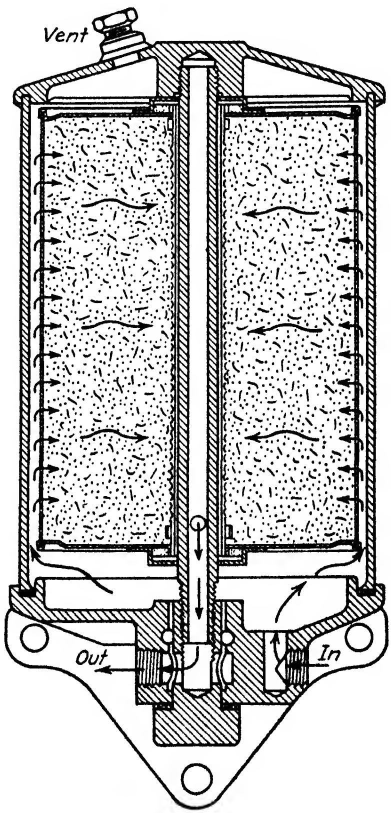


FIG. 2 (*Left*).—EDGE-TYPE OF FUEL FILTER.
FIG. 3 (*Right*).—FILTER COMPRISING A FILTERING ELEMENT OF ORGANIC MATERIAL.

of the central tube with an adapter from which connection can be made to an air hose or a hand pump. The air flow agitates the cleaning fluid, which promotes the removal of dirt, and the air pressure in the central tube prevents dirt in the cleaning fluid from entering the felt. After such felt filtering elements have been cleaned, say, a dozen times, they should be replaced by new ones. The various joints in the filter are rendered fuel-tight by gaskets or packing rings, and these should be renewed each time the filter is taken apart.

Bag Type of Filter—The third type of filter is the bag type (Fig. 4), the filtering element of which consists of a bag of filtering cloth which is folded or pleated to provide a large filtering area in a compact space. The bag is either passed over a supporting coil of steel wire and drawn into the spaces between adjacent convolutions by an additional coil of smaller diameter on the outside, or is wrapped around a spacer mat. These bag-filtering elements cannot be satisfactorily cleaned, as the cleaning operation will force dirt particles through the pores of the fabric, so that on reassembly they could enter the pump. Therefore, when the filtering element becomes clogged it is replaced. In these filters, also, the fuel passes through the filtering element from the outside to the inside. One manufacturer has combined a felt filtering element with a bag, the latter surrounding the former. The bag, which serves as the primary filtering element, has the advantage of inexpensive renewal, besides which it lengthens the life of the felt element. Combined bag- and edge-type filters also are being used.

The filtering efficiency of most filters increases with use, because the layer of dirt particles collecting on the filtering surface acts as an auxiliary filter. However, the flow resistance increases with the accumulation of dirt, and eventually a condition is reached where the fuel cannot get through the filter fast enough to meet engine requirements. Irregular operation of the engine, due to misfiring occasioned by an inadequate fuel supply, usually is a symptom of a choked filter. One method of proving the existence of excessive flow resistance in the filter consists in fitting a flexible hose or tube to the overflow manifold of the injection pump and allowing it to discharge into a container of suitable size. While the engine is being run at moderate speed, the rate at which the fuel collects in the container is measured. If this seems to allow an insufficient margin for full-load operation, the filtering elements are removed from the different filters in succession, and the increase in the rate of overflow after each removal is noted. From the results the condition of each filtering element can be judged.

No hard and fast rule can be given for the life of filtering elements, as it depends on the amount of dirt in the fuel and the conditions of operation. Engine manufacturers usually issue rules for the intervals between filter-element renewals, in terms of either hours of engine operation or of gallons of fuel consumed. However, they also advise users—and especially fleet owners—to determine the optimum intervals under their particular operating conditions experimentally, and to then make filter renewal a routine operation, preferably timed to coincide with other service operations.

Air Cleaners—All automotive Diesel engines are provided with air cleaners. These are of various types, some involving the use of strainer cloths, while others contain wire, steel wool, or fibers wetted with oil, which latter attracts the dust particles in the air and holds them. As these air filters become filled with dust, they must be cleaned, in order to maintain their efficiency and prevent loss of engine power due to choking of the air passages.

Fig. 5 is a sectional view of an air cleaner of the oil-bath type. Air enters through the perforations in the cap and passes down the central tube to the pool of oil at the bottom, where it is turned upward and passes through the filtering element to the outlet. Some of the dust in the air is adsorbed by the oil in the pool, and more adheres to oil bubbles that are carried up by the air current. A wire screen prevents the contaminated oil from getting into the engine cylinders. The filtering element can be readily cleaned in gasoline, but it is important to let it dry fully before reassembling it, as otherwise it might cause a "runaway" of the engine. Connections from the air cleaner to the engine must be kept air-tight, especially if the engine has to operate in a dust-laden atmosphere. In addition to the air cleaner on the inlet manifold, there generally is one also on the crankcase breather, and that also requires periodic inspection to see that it is in proper operating condition.

Priming Fuel Line—If the engine has been out of service for some time, air may have entered the fuel line, in which case the system is said to be "air-bound." The fuel system also may become "vapor-bound," especially if the engine is shut down after a continued hard run, which may have raised the fuel-carrying parts of the system to a temperature sufficiently high to vaporize some of the fuel. The vapor, of course, will condense again when the parts cool, but on doing so it may draw air into the system.

Air in the high-pressure line does not necessarily prevent starting of the engine, provided it has not entered the injection pump. In one installation fuel lines occasionally broke and had to be replaced without shutting down the engine. There was no opportunity to bleed the line after it had been replaced, because injection began almost immediately after the pump started to operate again. During the change of the line the pump was cut out. The fuel line was about 8 ft long, the pump plunger had a diameter of 22 mm, and the injector was set to open at 4000 psi. This shows that if the injection pump continues to operate, the fuel delivered by it will force the air out of the line. However, the usual practice is to prime the system after the line has been opened.

Hand-operated priming pumps are now generally provided, either on the transfer pump or elsewhere in the system. Individual units of the injection pump can be worked for prim-

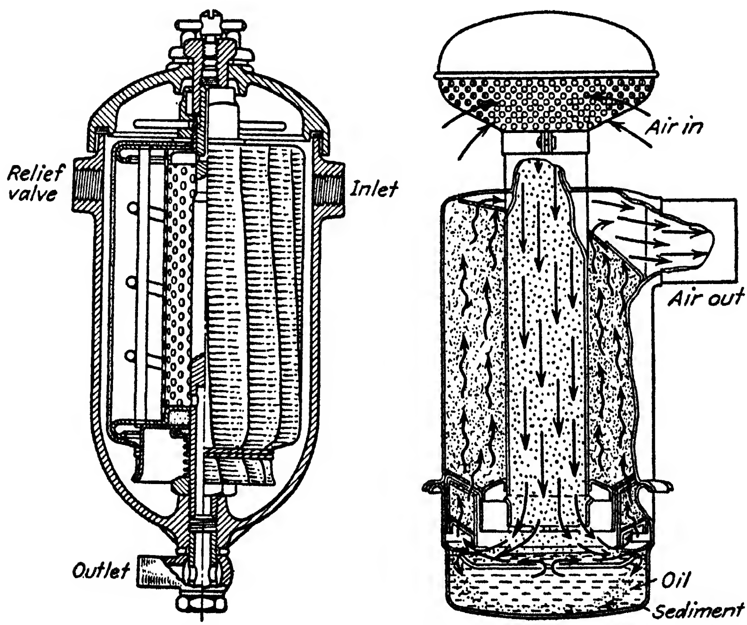


FIG. 4 (*Left*).—BAG-TYPE FUEL FILTER.
FIG. 5 (*Right*).—OIL-BATH-TYPE AIR CLEANER.

ing purposes by hand by means of a screwdriver. The cover plate in the side of the pump is removed, the point of the screwdriver is inserted under the spring washer of the pump plunger, and the plunger is then lifted by using the lower edge of the opening in the housing as fulcrum. The plunger is returned automatically by its spring.

In working on the fuel system to remove an air lock, the operator always should have his hands absolutely clean. If the hands are grimy, particles of grit will adhere to them starting the work, all connections of the fuel system should be washed off. The first thing to do to eliminate an air lock is to disconnect the fuel line from the transfer pump (if one is provided) and allow the fuel to run out until no air bubbles show in it. If there is a strainer with sediment chamber in the line, that should be drained, and if there is a screen in the connection to the transfer pump, that should be cleaned, or at least it should be checked to see whether it needs cleaning. Assuming that there is a main fuel filter between the transfer pump and the injection pump, the vent screw in the top thereof should be opened and fuel forced from it, by either operating the priming lever or cranking the engine, until there are no signs of air bubbles in the fuel stream. If there is no vent screw in the main filter, the fuel outlet from the filter must be disconnected, fuel forced through the opening until it is free of air bubbles, and the connection reestablished.

Next the pump is deaerated by loosening the vent screw or opening the petcock near the delivery connection and cranking the engine or working the priming lever until no air bubbles show in the oil flowing from the pump. Then the petcock or the vent screw is closed. The final step consists in venting the high-pressure lines and the nozzles. Some nozzles are provided with vent screws, but the majority do not have them. If there is such a vent screw it may be opened; otherwise the connection of the fuel tube to the nozzle is just sufficiently loosened so that it will leak oil when the engine is cranked. Here, too, the engine is cranked over half a dozen times or so, or the priming lever is given a similar number of strokes, by which time all of the air will have been expelled from the system, so that the vent screw can be closed or the connection of the high-pressure line to the nozzle tightened again. The engine should now be in condition to start.

As already mentioned, before the fuel system is opened at any point, the parts of the system near the joint should be thoroughly washed with fuel oil or kerosene. After the line has been disconnected, the openings should be plugged or capped to prevent dirt from entering. Cork plugs have been found suitable for the purpose.

Preventing Air Locks—Air naturally is most likely to get into the system at the suction line from the tank to the transfer pump, as this is under less than atmospheric pressure when the engine is running. The connections of this line therefore should be frequently inspected for tightness. If this tube shows any defects, it is generally the best plan to replace it with a new one, as any possibility of air locks in this part of the system is likely to give constant trouble. Among other causes of air lock may be mentioned too small a suction pipe, too little head on the fuel supply, a leaky spill-valve stem, and wax-choked fuel filters.

Cleaning Fuel Filters—All fuel filters and fuel strainers must have their filtering or straining elements either replaced or cleaned periodically, because if this is not done, they will become choked so that the fuel cannot flow through them freely. Cleaning is generally accomplished by washing the element in kerosene or fuel oil, which must be clean, of course. All fuel lines and also the fuel tank must at all times be kept clean, and it is not permissible to use cotton waste for this cleaning operation, as linters would be likely to get into the system. A clean, non-fraying cloth must be used. As the system is always air-bound when it is allowed to run dry, a good deal of trouble can be avoided by always replenishing the fuel supply in time.

Testing Nozzles—Among the parts most likely to need periodic attention are the injectors or nozzles, whose tips are exposed to the heat and corrosive effects of the gases of combustion, and whose stems are such a close fit in the nozzle bodies that the slightest heat distortion or the entrance of very minute particles of dirt may cause them to bind. If one of the nozzles of an engine fails to operate properly, the cylinder affected will produce little or no power, and a method of locating the faulty nozzle has been developed which is similar to that of locating a faulty spark plug in a spark-ignition engine. With the latter, if irregular firing of the engine indicates that one of the plugs is not producing a good spark, the operator idles the engine and then short-circuits one spark plug after another by means of a screwdriver. If a "good" plug is short-circuited, the engine will slow down, whereas if a faulty one is short-circuited it will not affect the speed of the engine. Some nozzles are provided with a bleeder valve, which can be opened and closed by means of a cross bar on its stem. If one of these bleeder valves is opened while the engine is idling, the particular cylinder receives no fuel. If that does not affect the speed, it shows that the cylinder receives no fuel (or at least not enough fuel) also when the bleeder valve is closed, and indicates that something is wrong with the nozzle. If the nozzles used have no bleeder valves, this test can be made only by disconnecting the fuel lines at the nozzles.

Nozzles can be tested also by causing them to inject into the atmosphere—by cranking the engine—and observing the spray pattern, which should be symmetrical. A satisfactory and a poor full-load spray pattern are shown in Fig. 6. The first is cone-shaped near the nozzle tip, with a rather dense core surrounded by a foggy shroud. At some distance from the tip the core dissolves, but the misty shroud retains its symmetry. In contrast to this, the faulty spray is jagged and unsymmetrical. Such a faulty spray may be due to injury to the pintle or to wear or corrosion of the orifice. Crater formation around the orifice also causes distortion of the spray pattern. Unless good service equipment is available, little can be done to restore the serviceability of such faulty nozzles in the field, and the best plan is to replace them with new ones and send them to the factory for repair.

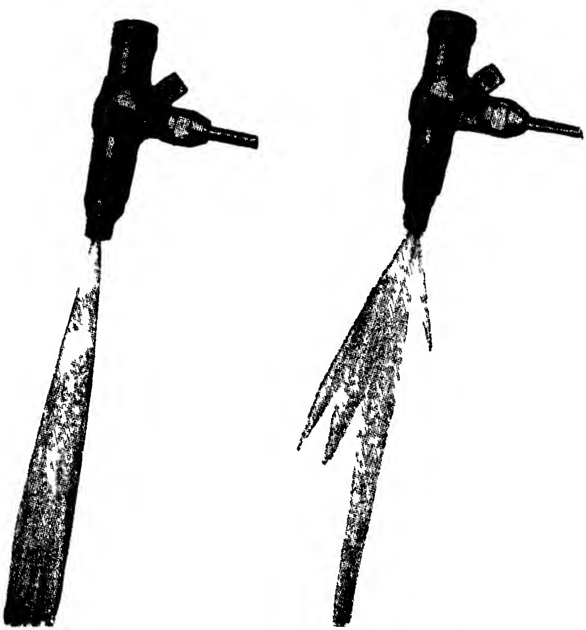


FIG 6—SHOWING A GOOD (*Left*) AND A BAD (*Right*) SPRAY PATTERN.

Fig. 7 shows the idling-spray pattern of a nozzle in serviceable condition, and also the spray or drip of a leaky nozzle, the curved path of the fuel indicating that there is little pressure behind it.

Servicing Nozzles—After a nozzle has been removed from the engine it must be taken apart, and in doing this it is advisable to follow the maker's instructions regarding disassembly and assembly. Nozzle valve and bushing (or body) should be cleaned in kerosene or fuel oil. These parts must never be worked on with hard tools, emery cloth or emery powder, as any scratches on them would be almost sure to cause leakage. After the nozzle sleeve has been soaked in gasoline for some time, its inside can be cleaned with a small stick dipped in gasoline or fuel oil and then rubbed off with a soft non-fraying rag. The orifices of the nozzle can be cleaned with a special tool (Fig. 8) obtainable from the manufacturers, and in some cases with a fine steel wire. Whenever a nozzle is replaced in the engine, the clamping plate should be tightened only after the nozzle has been tested by spraying into the atmosphere, or has been primed as explained in the foregoing.

FIG. 7.—NORMAL IDLING-SPRAY PATTERN AND PATTERN OF SPRAY FROM LEAKING NOZZLE.

If the nozzle valve is found to be stuck in the bushing, the nozzle must be removed from the engine, connected to the pressure line from the pump, and allowed to spray into the atmosphere. If the engine is turned over by the starter, the nozzle must give a fine spray. If it does not do so, then it must be taken apart and the parts (valve and bushing) soaked in gasoline and cleaned as explained above. If this does not cure the trouble these parts must be replaced by new ones. If the orifices are clogged the part containing the orifice or orifices (sometimes referred to as the burner plate) must be removed and the orifices cleaned out with the special tool or fine steel wire as explained. If necessary, the orifice plate must be replaced by a new one.

Nozzle Test Stand—When a nozzle has been taken apart and is reassembled, it is necessary to adjust it for valve-opening pressure. This can be properly done only with the aid of a testing fixture or test stand, which is supplied by the makers of the nozzle. The Bosch nozzle test stand, illustrated by Fig. 9, consists essentially of a fuel-injection pump mounted on a base plate, which can be actuated by hand by means of a lever Oil is supplied to the pump from a small tank in which there is a replaceable filter A globe valve

Fig 8—Cleaning Nozzle Orifices with Special Tool (General Motors)

with a pressure gauge reading up to 5000 psi is mounted above the injection pump. Connected to the globe valve is a double-ball check valve, to which the high-pressure discharge line is attached. When a nozzle is to be tested it is secured to the high-pressure line and centered over the fuel tank, after the lid has been removed from the latter, and the pump handle is then operated.

When a nozzle is allowed to spray into the atmosphere, great care must be taken that the spray does not hit the finger or hand of the operator, as great injury might be caused by the jet, owing to its high pressure.

If the fuel filter in the connection to the nozzle is clogged, the connection must be loosened, the filter element removed, its orifices carefully cleaned (with steel wire according to the recommendations of one manufacturer), and the element washed in fuel oil. Nozzle valve and nozzle bushing are never replaced separately, because a very accurate fit is required between them, and every owner should have one or more of these assemblies on hand so as to be able to make an immediate replacement in case of an emergency. When the nozzle

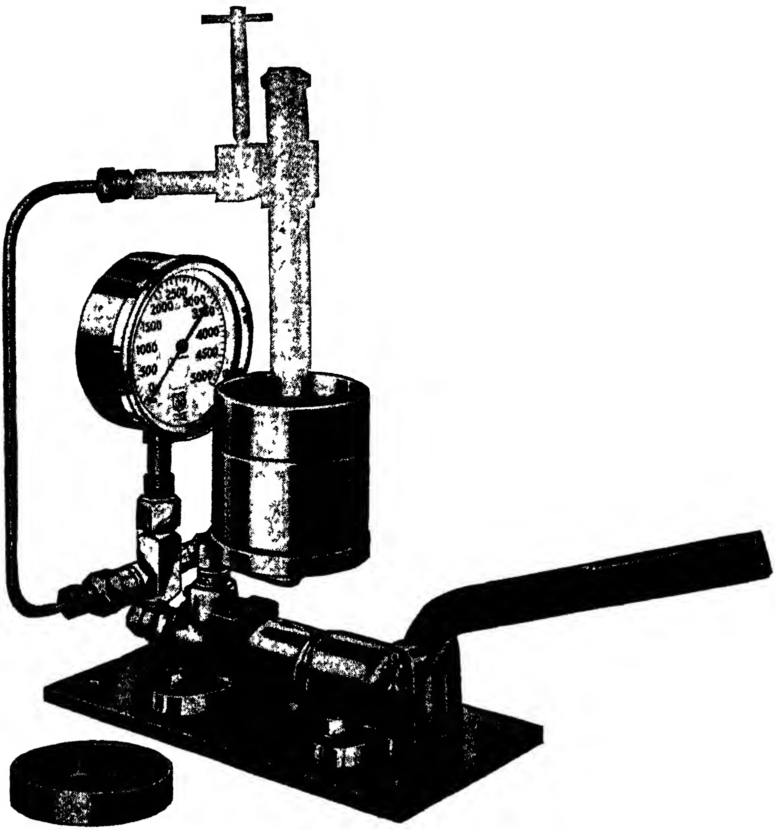


FIG 9—AMERICAN BOSCH NOZZLE-TESTING STAND.

is being replaced in the engine, care must be taken to see that the two nuts with which the clamping plate is forced down against the shoulder on the nozzle body are drawn up uniformly. If this is not done, the nozzle body may be distorted and the valve jammed in the bushing.

Retiming Fuel Pump—If the fuel pump has to be removed from the engine for any reason, the pump driving coupling must be disconnected, of course, and before doing this it is advisable to make marks opposite each other on both the driving and driven members of the coupling, which permits of reestablishing the proper timing without difficulty when the pump is being replaced. If no such timing marks have been made and it is desired to time a pump on an engine, the procedure is as follows: Timing is always effected by means of cylinder No. 1 and its corresponding pump unit. The crankshaft of the engine is therefore turned until the piston in No. 1 cylinder is in dead-center position at the end of the compression stroke. The compression stroke is determined from the fact that this is the up-stroke during which both valves are closed. The dead-center point is indicated by a mark on the flywheel which must be brought in direct opposition to a stationary pointer or mark carried by the engine. Next the crankshaft is turned back through an angle which corresponds to the lead of injection on the particular engine, which will be given in the instructions accompanying the engine. In a high-speed engine this is generally of the order of 15 to 20 crankshaft degrees. The point at which injection begins may be marked on the flywheel; if it is not, it may be found by measuring off on the circumference of the flywheel, from the top dead-center mark, in the direction in which the flywheel turns in operation, a distance corresponding to the injection-advance angle.

The injection pump is now fastened to its base or flange, but the driving coupling is left loose. The suction line and all of the high-pressure fuel lines, except the one to the nozzle in No. 1 cylinder, are connected up. With the control lever in the wide-open position the pump is primed. Next the control lever is moved to the stop position, and No. 1 delivery-valve holder is removed, the delivery valve and its spring are taken out, and the holder is replaced finger-tight. The control lever is now returned to the full-load position, whereupon fuel should issue from the delivery-valve holder when the transfer pump is operated. If it does not do so, the pump shaft is turned by means of its coupling half until the fuel begins to flow. It is then moved back to a position where there is no fuel flow. In this manner the pump shaft is turned back and forth a number of times until the position is found where the fuel delivery just begins, and the coupling is tightened up in this position. This corresponds to the point of the cycle where the pump plunger closes the fuel port and pressure begins to build up in the pump barrel. After the delivery valve and spring have been replaced, No. 1 fuel line is connected to its nozzle, all fuel lines are primed, and the engine is started.

Maintenance of Pump Capacity—Attention was called in the foregoing to the fact that an injection pump must accurately meter the charge of fuel not only while the pump is new, but must continue to do so over a long period of use. Evidence of the continuance of performance of such pumps is given by Fig. 10 supplied by the Bosch company, which shows the delivery of the pump per stroke as a function of the governor position when the pump was new, and also after it had served on a motor truck during 80,000 miles of operation. Naturally, the wear caused a decrease in the delivery, but the change is practically negligible.

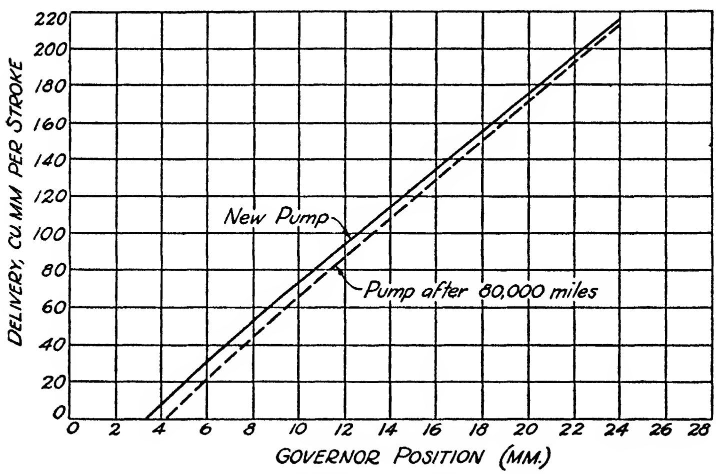


FIG. 10.—DELIVERY CHARACTERISTICS OF PUMP WHEN NEW AND AFTER 80,000 MILES OF TRUCK SERVICE.

Among the most delicate parts of the pump are the valves and valve seats, if valves of poppet or ball type are used. The wear and tear on these valves is naturally higher the greater the unit pressure on their seats. For long life of the valves it would therefore be desirable if the seats could be made wide, but in order to prevent leakage under high pressure the seats must be made narrow. Satisfactory life must then be assured by using hardened valves and valve seats.

Fuel Stop—All engines are provided with a fuel stop, which determines the limit to which the control rack of the fuel pump can be moved by the governor or by hand control.

To prevent abuse of the engine by drivers, this stop on some engines is sealed so it cannot be tampered with. This makes it impossible to feed the engine an excessive amount of fuel, and prevents smoke in the exhaust. One inconvenience of such a fixed stop is that it may make it impossible to get an adequate charge of fuel into the engine cylinders while cranking, because of the low volumetric efficiency of the pump at low speeds. A number of automatic or semi-automatic devices which overcome this difficulty have been developed in recent years. They are known as excess-fuel devices in Great Britain and as enriching devices in the United States. Some of the earlier ones involved hydraulic principles, the pressure of the lubricating oil or that produced by the fuel transfer pump being made use of to move the stop into position after the engine had started. These pressures are very low while the engine is being cranked, and the fuel stop is then held out of the way of the rack by a spring. Most of the later devices are purely mechanical in principle, however. One such excess-fuel device was illustrated and described in Chapter VII.

Large fleet operators determine the proper setting for the stop by means of exhaust-gas analyses. With increase in the fuel charge, the percentage of carbon dioxide in the exhaust increases, and for each design of engine the maximum fuel charge which can be burned with clean exhaust corresponds to a certain proportion of CO_2 in the exhaust.

Lubrication Service—Whereas in the gasoline engine there is a tendency for the oil in the crankcase to become thinner or less viscous with use, owing to dilution by fuel leaking by the pistons, in a Diesel engine the oil usually becomes more viscous with use, on account of oxidation (sludging). Owing to this thickening of the oil in service, it is necessary to flush the crankcase from time to time. After the old oil has been drained off, a certain amount of flushing oil is introduced into the crankcase through the filler, and the engine is then idled at low speed for a few minutes. The flushing oil is then drained off, and the oil filter should be cleaned or flushed at the same time. Some manufacturers recommend washing the crankcase out with the flushing oil, using rags (not waste) for the purpose. This obviates idling the engine on the flushing oil, which operation involves a certain risk because of the low lubricating value of that oil.

Aside from the engine itself, its accessories must be properly looked after with respect to lubrication. The fuel injection pump usually requires a little engine oil added to the camshaft housing at intervals. There is a bayonet gauge in this housing, by means of which the oil level can be checked. The water pump sometimes is grease-lubricated, in which case the cap of the grease cup must be given a turn now and then, while the radiator fan, if it is separate from the pump, also requires attention.

Cooling System—Diesel engines operate to best advantage at the maximum temperature which gives a fair margin of safety from loss of cooling medium by boiling. It is especially desirable that the engine block be brought quickly to near its normal working temperature when starting from cold. One reason for this is that after the engine has stood idle for a considerable period, there is likely to be very little oil on its bearing surfaces, and the oil film therefore has very little cushioning effect, which is particularly needed under these conditions because of the high bearing load due to the heavy combustion pressure.

To assure the maintenance of high jacket temperatures, most engines are provided with thermostats in the outlet from the engine water jacket, which limit the outlet temperature to between 150 F and 170 F. A sensitive type of thermostat (the balanced type) is best for this purpose, and it should be so arranged that when the jacket temperature is below the opening point of the thermostatic valve, the water circulates through a by-pass.

It is also recommended that radiator shutters of either the hand-controlled or the automatic type be installed. Where the shutters are hand-controlled, an engine-temperature indicator must be installed on the dashboard, as a guide to the operator in setting the shutters. Diesel engines of a certain output throw less heat into the cooling system than equivalent gasoline engines, hence there is a tendency to overcooling if a radiator core and a fan of the same size are used.

In a good many sections of the country the hydrant water contains much lime, and when heated to near the boiling point, this lime is deposited in the passages, so that the water spaces of the radiator core are clogged, decreasing the efficiency of the cooling system. In such districts it is well to refill the cooling system with soft water from cisterns or to treat the hard water with a suitable softening preparation (boiler compound).

Safety Shut-Down Valve—Large industrial engines sometimes are provided with automatic safety shut-down valves to protect them from injury. One such valve, which comprises two Fulton Sylphon bellows, will shut down the engine when either the water temperature at the jacket outlet becomes excessive, or when the pressure on the lubricating oil drops below the safe limit. Both bellows act on a valve in the fuel line. The pressure of the lubricating oil, which enters one of the bellows, holds the fuel valve open. When the engine is being started there is no pressure in the lubrication system, and the operator then holds the fuel valve open by means of a lever projecting from the housing of the safety valve. As soon as the oil pressure reaches 10 psi the lever can be released. When the engine coolant is at normal temperature there is no pressure in the other bellows, which is connected to a temperature bulb in the engine cooling system, and the elastic force of the bellows then holds it in the "open" position. When the jacket temperature exceeds 205 F, vapors are formed in the temperature bulb, and the pressure of these vapors expands the bellows and closes the fuel valve, shutting down the engine.

Loss of Compression—Loss of compression is a particularly serious fault in a Diesel engine, as ignition is dependent on compression, and if the seal in one of the cylinders is faulty it is likely to miss. The trouble may be due either to improper functioning of the piston rings or to excessive wear of the cylinder bore or liner. Leaky valves also will cause an engine to lose compression. Some trouble has been caused by the oil drain holes or slots in the scraper rings closing up with carbon, and this problem is receiving consideration. It may lead to the use of wider scraper rings permitting of larger drain holes. Excessive emission of blow-by from the breather pipe of the engine is a good sign that the piston seal is defective and that either the piston rings need attention or that the cylinder requires regrinding or the installation of new liners. The increase in oil consumption has been used as an index of the need of reconditioning of cylinder bores. Thus an engine used in large numbers by a public-service corporation in England has an oil consumption of one gallon per 1000 miles, approximately, when in good condition. When the oil consumption has increased fourfold, that is, when it has reached a figure of one gallon per 250 miles, the engine is sent to the shop for overhauling.

Measurement of Blow-By—The condition of the pistons and rings can be judged also by measurement of the rate of blow-by, which is made by means of a gas meter, gasometer, or venturi and pressure gauge connected to the breather opening of the crankcase, any other openings of the case being closed. Tests made in the laboratory of the French Air Service showed that in aircraft engines in good condition, the blow-by amounts to between 0.1 and 0.5 per cent of the engine's displacement, at normal operating speed and with lubricating oil and cooling water at a temperature of about 175 F. If the blow-by is as high as 1 per cent, the seal is inadequate, and this may be due either to wear or to stuck piston rings. Tests with numerous different fuels and lubricants led to the conclusion that sticking of piston rings is more or less independent of the qualities of these supplies and is due mainly to excessive temperature of the piston, too rapid heating of the piston, or delay in the supply of new lubricant to the rings when the engine is started up from cold. Stuck rings are more troublesome in a new engine than in one which has seen considerable service, which is probably due to the fact that in the latter the clearance of the rings is greater.

Checking Valve Timing—When a Diesel engine has been taken apart and is being reassembled, great care must be taken that the timing is absolutely correct before an attempt is made to run the engine. The reason for this is that in many designs, especially those with turbulence chambers, the piston comes within a very short distance of the valve head, of the order of $\frac{1}{32}$ in., when at the end of the up-stroke, and as the valve usually has a lift of $\frac{3}{8}$ or $\frac{7}{16}$ in., piston stroke and valve lift overlap a good deal, so that if the timing gear were even one pitch out of time, the valve would be likely to interfere with the piston motion and some part would have to yield (the rocker arm bend, for instance). The very small and somewhat variable clearance between piston and valves is a potential source of trouble. Several Continental manufacturers guard against such trouble by either sinking the valve seat some distance into the head, providing a clearance around it, or else machining a cylindrical recess in the piston head under each valve.

General Inspection—All nuts and bolts on the outside of the engine must be checked at intervals for tightness, and tightened when found loose. This applies particularly to the cylinder-head-stud nuts, oil-pan-stud nuts, bolts or nuts holding valve-timing gear and other covers in place, and also the engine mounting bolts. When the engine leaks oil while standing, it is a good sign that some of the nuts or bolts are not drawn up properly. Cylinder-head-stud nuts should always be checked for tightness with the engine hot, so as to make sure that the cylinder head gasket is tight with the engine in running condition.

CHAPTER XIX

Gas Turbines

During the past two decades the gas turbine has played an important role in the research and development programs of several large industrial concerns. Development work on gas turbines first began half a century ago, after the steam turbine had gained a firm footing, but no practical results were achieved at the time. During World War I there arose a demand for superchargers for aircraft engines. Most of the early superchargers or blowers were driven directly from the engine, but Rateau in France proposed to drive them by turbines operated by the exhaust from the engine. That line of development was taken up in this country by General Electric Company, and it gave us the turbocharger, which combines a gas turbine with a centrifugal compressor. Such turbochargers are now extensively used to supercharge large aircraft and Diesel engines. In this application the turbine operates on what would otherwise be waste energy, and its low thermal efficiency, which had precluded success in other fields, therefore was no serious handicap. The turbocharger, however, is merely an auxiliary and not a prime mover that can be built for any desired output, which had evidently been the object of the pioneers in the gas-turbine field.

Another recent application of the gas turbine also has to do with aviation: The powerplants of jet-propelled planes combine all of the same elements as the simple gas turbine used to generate power. The only difference between the two is that whereas in the latter all of the compressed and heated air passes through the turbine, in the jet-propulsion powerplant only enough passes through to develop the power necessary to drive the compressor, the remainder being bypassed and discharged toward the rear to generate the propelling thrust.

A new start in gas-turbine development was made in Switzerland during the early thirties by two large industrial concerns, Escher, Wyss & Co. and Brown, Boveri & Co. The work started by Brown-Boveri was taken up in this country by the Allis-Chalmers Manufacturing Co., which has made a considerable number of gas-turbine installations in petroleum refineries using the Houdry catalytic cracking process. In that process a large amount of air under considerable pressure is needed to regenerate the catalysts. Fuel is burned in this process, and the mixture of air and products of combustion, after it has served its purpose, is at a high temperature and considerable pressure, so that it contains a good deal of energy. Therefore, instead of being discharged directly into the atmosphere, it is passed through the turbine, which generates the power necessary to drive the compressor that furnishes the compressed air for the process.

Development work on gas turbines has been taken up by at least five other concerns in this country. The Brown-Boveri firm in Switzerland built a gas turbine-electric generating plant for a Swiss municipality as a stand-by for use in the event of air raids. An advantage of the gas-turbine plant for this purpose is that it requires no water for its operation. The same firm also designed a gas turbine to deliver 60,000 cfm blast air at 30 psi pressure for a steel works. In that case blast-furnace gas is used as the fuel. This normally is also a waste product. More recently most of the development work has been on turbines for locomotives.

Types—There are essentially two types of gas turbine, constant volume and constant pressure. The turbine of the turbocharger works on the constant-volume principle. The fuel which furnishes the energy on which this turbine operates is burned in the cylinders of the engine substantially at constant volume while the piston is at and near the end of the stroke. In that case the temperature of the gases of combustion is lowered by expansion in the engine cylinder, and the exhaust can be admitted to the turbine without danger of injuring the blades. If the cylinder in which the combustion occurred served only to supply working fluid to the turbine, communication with the latter would have to be established immediately after combustion had been completed and before there had been any appreciable cooling or expansion. At that moment the gases are at a temperature which is much too high for the turbine blades to withstand. The blades might be protected by injecting atmospheric air along with the gases of combustion, but it would certainly be extremely difficult to ensure a satisfactory life for the valve in the communicating passage between combustion chamber and turbine.

At present practically all of the development work is being done on the constant-pressure type. A diagram of a simple constant-pressure gas-turbine plant is shown in Fig. 1. It comprises a centrifugal compressor which compresses atmospheric air to the operating pressure; a combustion chamber in which some of the compressed air is used to burn fuel and in which the products of combustion are mixed with the remaining air; and a turbine in which the working fluid thus prepared is expanded to atmospheric pressure, and in which power is generated. Part of this power is used to drive the compressor and the remainder is the net output. The installation includes an electric motor which serves to start the turbine and compressor.

Working Pressures and Temperatures—In considering the problems of the constant-pressure turbine, the question

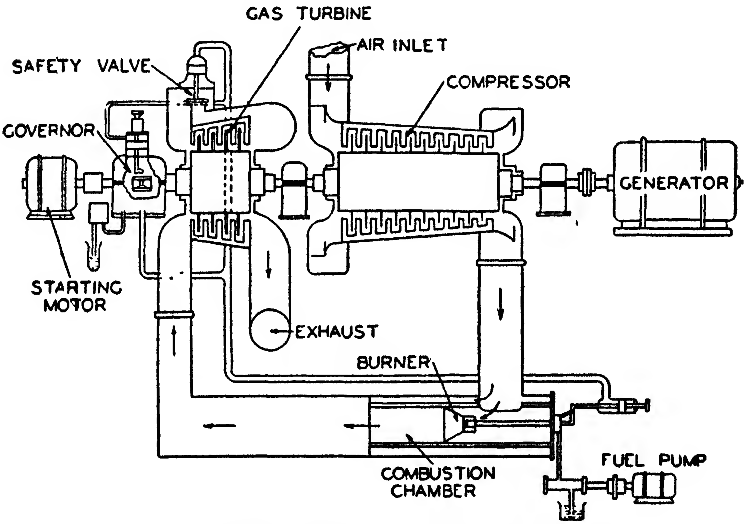


Fig. 1.—Diagram of Simple Gas-Turbine Plant.

of the optimum pressure and temperature at once arises. The objects aimed at are, of course, a high output from a machine of given size, and a high overall efficiency. The specific output is measured by the work done by one pound of air passing through the machine. Both the specific output and the efficiency increase with the temperature at the inlet to the turbine, but this temperature is limited by the stresses to which the working parts are subjected. The parts affected most by the heat are the turbine blades, which are subjected also to centrifugal and inertia stresses. Metals having relatively high strengths at high temperatures are used for these blades, but the best materials at present available limit the inlet temperature to about 1100 F if the machine is to be reliable in service and show a satisfactory life.

The best operating pressure varies with the efficiencies of the compressor and turbine. The higher these efficiencies the higher will be the optimum pressure. It seems that for compressor and turbine efficiencies of about 85 per cent a pressure ratio of 4 gives best results. Atmospheric air is then compressed to four atmospheres absolute (60 psi abs or 45 psi gauge). Designers naturally aim to increase the mechanical efficiencies of the compressor and turbine, and with them the optimum pressure ratio.

Operating Cycle—In Fig. 2 is shown what may be called a cycle diagram of a simple gas turbine. It shows the changes in volume and pressure a pound of air undergoes while passing through the machine. We start at A with one pound of atmospheric air at 60 F which, as the diagram shows, occupies a space of 12.8 cu ft. We pass this air through the compressor, in which it is compressed adiabatically to 60 psi abs. The volume occupied by the air at the end of compression can be found from the equation $pv^{1.4}$ = constant. Since we know the values of p and v at the beginning of compression, we can calculate the value of the constant:

$$15 \times 12.8^{1.4} = 532.5.$$

Since the absolute pressure at the end of compression is 60 psi, we have

$$60 \times v^{1.4} = 532.5,$$

which when solved gives $v = 4.76$ cu ft.

From the compressor the air passes to the burner without change in temperature or pressure. In the burner heat is added until the air reaches a temperature of 1000 F or 1460 deg abs, the pressure remaining constant. When air is heated at constant pressure its volume increases in the same proportion as its absolute temperature, hence at point C, the turbine inlet, the volume of the gas is

$$\frac{1460}{770} \times 4.76 = 9.00 \text{ cu ft.}$$

In the compressor the gas expands adiabatically from C to D. The ratio of volumetric expansion in the turbine is exactly the same as the ratio of compression in the compressor, and the volume at D therefore is

$$\frac{12.8}{4.76} \times 9.00 = 24.2 \text{ cu ft.}$$

The temperature of the gas at the end of expansion can be readily found from the equation relating the pressure, volume and temperature of 1 lb of air, viz.,

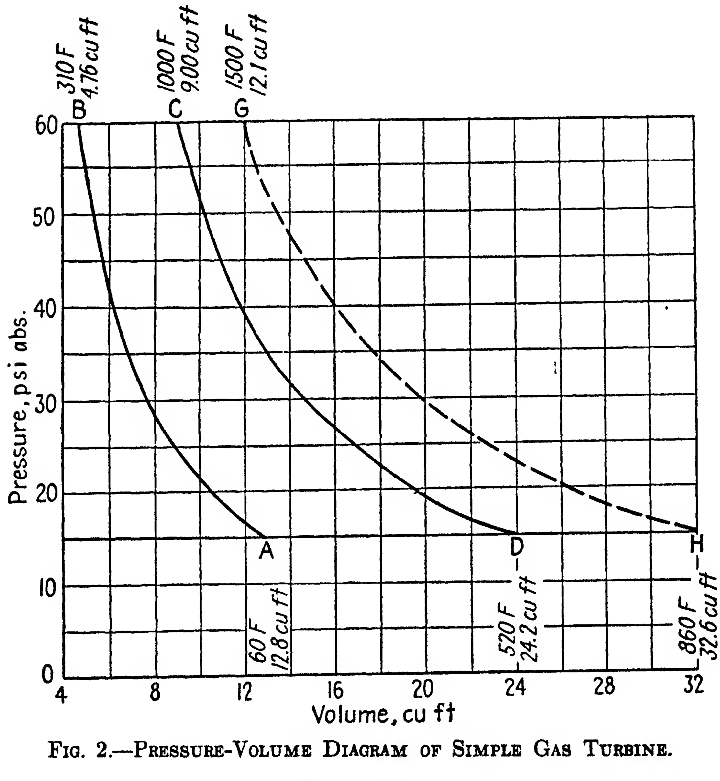


FIG. 2.—PRESSURE-VOLUME DIAGRAM OF SIMPLE GAS TURBINE.

$$T = 2.7pv,$$

where T is the absolute Fahrenheit temperature; p, the absolute pressure in psi, and v, the volume in cu ft. For point D we get

$$T = 2.7 \times 15 \times 24.2 = 980 \text{ deg abs or } 520 \text{ F.}$$

At the turbine exhaust, therefore, the pressure is back to atmospheric, but the temperature of the gas is still quite high.

In order to show how an increase in the turbine-inlet temperature will increase the specific output and the efficiency, an expansion curve for a turbine-inlet temperature of 1500 F is also shown in Fig. 2. Heating the gas to 1500 F will expand it to 12.1 cu ft and at the end of expansion it occupies 32.6 cu ft and has a temperature of 860 F. At any particular pressure the volume increase is 72 per cent over that with an inlet temperature of 1000 F, and the mechanical work done therefore is 72 per cent greater. It also takes 72 per cent more heat energy to raise the air from 310 F to 1500 F than to raise it from 310 to 1000 F. However, the heat supplied by the burner is only part of the energy imparted to the air, an equally important part being supplied by the compressor. But since the pressure remains the same, the work done by the compressor is the same for 1000 F and 1500 F turbine-inlet temperatures, and the ratio of energy available to energy supplied therefore is greater with the higher inlet temperature. As the gas expands through a much larger volume range, the output per pound of working fluid also is much greater with the higher turbine inlet temperature.

Fig. 3 shows the approximate variation of the overall efficiency with the turbine-inlet temperature for different combined efficiencies of the compressor and turbine. It might be pointed out that since the turbine output is greater than the compressor input, the efficiency of the former influences the overall efficiency more.

Economic Aspects of the Simple Turbine—When we compare the overall efficiencies of the simple gas turbine which Fig. 3 shows to be possible with present limiting operating temperatures and normal compressor and turbine efficiencies, with those of the Diesel engine, which in the larger sizes exceeds 35 per cent, we see that the turbine cannot compete under ordinary conditions. Improvements in the gas turbine are still possible, of course. Metals of higher heat strength may be found, and more efficient blade sections and angles may be discovered, but developments already have been carried pretty far in these directions. It is therefore likely that the simple gas turbine will remain confined to fields where fuel costs are a relatively small item, as in the case of emergency generating sets or where waste energy can be used, such as that of exhaust gases, process gases, or blast-furnace gases. One thing in favor of the gas turbine is that it will operate on the same low-grade fuels that are burned under steam boilers, whereas high-speed Diesel engines are somewhat critical with respect to fuel quality.

When only 18 per cent of the available heat energy in the fuel is usefully converted, 82 per cent is lost through various channels, and it is, of course, possible to recover some of these losses in part. Besides, the processes of compression and expansion can be rendered more efficient by carrying them out in two (or more) stages, with intercooling between stages during compression and reheating during expansion. Practical considerations limit the number of stages, and it is doubtful whether more than two stages will be used.

Regeneration—It has already been shown that the gases on being exhausted from the turbine are at a high tempera-

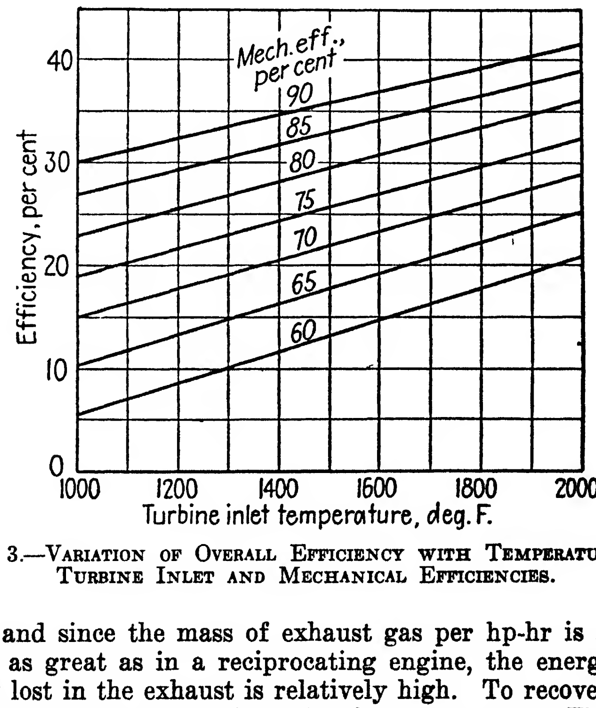


Fig. 3.—Variation of Overall Efficiency with Temperature at Turbine Inlet and Mechanical Efficiencies.

ture, and since the mass of exhaust gas per hp-hr is several times as great as in a reciprocating engine, the energy normally lost in the exhaust is relatively high. To recover some of the exhaust heat, use is made of a regenerator. This consists of a heat exchanger through which the exhaust and the air from the compressor are passed. From Fig. 2 it will be seen that the air leaving the compressor is at 310 F, while with a turbine-inlet temperature of 1000 F the exhaust from the turbine is at 520 F, so that there is a temperature difference of more than 200 deg between them. This difference, moreover, increases with the turbine-inlet temperature. It is, of course, impossible to economically transfer all of the available heat from the exhaust, as that would require a very large heat exchanger. The heat exchanger offers some resistance to air flow and causes a pressure drop. This must be made good by the compressor, which must compress to a correspondingly higher pressure and consequently will consume slightly more power. This loss, however, can be minimized by careful design. It has been found that when operating at 1200 F turbine-inlet temperature, 75 per cent of the available heat in the exhaust is about the most that can be transferred.

Intercooling—The power required to compress the air, which normally represents considerably more than one-half the gross output of the turbine, can be reduced by effecting the compression in two stages and intercooling between stages. The power generated by the turbine depends on the mass of air passing through, and it can readily be shown that it takes less power to compress a given mass of air in a certain proportion if the temperature of the air is low. Let us assume, for instance, that we have two cylinders of 1 cu ft capacity each, both filled with air to atmospheric pressure, the temperatures of the air in the two being 0 F and 200 F, respectively. If the air in each cylinder is now compressed to ½ cu ft, the work of compression will be the same for both cylinders, as both the initial and final pressures are the same, as are also the displacements against these pressures (½ cu ft). But the cylinder in which the initial temperature was 0 F contains about 43 per cent more air, and for the same mass of air the work of compression therefore is some 30 per cent less in the case of the cool air.

Intercooling also calls for a heat exchanger, in which the air heated by the first stage of compression gives up some of its heat to water circulating through the device. According to F. K. Fischer and C. A. Meyer of the Westinghouse Electric & Manufacturing Co., a single stage of intercooling reduces the power required by the compressor by about 15 per cent.

Reheating—The same as energy can be saved by intercooling between successive stages of compression, the available output of the turbine can be increased by dividing the expansion in it into two stages and reheating the discharge from the first stage before admitting it to the next. It has already been explained that the efficiency of the turbine is higher the higher the inlet temperature, but that a limit is set on this temperature by the limited hot strength of the blade material. In axial-type turbines the gases always expand through successive stages, and in each stage there is a temperature drop. Since all of the blades are made of the same material, the strength of the blades of the later stages is not fully utilized. This can be remedied to a certain extent by reheating. In a gas turbine reheating can be effected by merely injecting fuel into the passage connecting one stage to the next, as the working fluid consists of about 85 per cent of air and therefore will sustain combustion.

A diagram of a turbine installation with regenerator, two-stage compressor with intercooler, and two-stage turbine with reheater is shown in Fig. 4. In Fig. 5 are shown the gains in efficiency which may be expected from regeneration, intercooling and reheating, overall efficiencies being plotted against turbine-inlet temperatures. The curves are based on an as-

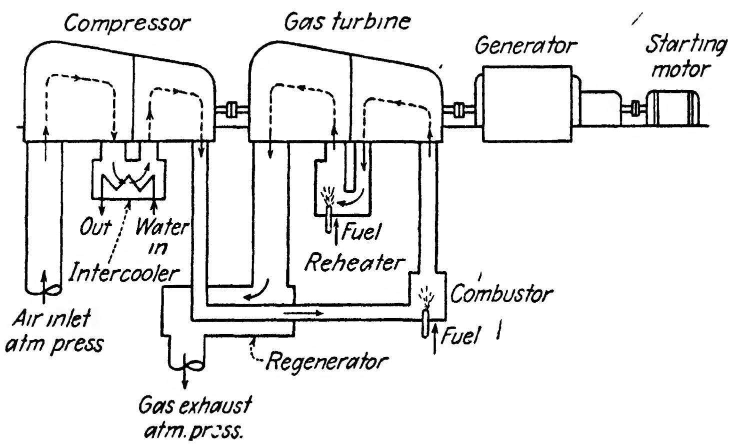


FIG. 4.—DIAGRAM OF GAS TURBINE WITH REGENERATOR, TWO-STAGE COMPRESSOR WITH INTERCOOLER, AND TWO-STAGE TURBINE WITH REHEATER.

sumed turbine efficiency of 85 per cent, a compressor efficiency of 84 per cent, an air-inlet temperature of 70 F, and a pressure drop of 5 per cent in the regenerator. Figs. 4 and 5 are redrawn from illustrations in a paper on The Combustion Gas Turbine by F. K. Fischer and C. A. Meyer of the Westinghouse Electric & Manufacturing Co.

Exhaust Boiler—Another way to recover some of the heat in the exhaust is by the use of an exhaust boiler, the steam generated in the boiler being used in a steam turbine. Drs. Davies and Fawzi of England, who made a thorough theoretical investigation of the subject, came to the conclusion that with an exhaust boiler the specific output of the installation, in ft-lb per lb of air, can be increased 27 per cent, and the overall efficiency 12 per cent, as compared with a simple gas turbine operating under the same temperature and pressure conditions. They also found that regeneration alone may increase the efficiency by as much as 40 per cent, but has little effect on the specific output. For maximum output the compressor and turbine must be compounded and provided with an intercooler and a reheater respectively, and the exhaust must be used in an exhaust boiler with steam turbine.

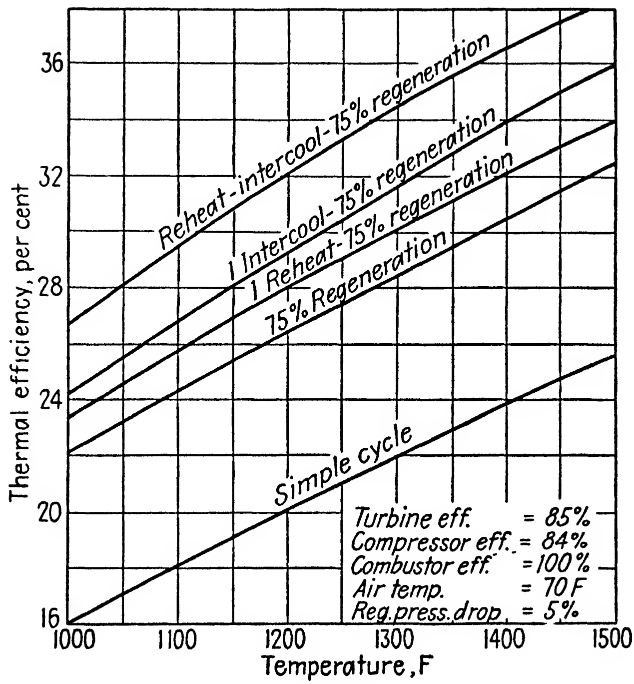


FIG. 5.—VARIATION OF EFFICIENCY OF VARIOUS SYSTEMS WITH TURBINE INLET TEMPERATURE.

If intercooling and reheating are carried to 100 per cent, the output will be increased 118 per cent, as compared with the simple gas turbine. On the other hand, for maximum efficiency the exhaust should be used for regeneration, and this combined with two-stage compression and expansion. With 100 per cent intercooling and reheating the efficiency will be increased 77 per cent and the specific output 55 per cent.

Closed Systems—Closed systems, in which the working medium is recirculated, have been proposed by Escher-Wyss and Westinghouse. One advantage of the closed cycle is that the initial pressure can be made quite high, and if the pressure multiplication in the compressor remains the same, the final absolute pressure and the mean pressure will be increased in the same proportion. The turbine output increases in proportion. In such a system, however, the gases leaving the turbine must be cooled before they are allowed to reenter the compressor, and rather bulky heat exchangers are required for the purpose. In the Escher-Wyss system the working fluid is heated by external combustion. As no gases of combustion pass through the system it is possible to burn coal, but the combustion efficiency naturally is lower than with internal combustion. The heater operates on the same

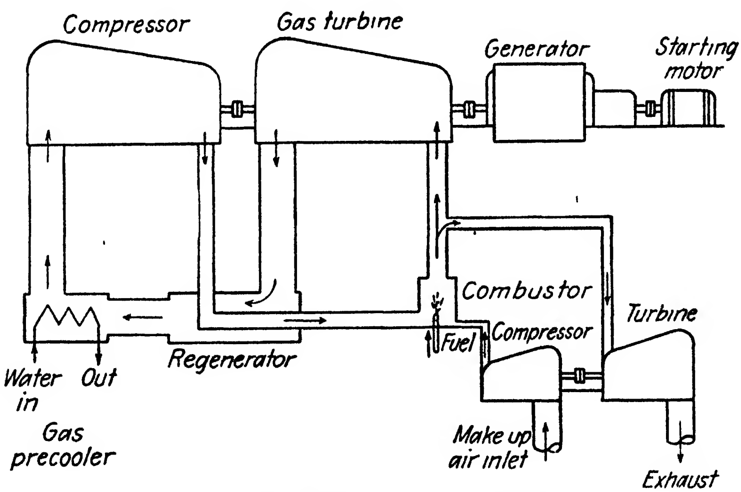


FIG. 6.—DIAGRAM OF WESTINGHOUSE CLOSED CYCLE.

general principle as the furnace of a steam boiler, but for the same rate of heat transfer the heating surface must be materially greater, because it is in contact with gaseous media on both sides.

A turbine of the closed-cycle type is under development by the Westinghouse company and is shown in diagram in Fig. 6. The unit is designed for a compressor-inlet pressure of 150 and a turbine-inlet pressure of 600 psi, or 10 times the pressures used in simple gas turbines. Internal combustion is to be used, and while the bulk of the fluid is being recirculated, enough fresh air is constantly added to support the combustion. This air is supplied by a separate small turbine-compressor unit, which also serves to maintain the pressure in the main system.

Where gas turbines are to be used on locomotives it is evidently necessary to provide a drive which picks up the load gradually and allows the turbine to be operated at full speed while the train is being started.

Road-Vehicle Applications—A number of experimental installations of gas turbines in road vehicles have been made. In England a Rover passenger car was equipped with one, and was shown at the British Industries Fair in 1948. It probably provided good publicity for the Fair, but nothing further has been heard of it since. In this country, under the sponsorship of the Navy Department's Bureau of Ships, a Boeing gas turbine was installed in a Kenworthy truck and subjected to tests which were the subject of an S.A.E paper.*

While the gas turbine possesses some very attractive features from the standpoint of vehicular propulsion, such as a remarkably low weight per unit of output, a very satisfactory torque curve showing a constant increase in torque with decrease in speed down to the stalling point, simplicity of construction (a small number of parts), freedom from vibration, and ability to operate on a wide range of fuels, its low thermal efficiency is an obstacle to its practical use in this field that it will be difficult to overcome. Tests of the Kenworthy truck (68,000 lb gwv, 175-hp gas turbine) extending over more than a year, showed a performance a little better than one mile per gallon of fuel. Diesel trucks of similar capacity will do better than 4 mpg under normal conditions. Some of the loss due to the lower fuel economy of the turbine is offset by a gain in payload capacity due to the lower weight of the turbine. The complete powerplant of the turbine truck weighs about 3000 lb less than that of a Diesel truck.

Application of turbine power to truck wheels involves certain complications. In the Kenworthy truck the turbine drives through a 9.06:1 reducing gear incorporated with it, through a reversing gear, through a seven-speed-and-reverse planetary transmission, and through a three-speed auxiliary transmission. One purpose of the reversing gear is to change the direction of rotation, the turbine rotating in a direction opposite to that of the conventional truck engine. In addition the gear makes it possible to use the turbine as a vehicle brake.

Gas-Turbine Locomotives—Gas-turbine locomotives with electric drive have been under development in the United

* Gas Turbine Propulsion for Ground Vehicles, by W. M. Brown, S.A.E. Quarterly Transactions, Jan., 1951.

States since the end of World War II by at least four concerns—Allis-Chalmers, Elliott, American Locomotive in association with General Electric, and Baldwin Locomotive in association with Westinghouse. An Alco-G.E. 4500-hp locomotive in 1949 was reported to have completed satisfactory preliminary runs and was then turned over to the Union Pacific Railroad for additional road tests in freight service. During the next 20 months it was operated in revenue freight service on three different railroads. At the end of that time it was withdrawn from service and a report on its performance was issued. It had operated more than 100,000 miles, most of it on the Union Pacific, where the average fuel consumption had been 4.2 gal per 1000 gross ton-miles. During the last two months, under somewhat more favorable conditions, this was reduced to 3.51 gal. The fuel economy was found to be highest at high-load factors, from which it was concluded that the gas-turbine locomotive is best suited to heavy through-freight service.

The Baldwin-Westinghouse locomotive, which operates on residual fuel oils, after preliminary tests in the yards of the Westinghouse company, was operated experimentally on the Pittsburgh & Lake Erie Railroad for some time, and was then withdrawn to have some changes made. In 1951 a coal-burning gas-turbine electric locomotive was under development at the Dunkirk, N. Y., plant of the American Locomotive Company, under the sponsorship of the Locomotive-Development Committee of Bituminous Coal Research, Inc.

Gas Turbine vs. Diesel Engine—The following evaluation of the prospects of the gas turbine in the field of power generation was presented at a meeting of Diesel-Engine manufacturers in Chicago in 1948 by Dr. John T. Retaliata, who was connected with Allis-Chalmers when that firm pioneered the large gas turbine in this country: "I don't think the gas turbine is going to put the Diesel engine out of business. In most applications the turbine is not a direct competitor. Up to 2500 hp the Diesel will have no serious competition from the turbine. In applications of from 5000 to 20,000 hp the gas turbine will have a field of its own."

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