INTERNAL-COMBUSTION ENGINES

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Theory, Design, Analysis, Application, Performance, and Economics

BY

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PREFACE

This book presents the theory, design, application, and performance of internal-combustion engines. It was written with the assumption that the reader is familiar with thermodynamics and mechanics; therefore it gives only a working review of the principles of these studies, with more specific treatment of those parts of thermodynamics and relationships in mechanics which are essential to a better understanding of the theory of internal-combustion engines.

The successful performance of an internal-combustion engine is essentially a problem of fuels and combustion; consequently, considerable material has been included on the properties of fuels and the chemistry of their combustion. The fuel problem is a significant one, particularly with the present trend toward increasing engine rotative speeds. The effect of the shape of the combustion chamber and methods of supplying the fuel to the engine cylinder, which are of much importance in connection with the combustion process, have been emphasized.

Brief descriptions of the design and operating features of representative engines have been given in order that the reader may obtain a better understanding of the subject, recognize the various types of engines, and study the advantages and disadvantages of each. Because of the rapid development in the design and manufacture of internal-combustion engines, it has been impossible to include some worthy historical and recent types. The applicable theory is presented, as well as typical examples of calculations whenever necessary for clarity; numerous problems (some with answers) are included in various parts of the text.

Separate treatments of the subject matter on gas and gasoline engines are included because of the increasing use of gas engines and convertible gas-Diesel types and the influence of automotive and aircraft experiences on the progress of the high-speed Diesel engine types. The development of the high-speed Diesel engine is described in a later chapter, together with its tendency to depart widely from the established principles of the slow-speed Diesel engine and to follow more closely the gasoline-engine design and operating characteristics.

Distinctive design features of typical engines are included in appropriate parts of the book; recent practices in the design of major engine parts are presented in a later chapter. Engine details, accessories, auxiliaries, and the applications of new materials are also described.

The prime necessity of testing the operation and determining the performance of internal-combustion engines has led to the inclusion of a chapter on the methods of conducting various tests and of computing and presenting the results in accordance with standard test codes, as prepared by the American Society of Mechanical Engineers and the Society of Automotive Engineers. Tabulated test results and characteristic performance curves for representative engines are included. The increasing importance of and public interest in such related problems as waste-heat utilization, sound- and vibration-isolation, the economics of gas and oil power, and the use of internal-combustion engines in combination with steam for power and heating purposes have suggested the necessity of including material on these and other significant trends.

The choice of the terms Otto-cycle engine instead of *spark-ignition* engine, Diesel engine instead of *compression-ignition* engine, two cycle instead of two-stroke cycle or *two stroke*, and four cycle instead of four-stroke cycle or *four stroke* has been made solely for the purpose of adherence to current usage, although the adoption by many leading authorities of the terms italicized will probably lead to general recognition of these names by the press and public in the course of time.

The scientific and engineering symbols and abbreviations used in this book are in general accordance with the American Tentative Standards approved by the American Standards Association.

The author wishes to express his thanks to those manufacturers who kindly furnished drawings, photographs, and data, and to those publishers and authors who permitted their work to be used in the text of this book. He also expresses his thanks to his associates, Professors Alex Vallance and George H. Fancher, who assisted him in the preparation of portions of the book.

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CHAPTER I

DEVELOPMENT AND APPLICATION

The development of the steam engine was in a large measure responsible for the rapid advancement of civilization during the nineteenth century; it improved the means for manufacture, transportation, and communication. Steam engines made possible the erection of factories, the use of machine tools, the operation of steel mills, etc.; in fact, it is to steam that the whole organization of modern industry is due. It provided a degree of mobility on land and sea, whereby the frontiers of civilization were extended, the peoples of the world were brought into closer and more frequent contacts with one another, and the continued expansion of trade was made possible at home and abroad.

1. Progress of Heat Engines. Since the opening of the twentieth century the position of steam as a source of power has been challenged. Gas and oil engines have been developed with a rapidity unequaled in any other branch of engineering, and it seems likely that the influence of the internal-combustion engine upon the history of the present century will be as great as that of the steam engine upon the nineteenth century. Mere replacement of steam engines by gas or oil engines has not caused radical changes in conditions of living, but the improvement and more extensive use of internal-combustion engines has made it possible to serve better a greater number of people. The use of the automobile, the airplane, the tractor, and other portable power machines has produced a considerable influence on our daily life.

The World War was a great stimulus to the development of the internal-combustion engine; the airplane, airship, submarine, and tank were very much in evidence during that time. The measure of effectiveness of these machines was mainly dependent upon the stage of development of the internal-combustion engine. Regardless of the progress in automobile engines, in stationary engines, and in marine engines, it is in connection with air locomotion that internal-combustion engines will probably exert an even greater influence on the civilization of the future by extending the use of the airplane and airship to a more general application to commercial and passenger transport service. The mobile internal-combustion engine is no longer a luxury; it has become one of the necessities of peaceful civilization and is of prime importance in time of war.

The uses of the internal-combustion engine and steam engine overlap only slightly; the former employs a cycle which, in general, is more efficient than the steam cycle, and consequently more economical. The high-powered internal-combustion engine is a complex machine whereas the steam turbine is a simple piece of mechanism and can be built in units up to 100,000 hp or in compound units to more than 200,000 hp. Internal-combustion engines of this size are out of the question at present; hence, for large installations such as are required in electric generating stations, battleships, etc., the steam turbine is used, and it has supplanted the reciprocating steam engine. For comparatively small-powered installations and particularly where compactness of plant is desirable, the internal-combustion engine is much used: for example, airplanes, airships, submarines, automobiles, locomotives, tractors, excavating machinery, etc. In plants having combustible by-product gases, such as those from blast furnaces and coke ovens, large-size units are frequently used. This type of engine has already replaced the steam engine for the smaller classes of shipping and is gradually extending to the larger vessels, but here its progress will be retarded, for the turbine shows to particular advantage as a marine engine, because in this field there is always available that upon which its efficiency largely depends, namely, an unlimited supply of cold water.

2. Principles of Operation. Some of the heat generated by the combustion of fuel in a furnace passes through the metal plates or tubes of the boiler to the water, and the steam generated may be used in an engine cylinder or turbine to do work. The working medium is therefore entirely different from the products of combustion of the fuel. This intermediate step, with its numerous heat losses, is not necessary in an internal-combustion engine, as combustion of the fuel takes place within the cylinder and work is done by expansion of the resulting gases. The working substance, therefore, undergoes a chemical change during this operation, and the thermodynamic process is not cyclic. In the early stages, before combustion, the substance is a mixture of fuel with air; generally an excess of air is required to provide enough oxygen for complete combustion. In the later stages, after combustion, there is a mixture of the products of combustion with nitrogen and any surplus of air. The fuel commonly enters as a gas or vapor and is drawn in with a suitable proportion of air; but the fuel may be injected into the cylinder as a liquid, becoming vaporized after admission or being burned as it enters.

Typical examples of internal-combustion engines are the ordinary gas and gasoline engines, in which a "charge" of air, mixed with combustible gas or vaporized liquid fuel, is drawn into the cylinder by the movement of the piston. The mixture is compressed into a clearance space, and there ignited by an electric spark, so that explosive combustion takes place while the volume of the charge is nearly constant. The heat thus internally developed gives the working substance a high temperature and pressure; expansion of the gases occurs, and work is done as the piston advances.

Considered thermodynamically, internal-combustion engines have the advantage, over the steam engine and turbine, that the working substance "takes in" heat (by its own combustion) at a very high temperature. In the combustion of the charge a temperature of 4000 F[1] or more may be reached momentarily. The full thermodynamic advantages of a high initial temperature cannot be realized in practice, for if the cylinder walls were allowed to reach this high temperature they would soon be destroyed. Lubrication of the piston would also be impossible, to say nothing of the probable premature ignition of the incoming fuel mixture; hence the cylinder is generally water-jacketed to keep the walls and engine parts cool. With large gas and oil engines the difficulty is to keep the cylinder, piston, and valves cool; with steam engines, the cylinder should be kept hot to reduce the losses due to condensation of the steam. The average temperature at which the heat is developed in an internal-combustion engine is far above that at which heat is received by the working medium of a steam engine or turbine. On the other hand, internal-combustion engines do not discharge heat at as low a temperature as do steam engines and turbines. But the actual working range of temperature is so large that a gas engine can convert into work a larger fraction of the heat energy of the fuel than is utilized by any steam engine or turbine. A good internal-combustion engine will convert 35 to 40 per cent of the energy of the fuel into work; the best steam engines convert no more than about 20 per cent; and the best steam turbines convert no more than 30 per cent of the energy of the fuel into work.

3. Development. Long before James Watt built his crude atmospheric steam engine, and before the early beginnings of steam power as we know it today, scientists had already suggested that useful work could be accomplished by exploding gunpowder in a closed vessel with a movable top. As early as 1680 Huyghens, a Dutch physicist, actually built an engine on this principle. Yet experimentation was necessarily postponed, development arrested.

[1] "Flame Temperatures in an Internal-Combustion Engine," A. E. Hershey and R. F. Paton, *University of Illinois Engineering Experiment Station Bulletin* 262, 1933.

Previous to the year 1860, the gas engine was in the experimental stage. Many attempts were made to improve it, but none of the inventors sufficiently overcame the practical difficulties to make their engines a commercial success.

Lenoir, a Frenchman, was the inventor of the first practical, marketable gas engine. This engine was built in Paris in 1860 and resembled a double-acting steam engine. It was a horizontal, double-acting, non-compressing, reciprocating unit with crank and flywheel. The engine used benzene as fuel and approximated *two-stroke* cycle operation; ignition was obtained by an electric jump spark. No shock whatever was heard from the explosion, the running of the engine being as smooth and silent as that of a steam engine. The engine, however, was very uneconomical in the use of fuel and had a thermal efficiency of only 4 per cent; the heat transmitted to the engine parts was so great that the piston had to be flooded with oil. The uneconomical operation of the engine was due to the lack of compression, to ineffective combustion, and to loss of heat through the cylinder walls.

In 1862, Beau de Rochas, a French designer, took out a patent setting forth the best theoretical working conditions for an internal-combustion engine, with a view to utilizing more completely the heat supplied. His proposed cycle of operation was in all respects the same as that in use at the present time in the so-called Otto-cycle engines. The following four conditions were embodied in his patent:

1. Largest cylinder volume, with smallest exposed surface.
2. Maximum possible piston speed.
3. Highest possible pressure at the beginning of the expansion or working stroke.
4. Greatest possible expansion.

To obtain the results which he had laid down as being necessary for high efficiency, de Rochas proposed to use a single cylinder and to carry out the cycle in *four strokes* of the piston as follows:

1. Admission of charge of air and gas.
2. Compression of air and gas.
3. Ignition at the dead (center) point with subsequent explosion and expansion.
4. Discharge of the products of combustion from the cylinder.

At the Paris Exposition of 1878, Otto and Langen, two Germans, exhibited the celebrated Otto silent engine, which revolutionized the gas engine industry. In this engine the whole cycle of de Rochas was for the first time successfully carried out in one cylinder. The cycle was divided into four piston strokes, two forward and two backward, covering two revolutions, thus obtaining one explosion and a working stroke for each two revolutions in a single-cylinder, single-acting engine. The *Otto cycle*, named after the maker of these first engines, is now used in practically all automobile, truck, tractor, airplane, and small marine engines. However, other types of engines have also gained in prominence.

In 1879, a modification of the Otto engine was built by Dugald Clerk, an English engineer. In the Clerk engine the charge was compressed and exploded once every revolution, as compared with one explosion every two revolutions in the engines of the Otto type. The Clerk engine is known as the *two-stroke* cycle type.

Before Otto took out his patent for the engine which influenced practically all gas engines built afterwards, an American, George B. Brayton, took out patents in 1872 on a gas engine and an oil engine. In both of these engines combustion took place at nearly constant pressure. The gas engine was never successful, so oil was resorted to as a fuel. This engine was mechanically better than any previous design, but its efficiency (about 6 per cent) was insufficient to enable it to compete with other types. The Brayton engine employed low compression pressures, and a pilot light was required to ignite the fuel; in this respect it was still far from the *Diesel* principle.

In 1892, Rudolf Diesel, a German scientist and inventor, proposed a hypothetical internal-combustion engine, which would burn coal dust instead of oil. Air in the cylinder was to be compressed to such a pressure that the resultant temperature would ignite the coal; fuel was to be injected at constant pressure during 10 per cent of the ensuing stroke. The first successful Diesel engine was completed in 1897. The present-day Diesel engine is an internal-combustion engine that compresses air before the fuel is injected and the heat of compression ignites the fuel as it enters the cylinder, thus replacing the carburetor and ignition system of the Otto-cycle engine with a Diesel high-pressure fuel injection pump. Liquid fuels, ranging from kerosene to coal tar, are generally employed, although vegetable oils and animal fats may be utilized; Diesel proposed to use pulverized solid fuels, and several such engines [2] have proved commercially successful and economical. The essential feature of the Diesel engine consists in the compression of atmospheric air to a temperature of about 1000 F which is sufficient to ignite and burn the fuel, the fuel being injected at a predetermined rate during the first part of the expansion or working stroke.

The principal differences between the Otto-cycle and Diesel-cycle types of internal-combustion engines are:

[2] For references, see footnote on page 17.

1. Fuel is injected into air *after* compression in the Diesel- and mixed with air *before* compression in the Otto-cycle engine.

2. Much higher compression pressures are used in the Diesel- than in the Otto-cycle engine.

3. The Diesel engine requires no external source of ignition, whereas the Otto-cycle engine requires an outside source of ignition to produce combustion.

4. The Diesel engine approaches constant-pressure combustion of the fuel, while the Otto cycle employs approximately constant-volume combustion (explosion) of the fuel-air mixture.

4. Application. There is no universal internal-combustion engine adaptable for gas, gasoline, kerosene, or fuel oil; and equally adaptable for ships, automobiles, aircraft, locomotives, or industrial purposes. Each type of engine must be built to meet a specific duty, and hence there are motor-ship engines, automobile engines, tractor engines, airplane engines, locomotive engines, stationary engines, low-speed engines, and high-speed engines.

The injection oil engine of the Diesel type, compared with the carburetor (Otto-cycle) engine, introduces some interesting factors. The Diesel cycle presupposes that air alone will be under compression, and that somehow, whenever it pleases the designer, the fuel will be injected. Thus, by properly choosing the time of fuel injection, the compression may be as high as the designer desires. In the Diesel cycle a high temperature of ignition is possible because during compression no fuel is present to ignite (or pre-ignite). As a consequence, the Diesel engine has higher possibilities of efficiency; those possibilities are attainable, and generally they are obtained.

If the compressions that correspond to really high efficiencies were used, this would necessarily result in very high pressures; the resultant additional stresses would require an increase in the weight of the engine. A very high compression may be used for efficiency in one engine; in another, some of the possible efficiency may be sacrificed by lowering the compression in order to obtain a lighter and smoother operating engine. The ideal internal-combustion engine, stationary or portable, of the future, is an engine which will develop the most power from a given weight of materials and with a minimum of fuel consumption.

The selection of any particular type of internal-combustion engine is frequently based on reliability and cost. Although, to a certain extent, design likes and dislikes do influence the choice of equipment, the final analysis must take into consideration all the factors that influence the cost of delivered power. With this idea in mind, it is safe to state that every type of fuel-using installation in use today has and will for some time to come continue to have its field of application. There is little likelihood that any one type of power-generating equipment will eliminate all others.

A feature of power-plant practice that is receiving increased attention is the use of internal-combustion engines in conjunction with steam engines or turbines to balance the heat and power output. In many industrial and commercial establishments the demand for power is increasing more rapidly than that for heat. This ratio is changing because the general trend of industrial progress is toward the more efficient use of steam in heating and process work, thus reducing the quantity of steam per unit of product; at the same time the growing use of machinery is steadily increasing the power required per unit of product. Similarly, in large office buildings, department stores, and hotels, heat demands have changed very little, whereas power requirements have increased rapidly because of the introduction of high-speed elevators, escalators, air-conditioning systems, increased illumination, and other conveniences for improved human comfort and efficiency. Internal-combustion engines are being used to supplement the power generated by existing steam equipment for the above applications; this has resulted in lower over-all costs for both heat and power.

The fuel problem deserves serious thought and constructive work on the part of individuals, corporations, and governments. The solution demands readjustment and development in the petroleum and related industries, and in the internal-combustion-engine field. A partial solution of the fuel problem lies in the further development and application of Diesel engines which will more successfully utilize the heavier oils as fuel. Undoubtedly, the Diesel engine will find increased use on boats, locomotives, excavators, tractors, trucks, airplanes, and no doubt pleasure cars. One of the chief obstacles to the use of heavy-oil engines has been their weight, necessitated by the excessive pressures within the cylinders.

In recent years there has been a definite trend toward greater commercialization of the smaller Diesel engines. Improved engine designs, materials, and manufacturing methods have made it possible to produce high-speed Diesel engines of smaller weight, space, and cost per horsepower with better economy, performance, and reliability. These developments, in turn, have resulted in smaller foundations, lighter reciprocating parts, less vibration, and easier handling of parts. Some fields of the light-duty gasoline engines have not yet been reached commercially by Diesels, but the Diesels are continuously moving toward lighter-duty applications.

5. Classification. Internal-combustion engines may be classified according to: *cycles of operation,* two-stroke cycle and fcur-stroke cycle;

number of cylinders, one, two, three, four, etc., or single-cylinder and multi-cylinder; *cycles of combustion,* Otto cycle, combustion at constant volume, and Diesel cycle, combustion at constant pressure; *arrangement of cylinders,* vertical or horizontal, single-acting or double-acting; *uses,* stationary, portable, marine, automobile, tractor, airplane, etc.; *fuels employed* and how the fuel is supplied to the engine cylinder, as gas, oil, gasoline, kerosene, etc., and carburetor, hot-bulb, solid-injection, air-injection, etc.

No single system of classification serves for all engines; on the other hand it is not necessary to state all the above characteristics to define an engine. If an engine is said to be an automobile engine, it is partly classified, and would be completely so if information regarding the arrangement and number of cylinders were given. A mental picture is immediately formed of a four-stroke cycle Otto engine using gasoline as fuel. However, a complete description would be as follows: Otto cycle, six-cylinder, single-acting, vertical, four-stroke cycle, gasoline, carburetor, 100 bhp at 3400 rpm. A certain type of engine is called a marine engine, but the engine can be more completely identified thus: Diesel, four-cylinder, double-acting, vertical, two-stroke cycle, heavy fuel-oil, solid-injection, 300 bhp at 350 rpm.

CHAPTER II

FUELS AND FUEL-AIR MIXTURES

The fuels used in gas and oil engines are extremely different in origin, in composition, and in heat value. They consist almost entirely of the chemical elements carbon, hydrogen, and oxygen; and their compounds. These constituents in chemical combination with oxygen furnish the heat (and resultant power) developed in an engine cylinder.

6. Classification of Fuels. The principal products of nature convertible into use as fuel for internal-combustion engines are coal, natural gas, and petroleum oil. Some of the artificial or manufactured fuels available are pulverized coal, illuminating gas, blast-furnace gas, coke-oven gas, producer gas, alcohol, petroleum distillates, residual liquid fuels, and blends of distillate and residual fuels.

Internal-combustion engines may be operated on a gaseous fuel, as well as any other fuel which can be transformed into a dust, vapor, or gas. By vaporized fuel is meant any fuel, such as gasoline, kerosene, oil, or alcohol, which may be used in the engine cylinder without the intermediate step of conversion into a gas. All combustible substances may be transformed into a gas or have gaseous products driven off by the action of heat. Any one of these fuels may be used efficiently in the cylinder of an internal-combustion engine. The power obtained from any fuel first converted into a gas and then "burned" in the cylinder of an internal-combustion engine is invariably greater than if the same amount of fuel were burned under an intermediate member, such as a steam boiler, and the heat imparted to the steam used to operate a steam engine or turbine.

7. Combustion Chemistry. During the combustion of a fuel the elements carbon and hydrogen unite with the oxygen of the air to form molecular compounds. The product formed by the complete combination of carbon and oxygen is carbon dioxide, CO_2; and that formed by the union of hydrogen and oxygen is water, H_2O. The products of combustion are formed with mathematical precision and according to definite chemical reactions which depend on the combining power of each element.

The equations indicate for the respective elements the combining proportions for both weight and volume. As an illustration of the method for writing such an equation, consider the combustion of methane, CH_4. Since both C and H are present the products will be CO_2 and H_2O. The chemical reaction (equation) is therefore

$$1CH_4 + 2O_2 = 1CO_2 + 2H_2O \tag{1}$$

The weight relation is obtained by substituting in the equation the molecular weights (see Table I) of the respective elements; the volume relation is indicated by the respective coefficients, which designate the number of units of volume. An analysis of the above reaction shows that

$$\begin{array}{llllllll}
16\text{ lb} & CH_4 + & 64\text{ lb} & O_2 = & 44\text{ lb} & CO_2 + & 36\text{ lb} & H_2O \\
1\text{ lb} & CH_4 + & 4\text{ lb} & O_2 = & 2.75\text{ lb} & CO_2 + & 2.25\text{ lb} & H_2O \\
1\text{ vol} & CH_4 + & 2\text{ vol} & O_2 = & 1\text{ vol} & CO_2 + & 2\text{ vol} & H_2O \\
1\text{ cu ft} & CH_4 + & 2\text{ cu ft} & O_2 = & 1\text{ cu ft} & CO_2 + & 2\text{ cu ft} & H_2O \\
1\text{ mol} & CH_4 + & 2\text{ mols} & O_2 = & 1\text{ mol} & CO_2 + & 2\text{ mols} & H_2O
\end{array}$$

The *mol* or pound-mol is a convenient unit of measurement, and is the weight in pounds equal to the molecular weight of the element or compound. *Avogadro's law* deduced experimentally shows that, when at the same temperature and pressure, equal volumes of all gases contain the same number of molecules. Thus,

$$\begin{aligned}
1\text{ mol } H_2 &= 2\text{ lb } H_2 \\
1\text{ mol } O_2 &= 32\text{ lb } O_2 \\
1\text{ mol } CH_4 &= 16\text{ lb } CH_4\text{, etc.}
\end{aligned}$$

One mol of any gas occupies a volume of 358.7 cu ft under standard conditions of 32 F and 29.92 in. mercury, absolute pressure. This value is obtained by applying the equation of a perfect gas to a mol of any gas,

$$\text{for } O_2, \qquad V = \frac{MRT}{P} = \frac{32 \times 48.24 \times 491.6}{14.7 \times 144} = 358.7\text{ cu ft}$$

$$\text{for } CH_4, \qquad V = \frac{MRT}{P} = \frac{16 \times 96.32 \times 491.6}{14.7 \times 144} = 358.7\text{ cu ft}$$

Considering one mol of any gas, at a temperature of 32 F and an absolute pressure of 29.92 in. mercury, the equation of a perfect gas gives

$$MR = \frac{PV}{T} = \frac{14.7 \times 144 \times 358.7}{491.6} = 1544 \tag{2}$$

or

$$R' = MR = 1544$$

TABLE I

COMBUSTION DATA

Absolute pressure, 14.7 lb per sq in. Temperature, 60 F.

Substance	Chemical Symbol	Molecular Weight	Mean Specific Heat,* 32 to 400 F		Weight of Gas in lb per cu ft	Theoretical Air Required for Complete Combustion		Heating Value of Fuel, Btu per lb		High Heating Value, Btu per cu ft	
			c_p	c_v		Lb per lb of Fuel	Cu ft per cu ft Fuel	High	Low	Of Gas	Of Gas-air Mixture
Acetylene	C_2H_2	26	0.447	0.348	0.0686	13.27	11.93	21,600	20,820	1482	114.6
Air	....	29	0.242	0.173	0.0763	....	....	....	....	....	....
Alcohol vapor	C_2H_6O	46	0.324	0.287	0.1213	9.03	14.35	12,800	11,500	1550	101.0
Carbon	C	12	....	....	....	11.52	....	14,600	14,600	....	....
Carbon dioxide	CO_2	44	0.216	0.171	0.1161	....	....	....	....	....	....
Carbon monoxide	CO	28	0.250	0.179	0.0739	2.48	2.40	4,370	4,370	323	95.0
Ethane	C_2H_6	30	0.429	0.363	0.0793	16.08	16.71	22,230	20,200	1763	99.5
Ethylene	C_2H_4	28	0.401	0.331	0.0740	14.77	14.32	21,600	20,160	1598	104.3
Gasoline vapor	C_8H_{18}	114	0.560	0.543	0.3005	15.13	59.75	20,300	18,700	6100	100.0
Hydrogen	H_2	2	3.224	2.232	0.0053	34.56	2.40	62,000	51,900	329	96.8
Kerosene vapor	$C_{12}H_{26}$	170	0.560	0.549	0.4493	15.07	88.68	19,600	18,000	8806	98.2
Methane	CH_4	16	0.660	0.536	0.0423	17.32	9.55	23,850	21,320	1010	95.7
Nitrogen	N_2	28	0.249	0.179	0.0741	....	....	....	....	....	....
Oxygen	O_2	32	0.218	0.156	0.0844	....	....	....	....	....	....

* Calculated from equations given in *Bulletin* 139, University of Illinois Engineering Experiment Station.

in which R' = the *universal gas constant.*
R = constant in the equation of a perfect gas.
M = pounds of gas numerically equal to the molecular weight.

Equation 2 is particularly useful in determining the gas constant R for a " compound " gas.

A compound gas is a mixture of perfect gases, each exerting a pressure on the sides of the container equal to that which the gas would exert if it were alone in the container at the same temperature. By *Dalton's law,* the total pressure exerted is equal to the sum of the pressures due to each of the constituent gases. Let the weight of a mixture be represented by M, the weights of the constituents by M_1, M_2, M_3, etc.; and the pressure of the mixture by P, the pressures of the constituents P_1, P_2, P_3, etc.; then $P_1 + P_2 + P_3 +$ etc. $= P$. If R_1, R_2, R_3, etc., represent the gas constants of the constituent gases, and R the gas constant of the mixture, the equation of a perfect gas can be set up for each constituent, assuming that each occupies the volume of the mixture at the same temperature.

$$P_1V = M_1R_1T$$
$$P_2V = M_2R_2T$$
$$P_3V = M_3R_3T, \text{ etc.}$$

Adding these equations

$$PV = (M_1R_1 + M_2R_2 + M_3R_3 + \text{etc.})T$$

For the mixture

$$PV = MRT$$

Hence,

$$MR = M_1R_1 + M_2R_2 + M_3R_3 + \text{etc.}$$

$$R = \frac{M_1}{M}R_1 + \frac{M_2}{M}R_2 + \frac{M_3}{M}R_3 + \text{etc.} \tag{3}$$

The oxygen required for combustion is generally taken from the atmosphere. Atmospheric air is a mixture of 23 per cent of oxygen and 77 per cent of nitrogen by weight, and about 21 per cent of oxygen and 79 per cent of nitrogen by volume; hence, one mol of O_2 carries with it $\frac{79}{21} = 3.78$ mols N_2. If air were used instead of oxygen for the combustion of CH_4, the equation for the process becomes:

$$1 \text{ mol } CH_4 + 2 \text{ mols } O_2 + 7.56 \text{ mols } N_2$$
$$= 1 \text{ mol } CO_2 + 2 \text{ mols } H_2O + 7.56 \text{ mols } N_2$$

In practice an excess of air over the amount required for the combustion is frequently used. This excess is usually 20 to 30 per cent, though in some engines an excess of 50 per cent may be necessary to insure complete combustion.

Compound gaseous fuels (see Table II) are mixtures of various combustible constituents along with some non-combustible components. The gas analysis, as determined, is on a volumetric basis, but it may be desirable to use the analysis by weight. This change may be effected by considering unit volume of the compound gas to be one mol, then the volumetric analysis gives the part of a mol occupied by each constituent. For each gas, the weight is determined by the product of the decimal part of a mol and the weight per mol of the respective gas. This method is used for each constituent, and the total of these results gives the weight per mol of the compound gas; from this total the analysis of the gas by weight is obtained. The following example illustrates the procedure.

Example. Determine (by volume and weight) the air supplied and the products of combustion formed when the blast-furnace gas, whose composition is given in Table II, is burned with 25 per cent excess air.

Solution. (a) *Volume of air supplied per cubic foot of gas.*

Volume Composition in Mols	Reaction Equations
$H_2 = 0.052$	1 vol H_2 + $\frac{1}{2}$ vol O_2 = 1 vol H_2O
$CO = 0.268$	1 vol CO + $\frac{1}{2}$ vol O_2 = 1 vol CO_2
$CH_4 = 0.016$	1 vol CH_4 + 2 vol O_2 = 1 vol CO_2 + 2 vol H_2O
$CO_2 = 0.082$	2 lb H_2 + 16 lb O_2 = 18 lb H_2O
$O_2 = 0.002$	28 lb CO + 16 lb O_2 = 44 lb CO_2
$N_2 = 0.580$	16 lb CH_4 + 64 lb O_2 = 44 lb CO_2 + 36 lb H_2O

Multiplying the coefficients of O_2, CO_2 and H_2O in the above reaction equations by the weights of the respective constituents in mols, the following is obtained:

Constituent	Required O_2	CO_2 Formed	H_2O Formed
H_2	$0.052 \times \frac{1}{2} = 0.026$	....	$0.052 \times 1 = 0.052$
CO	$0.268 \times \frac{1}{2} = 0.134$	$0.268 \times 1 = 0.268$	....
CH_4	$0.016 \times 2 = 0.032$	$0.016 \times 1 = 0.016$	$0.016 \times 2 = 0.032$
	0.192	0.284	0.084
Minus O_2 in fuel...........	0.002		
Net O_2 required............	0.190		
25 per cent excess..........	0.047		
Net O_2 supplied............	0.237		

Nitrogen introduced with air, $0.237 \times \frac{79}{21} = 0.893$ mol. Air supplied = 0.237 + 0.893 = 1.13 mols. Hence, 1.13 cu ft of air is supplied per cubic foot of the blast-furnace gas.

(b) *Volume of products of combustion per cubic foot of gas.*

Original Mixture	Mols	Products of Combustion	Mols
H_2	0.052	H_2O	0.084
CO	0.268	CO_2 .... 0.284 + 0.082 =	0.366
CH_4	0.016	Excess O_2	0.047
CO_2	0.082	N_2 .... 0.580 + 0.893 =	1.473
O_2 .... 0.002 + 0.237 =	0.239		1.970
N_2 .... 0.580 + 0.893 =	1.473		
	2.130		

The contraction of volume (original to final) is 2.130 − 1.970 = 0.160 mol, or about 7.5 per cent. If the pressure of the products' mixture were 14.7 lb per sq in., the partial pressure of the H_2O would be $\frac{0.084}{1.970} \times 14.7 = 0.627$ lb per sq in. abs. The saturation temperature of steam at this pressure is 86.5 F; hence, no condensation of water vapor could occur until the temperature of the mixture was less than this value.

(c) *Weight of air supplied per pound of gas.*

Gas	Mols		Molecular Weight		Pounds per Mol	Pounds per Pound
H_2	0.052	×	2	=	0.104	0.0037
CO	0.268	×	28	=	7.504	0.2702
CH_4	0.016	×	16	=	0.256	0.0092
CO_2	0.082	×	44	=	3.608	0.1299
O_2	0.002	×	32	=	0.064	0.0023
N_2	0.580	×	28	=	16.240	0.5847
					27.776	

The weight of air required to burn 1 lb of the gas is:

H_2	0.0037 × 8 ÷ 0.23 =	0.129 lb of air
CO	0.2702 × 0.572 ÷ 0.23 =	0.672
CH_4	0.0092 × 4 ÷ 0.23 =	0.160
		0.961
Air equivalent of O_2 in gas, $\frac{0.0023}{0.23}$	=	0.01
Net air required		0.951
25 per cent excess		0.238
Air supplied per pound of gas		1.189 lb

(d) *Weight of products of combustion per pound of gas.* Refer to chemical reactions in (a) for combining weights.

Gas	Pound per Pound	H_2O	CO_2	N_2
H_2	0.0037	0.0333	......	0.0991
CO	0.2702	......	0.4246	0.5169
CH_4	0.0092	0.0207	0.0253	0.1232
CO_2	0.1299	......	0.1299	......
O_2	0.0023	......	......	−0.0077
N_2	0.5847	......	......	0.5847
		0.0540	0.5798	1.3162

Products of combustion	= 1.950 lb per lb gas
Excess air supplied (from part *c*)	= 0.238
Total weight of products of combustion	= 2.188 lb per lb gas

8. Heat of Combustion. The heat liberated by the complete and rapid oxidation (or burning) of a unit weight or volume of a fuel is called the *heating* or *calorific value* of the fuel. In this country, the heating value is generally expressed in Btu per pound for solid or liquid fuels, and Btu per cubic foot (at 60 F and 14.7 lb pressure) for gases.

Fuels which contain hydrogen or carbon-hydrogen compounds have two heating values, the *higher* and the *lower*. The hydrogen in these fuels burns to form water which is vaporized and superheated in the cylinder of an internal-combustion engine at (or above) the exhaust-gas temperature. The heat thus required is lost, and only the lower heating value of the fuel is available for power. The higher heating value disregards this loss of heat, above the initial temperature of the fuel, and is the value generally used in engineering practice.

The heating value of a fuel may be determined by two methods: by calculation, and by means of a fuel calorimeter. When the ultimate analysis of a solid fuel is known the heating value may be computed by *Dulong's formula,*

$$H_f = 14{,}600C + 62{,}000\left(H - \frac{O}{8}\right) \qquad (4)^1$$

in which H_f = Btu per pound of solid fuel; and the symbols C, H, and O represent the weights of carbon, hydrogen, and oxygen in 1 lb of the fuel.

For a fuel containing hydrocarbons, as CH_4, the following approximate formula gives results which check the calorimeter tests closely:

$$H'_f = 14{,}600C + 52{,}000\left(H - \frac{O}{8}\right) \qquad (5)^1$$

in which H'_f = Btu per pound of liquid or gaseous fuel. The difference between equations 4 and 5 is due to the fact that the heat to break up the compound of carbon and hydrogen is not available; hence the actual heat evolved is the theoretical heating value less the heat of this chemical union.

The data in Table I will facilitate the heating-value calculation of a compound fuel.

[1] Sulfur, being undesirable, is not usually considered when computing the heating value of fuels for internal-combustion engines.

Example. (*a*) Check the heating value and theoretical air required per cubic foot of the blast-furnace gas given in Table II.

(*b*) If 25 per cent excess air were mixed with this gas at 120 F and 13.5 lb abs, find the heating value per cubic foot of the air-gas mixture.

Solution. (*a*) *Heating value, Btu per cubic foot of gas.* From Table I,

$$\begin{aligned} H_2 &= 0.052 \times 329 = 17.11 \text{ Btu} \\ CO &= 0.268 \times 323 = 86.56 \\ CH_4 &= 0.016 \times 1010 = 16.16 \\ & \qquad\qquad 119.83 \text{ Btu} \end{aligned}$$

and the air required,

$$\begin{aligned} H_2 &= 0.052 \times 2.40 = 0.1248 \text{ cu ft} \\ CO &= 0.268 \times 2.40 = 0.6432 \\ CH_4 &= 0.016 \times 9.55 = 0.1528 \\ & \qquad\qquad 0.9208 \text{ cu ft of air required} \end{aligned}$$

(*b*) *Heating value of air-gas mixture.*

1 cu ft gas + 0.92 cu ft air = 1.92 cu ft
25 per cent excess air...... = 0.23 cu ft

Volume of air-gas mixture 2.15 cu ft at 60 F and 14.7 lb pressure.

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2} \quad \text{or} \quad V_2 = \frac{P_1V_1T_2}{P_2T_1}$$

$$V_2 = \frac{14.7 \times 144 \times 2.15 \times 580}{13.5 \times 144 \times 520} = 2.611 \text{ cu ft of air-gas mixture at 120 F}$$

and 13.5 lb pressure.

$$\text{Heating value of charge} = \frac{119.83}{2.611} = 45.9 \text{ Btu per cu ft.}$$

The most satisfactory method of determining the heating value is to burn a quantity of the fuel in a calorimeter [2] and to measure the temperature rise induced in a given quantity of water. When the proper corrections are applied for the calorimeter parts, radiation, etc., the heat value per unit of the fuel may be determined from the calorimeter data.

9. Pulverized Coal. Although pulverized coal as a fuel is somewhat new in its present-day application, it has been used under steam boilers from time to time for the past 30 years. The present success of this fuel is due to the tireless efforts of engineers in the application of scientific principles to meet the demands of industry for cheap power.

Coal (coke, lignite, etc.), in order to be utilized with the flexibility and efficiency of gaseous or liquid fuels, must be broken up by distillation or by mechanical atomization. The latter is realized with pul-

[2] For additional information on solid and liquid-fuel calorimeters, see "Steam, Air, and Gas Power," Severns and Degler, pages 95–99, Third Edition, 1939, John Wiley & Sons, Inc.

verized coal[3] which is ground very fine and then suspended in an air stream, so that it possesses the qualities of a gas or finely atomized oil. In order to obtain rapid and efficient combustion, the fuel must present the maximum surface to the combining air (oxygen). A piece of coal represented by a 1-in. cube has 6 sq in. of exposed area. If this same piece of coal were pulverized, so that 80 per cent of it would pass through a 200-mesh sieve, it would have many times as much exposed surface. The coal would be so fine that it could easily be supported in air as a dust and could be handled much the same as a gas.

10. Gaseous Fuels. Table II was prepared to give the reader some idea of the properties of typical gaseous fuels. The last column, giving heating value per cubic foot of charge (gas and air), was calculated by assuming 25 per cent excess air, the suction pressure 13.5 lb abs, and the temperature 120 F. For additional information and properties of gaseous fuels, the reader is referred to the "Gas Engineers' Handbook," published by the McGraw-Hill Book Company.

Natural Gas. This fuel is taken directly from gas wells, and usually issues with the oil from oil wells; it is found in a number of localities in the United States and Canada, also to some extent in several European countries. Natural gas is the most convenient of any of the gaseous fuels for use in internal-combustion engines, as it requires no cleaning, usually contains no sulfur or other acid-forming constituents, and is very rich in heat-producing elements. The heat value of natural gas ranges from 800 to 1200 Btu per cu ft. The chemical constituents vary widely in different localities, and also even in the gas taken from the same region. When used in large quantities, as for power generation, natural gas may be purchased for 10 to 30 cents per 1000 cu ft. It has been conveyed long distances through welded pipe lines, because of its convenience, excellent properties, and low cost. The supply is being exhausted, however, and in some localities natural gas is no longer relied upon when the demand is large.

Natural gas is composed mainly of methane (CH_4) and ethane (C_2H_6); in some cases varying percentages of hydrogen sulfide, carbon dioxide, nitrogen, and helium may be present. Any gas containing a large amount of methane or hydrogen will withstand only low compression without pre-ignition. Some of the leaner gases, such as blast-furnace and producer gas, which will allow higher compression pressures, are generally more efficient fuels. For a given power, internal-combustion engines designed for natural gas require smaller cylinder dimensions

[3] "A Pawlikowski Coal-dust Engine Tested in Czechoslovakia," *Power*, page 515, March 31, 1931. "The Coal-dust Engine," *Mechanical Engineering*, pages 134–136, February, 1932. "This Engine Runs on Anything That Will Burn," *Power*, page 258, May, 1936.

TABLE II

DATA ON GASEOUS FUELS

Kind of Gas	Constituents of Gas in Per Cent by Volume								Higher Heating Value of Gas, at 60 F, Btu per cu ft	Cu ft of Air at 60 F Required per cu ft of Gas	Higher Heating Value of Actual Charge,* Btu per cu ft
	H_2	CO	CH_4	C_2H_4	C_2H_6	O_2	CO_2	N_2			
Natural gas, California	......	......	59.2	......	13.9	......	26.2	0.70	843	7.98	63
Natural gas, Kansas	......	......	62.2	......	18.4	......	......	18.60	953	9.01	64
Natural gas, Ohio	......	......	83.5	......	12.5	......	0.2	3.80	1064	10.06	65
Natural gas, Texas	......	......	98.0	......	......	......	0.7	1.30	990	9.36	64
Illuminating gas	11.70	13.70	23.6	10.5	......	0.70	7.2	32.60	506	4.62	62
Blast-furnace gas	5.20	26.80	1.6	......	......	0.20	8.2	58.00	120	0.92	46
Coke-oven gas	53.00	6.00	35.0	2.0	......	......	2.0	2.00	582	5.09	65
Producer gas	8.00	24.00	3.0	0.5	......	0.50	5.0	59.00	143	1.14	49

* See text for conditions assumed.

because of the higher heating value (Btu per cubic foot) of the charge.

It has been estimated that 80 per cent of the gas produced in the United States is natural gas. Some of the natural gas produced is wasted because it is a by-product necessary to the production of oil. In general, natural gas is found under such circumstances that either a favorable market does not exist or the cost of gathering and transporting the quantity available is not economical. The expansion of facilities for distributing natural gas from the centers of production continues to be nation-wide. The 1000-mile 24-in. line from the Texas Panhandle to Chicago was completed in 1931, and it is proposed to lay a second carrier of 30-in. size, parallel to the first, to be extended to the New England states. The development of more extensive pipe lines seems assured because of the new gas resources continually being found and legislation being enacted for conserving gas and oil production.

Illuminating or Water Gas. This gas is usually made by passing steam over white-hot coke. The gas made from coke is less liable to variation in composition than that made from coal, with its varying hydrogen content. The gas is clean, and the higher heating value averages 450 to 600 Btu per cu ft. As commonly produced this gas is too expensive (seldom less than 75 cents per 1000 cu ft) for extensive use in a gas engine; it is feasible only for small engines or for engines that operate infrequently.

Blast-furnace Gas. Blast-furnace gas, or "downcomer" gas, is a by-product of combustion in the blast furnace. It may be used to preheat the air blast and for various other purposes in the reduction of iron ore. At some blast-furnace plants, the gas from the furnace serves for fuel in the gas engines that drive the blowers which supply the air necessary to support combustion in the blast furnace. The principal combustible in blast-furnace gas is carbon monoxide (CO), which averages 30 per cent; the heating value ranges from 90 to 120 Btu per cu ft.

The amount of gas liberated from the average blast furnace under normal operating conditions is about 150,000 cu ft per ton of pig iron produced. Of this gas, the following distribution represents average conditions:

	Percentage
Consumed by hot-blast stoves	30 to 32
Consumed by gas-blowing engines	10 to 12
Consumed by auxiliaries, and leakage	8 to 10
Surplus gas, approximately	50

This surplus gas, when used in gas engines for the generation of power to be utilized outside the blast-furnace plant, will deliver continuously at least 25 hp per ton of iron produced, which is more than sufficient to operate a blast furnace and steel plant in conjunction.

The most expensive part of the blast-furnace gas power installation, aside from the engines themselves, is the cleaning system. This system is generally elaborate, cleaning the gas by both dry and wet scrubbing. The moisture of the air is dissociated in the blast furnace, and the hydrogen thus formed raises the heating value of the gas, depending upon the humidity of the air supplied and the temperature in the furnace.

One of the interesting characteristics of the gas-burning (Otto-cycle) engine is that the higher the efficiency desired the higher must be the compression ratio; see Table VII. Because of the low heating value (46 Btu per cu ft) of the gas-air mixture entering the cylinder of a blast-furnace gas engine, the pressure at the end of compression can be high and the engine clearance volume must be relatively small. The compression pressure at rated load of a gas engine using natural gas averages 110 lb abs, illuminating gas 130 lb, and blast-furnace and producer gas 165 lb.

Coke-oven Gas. Coke was formerly prepared in " beehive " ovens, in which all the by-product gases were burned to keep the oven hot and to furnish heat for the coking of the coal. Coke is now made in by-product ovens; as the gas is driven off from the coal, it is led away and cooled, then washed. This gas makes an excellent engine fuel and is used as such in localities where it is available. Coke-oven gas contains about 50 per cent hydrogen (H_2) and 35 per cent methane (CH_4), and has a heating value of 450 to 600 Btu per cu ft.

After the gas leaves the coke ovens, the dust, tar, and ammonia are removed; about half of the gas is taken back to the ovens where it is burned to furnish heat for the coking. The other half is available for power purposes, and this gas contains 5 to 10 per cent of the heat of the coal originally charged in the coke oven. Therefore, the heat contained in the by-product gas per ton of coal would be $2000 \times 13{,}000 \times 0.08 = 2{,}080{,}000$ Btu in 4000 cu ft of gas. If the efficiency of the gas engine were 25 per cent, the power obtained from this gas would be

$$\frac{2{,}080{,}000 \times 0.25}{2545} = 204 \text{ hp-hr}$$

Thus, in the manufacture of pig iron two valuable by-product gases are available for efficient use in the internal-combustion engine: coke-oven gas when coal is burned to coke in coke ovens, and blast-furnace gas when iron ore is reduced and purified with the addition of coke and limestone flux in the blast furnace.

Producer Gas. This gas is formed by the combustion of any grade of coal, coke, or peat with a deficient air supply. Incomplete combustion results, with the formation of a considerable quantity of carbon monoxide. Producer gas has a high heating value—120 to 160 Btu per cu ft. Gas producers are of three general types, suction, pressure, and balanced draft, depending upon whether the operating pressure in the producer is below, above, or at atmospheric pressure.

Both the action of a furnace and that of a retort are partly reproduced in the gas producer. The producer is a vertical brick-lined air-tight cylindrical vessel, provided with a grate and ashpit, a deep firepot, inlets for solid fuel and moist air, and outlets for gas, ash, and clinkers. The fuel is charged or fed from above into the firepot and undergoes distillation of the volatile matter. The blast furnace is in reality a large gas producer, which uses coke as fuel with a high rate of combustion and in which the producer action is combined with a metallurgical reaction, the reduction of iron from its ore being accomplished mainly by the action of the carbon monoxide produced. Producer gas and blast-furnace gas are similar in composition and characteristics; hence the previous discussion of the use of high compression pressures for blast-furnace gas also applies, in general, to producer gas.

Sewage or Sludge Gas. Sewage and sludge can be made, paradoxically, to help dispose of themselves, if the process gas generated during disposal is utilized. Charlotte, N. C.; Coney Island, N. Y.; Ann Arbor, Mich.; Peoria, Ill.; Madison, Wis.; Cedar Rapids, Iowa; and several California cities are using this gas in their sewage-disposal plants. Operation of these plants indicates high reliability, low maintenance, and practical elimination of fuel cost; the size of the engine varies from 40 to 750 hp and is dependent only on the available gas supply. These engines are direct-connected to centrifugal pumps, air compressors, or electric generators. The average fuel consumption is 20 to 25 cu ft of sewage gas per hp-hr, varying somewhat with the heating value. The gas is a product of the sludge digestive tanks and averages 600 to 750 Btu per cu ft; it contains about two-thirds CH_4 and one-third CO_2 with small percentages of other gases, including hydrogen sulfide.

Because of the high percentage of CO_2, sewage gas is a slow-burning fuel and may be used in a conventional high-compression (about 6 to 1) multi-cylinder gas or gasoline engine. This fuel has a narrow firing range, thus mixing valves must be designed to supply a low gas-air ratio of fuel admitted to the cylinders and also must be capable of readily varying the incoming air supply to facilitate starting and permit adjustments which may be necessary owing to variations in gas quality. The hydrogen sulfide contained in the gas affects the potential compression ratio, hastens maintenance, and tends to detonate; therefore, if more than 0.5 per cent is present, the engine compression ratio must be decreased. The resultant formation of sulfuric acid gas and corrosion of valves, cylinders, pistons, and exhaust piping may be reduced by the use of corrosion-resisting alloys.

11. Air-gas Mixtures. An important characteristic of a fuel for an internal-combustion engine is its explosibility with an excess or deficiency of air. It is not possible or desirable to regulate the air supply to an engine so that there shall always be present exactly the amount theoretically (chemically) necessary. Other things being equal, that fuel is best which will permit the largest variation of the ratio of air to fuel without failure to ignite. Illuminating gas (550 Btu per cu ft), which unites with 4 to 7 times its own volume of air, will ignite—at atmospheric pressure—with any amount of air between 3 and 11 times its own volume; that is, an engine using this gas will function under a large variation of the ratio of air to gas.

To get complete combustion, the air supply in any gas-air mixture must always be somewhat in excess of that theoretically necessary. However, if it were much in excess of the theoretical amount, the combustion might be complete, but it would be slower and less efficient. In general, a lean mixture (large air-gas ratio) will be more efficient and more economical than a rich mixture (small air-gas ratio); but the rich mixture will develop a greater amount of power in a given engine.

A study of the physical phenomena involved in the explosions of various gas-air mixtures[4] has been made at the University of Illinois. The curves in Fig. 1 have been plotted from data obtained in this investigation. They show the effect of the ratio of air to gas on the maximum pressure obtained by explosion, and on the time it takes the exploded mixture to reach this maximum pressure, with and without turbulence of the mixture. The effect of turbulence during explosion is to cause an increase of maximum pressure and a decrease in the time of explosion. This effect is greater for lean than for rich mixtures. The maximum pressure is also produced with a slightly greater air-gas ratio than for the case where no turbulence exists. These improved effects of turbulence are undoubtedly due to the more intimate mixing of the gas and air before ignition, thus bringing more molecules into contact, rather than to the projection of the flame into the unburned parts of the gas mixture.

Figure 1 shows that a ratio of 1 part of illuminating gas to about 4 parts of air gives a maximum pressure of 88 lb per sq in. abs (with turbulence); also, that the same mixture gives the minimum time for the

[4] "A Study of Explosions of Gaseous Mixtures," A. P. Kratz and C. Z. Rosecrans, *University of Illinois Engineering Experiment Station Bulletin* 133, 1922.

explosion, about 0.02 sec. With a 10 : 1 mixture, the time required (without turbulence) to reach maximum pressure—which is approximately the time required for complete combustion—is nearly 0.50 sec. A small gas engine may run at 360 rpm or 6 rps; hence only 0.083 sec is available for each stroke, and consequently an explosion requiring half a second would be impracticable and uneconomical.

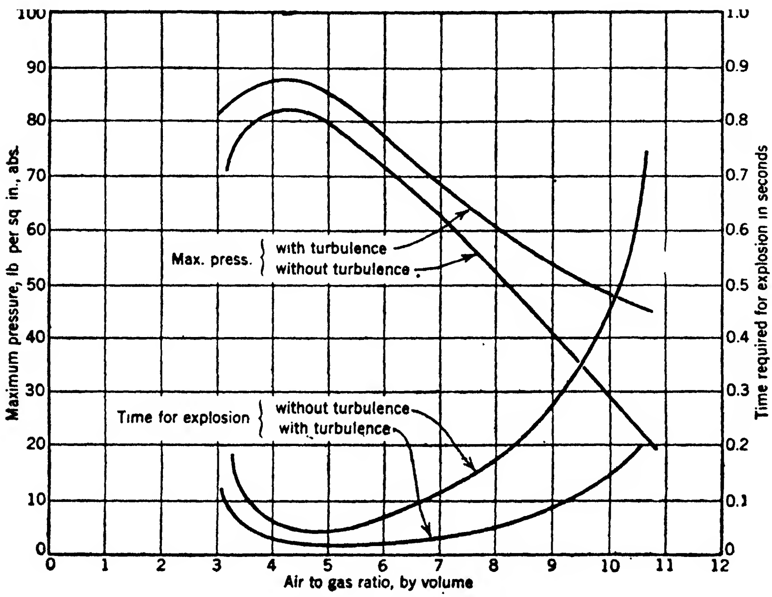


FIG. 1. Explosion curves for air-gas mixtures in L-shaped vessel with ignition at center of head. Charge initially at atmospheric pressure.

The compression of the charge, which takes place in all gas engines, makes the pressure of the explosion and its duration less than those

TABLE III

EXPLOSIVE LIMITS FOR AIR-GAS MIXTURES

Kind of Gas	Cu Ft of Air Required for Combustion per Cu Ft of Gas		
	Theoretical	Low Limit	High Limit
Natural.........	9 to 10	5	14
Illuminating.....	4 to 6	3	11
Blast-furnace....	1	$\frac{1}{2}$	2
Coke-oven.......	5	3	10
Producer........	1	$\frac{1}{2}$	2

shown in Fig. 1. With a compression to 60 lb per sq in. of the best mixture of illuminating gas and air, the explosion in a small engine may be complete in 0.01 sec; for a gasoline-air mixture the time is considerably less. Table III gives the average limiting proportions of air to gas between which the mixture is explosive for the commonly used gaseous fuels.

12. Liquid Fuels. At the present time, by far the most important fuels used in internal-combustion engines are those derived from crude petroleum. Liquid fuels may be divided into five general classes:

1. Crude petroleum or crude oil.
2. Gasoline.
3. Kerosene.
4. Fuel oil.
5. Alcohol.

These fuels consist chiefly of mixtures of compounds of carbon and hydrogen. The oils, of which both crude and refined are used, are composed principally of closely related compounds, some of which on separation are gaseous, some liquid, and others solid at ordinary temperatures. The various liquid constituents may have different densities and volatilities, varying from an extremely light liquid which evaporates rapidly at atmospheric temperature (as ether does) to heavy viscous liquids which must be raised to a high temperature before vapors are evolved. Thus, petroleum and petroleum products cannot be regarded as chemical compounds with definite formulas but as mixtures of definite chemical compounds, namely hydrocarbons.

The petroleums of Pennsylvania, New York, eastern Ohio, and West Virginia are usually referred to as "paraffin-base" oils because of the fact that most of the hydrocarbons present belong to the paraffin series. The Gulf Coast (Texas and Louisiana), Mexico, and California oils usually contain smaller quantities of the paraffin hydrocarbons associated with larger quantities of other hydrocarbons frequently of an asphaltic (hence highly unsaturated and complex) nature. The north Texas, Oklahoma, Kansas, and Wyoming crudes have properties of both the paraffin-base and asphaltic oils, and therefore such crudes are called mixed-base.

Paraffin crude oils, as well as distillates, are less viscous than asphaltic crudes or distillates of equivalent density. The lubricating stocks from paraffin-base crude oils have a much lower rate of change in viscosity with temperature than those prepared from asphaltic crudes. Results of carefully conducted comparative wear tests in the laboratory and on the road made with engines using asphaltic, naphthenic, and paraffinic lubricants show that there was less wear when using the paraffin-base oils than any of the others. However, it is well known that good lubricants can be made from practically any crude petroleum by recently developed methods of refining, but a superior lubricant can be produced from paraffin crudes with much less difficulty and cost than from asphalt-base or mixed-base crudes, primarily because the percentage of desirable constituents is greater in the paraffin-base oils.

Oil is increasingly difficult to find and more expensive to produce, owing to deeper wells and more troublesome strata. The cost of production from oil pools increases with the naturally declining rate of flow because of increasing costs of pumping as well as the expense for artificial stimulation that may be needed, as air drives and water flooding. The ultimate effect will, of course, be a rise in the price of the finished products in least competition, and a withdrawal of oil products from the more competitive markets. It is conceivable that we will have either to look for new sources of liquid fuels or change the power units for the many applications of internal-combustion engines.

Our other sources of liquid fuels can be the fossil solid fuels, i.e., coals of any or all ranks and types, oil shales, or agricultural products. It has been shown many times that agriculture cannot begin to furnish sufficient liquid fuel to meet our present demands. The production of oil from shale involves a high handling charge which makes that source unattractive when compared to coal, which is our third alternative. Our reserves of coal are enormous in comparison with our reserves of oil and gas.

Technologically, liquid fuels of suitable quality and in sufficient quantity can be made from coal. The question is, at present, one of economics, to which, in many European countries, is added a desire for an autonomous economy. In the period 1830 to 1859, oil from coal and shale largely replaced sperm and whale oil as illuminating oil. These oils were, in turn, replaced by petroleum oils after 1859.

Liquid fuels may be made from coal by three basic processes: (1) recovery and refinement of the products of either the high- or low-temperature carbonization processes; (2) destructive hydrogenation and refinement of the resulting products; and (3) synthesis of hydrocarbon or alcohol mixtures from carbon monoxide and hydrogen obtained in the gasification of solid fuel. All three processes are now in commercial or semi-commercial use. Our abundant supplies of petroleum check the economic urge to develop these processes in this country; development has been carried furthest in those countries without a national supply of petroleum but with a highly developed industrialism and a highly nationalistic spirit, such as Germany, England, France, Japan, and Italy.

13. Crude Petroleum. The paraffin-base and mixed-base crude oils can be used in oil engines. The asphaltic crude oils may also be used,

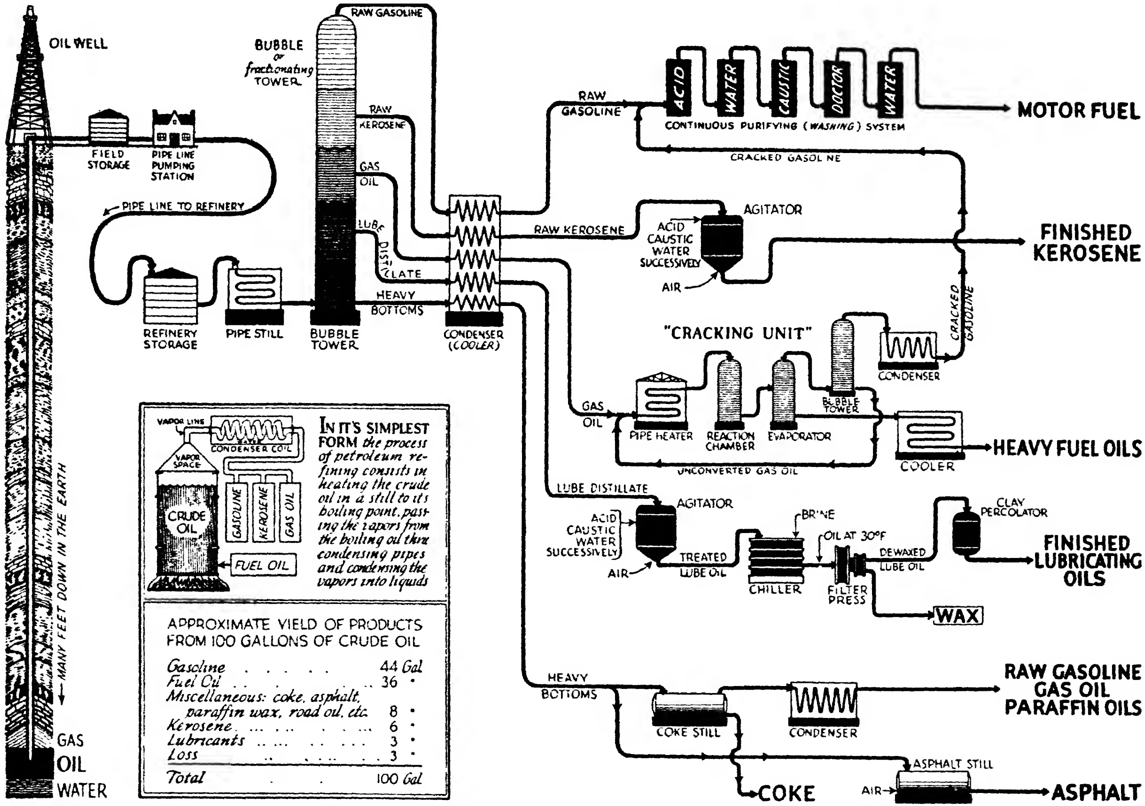


FIG. 2. Typical flow chart tracing crude oil from well to finished product. Courtesy of *American Petroleum Institute.*

but only with difficulty and in engines especially designed for such oils. Crude petroleum, because it is a mixture of chemical compounds of very divergent physical properties, is not a satisfactory fuel; those engine conditions which are favorable for burning one part of the oil are not necessarily favorable for the other constituents.

Crude petroleum is generally refined before using. Figure 2 shows a typical flow chart tracing the crude oil from the well to the various finished products; the approximate yield of products per 100 gal of crude oil is also tabulated. Refining is a process of distillation carried on at present in pipe stills. When crude oil is slowly heated, it gives off as vapor its various constituent elements, the more volatile being given off at the lower temperatures, and the residue becoming continuously more dense and more viscous. In the refining of petroleum, the vapors given off at various temperatures are condensed and collected separately; the names given to the various products are an index chiefly to the temperature at which they are vaporized. The first portions, distilling off below 150 F and called petroleum ethers, are not suitable for transportation because of their high volatility. The gasoline fractions, used for blending the several grades of motor fuel, are obtained next; kerosene is the next fraction having a higher boiling point than gasoline. After kerosene has been driven off, the dense petroleum which remains is called reduced crude, from which various commercial petroleum products may be obtained by further distillation or other refining processes. These products include gas oil, fuel oil, lubricating oil, cylinder oil, paraffin wax, asphalt, petroleum coke, and others.

Few of the operations involved in the refining of oil are carried out today in the same equipment and in the same manner as they were ten or even five years ago. In the petroleum industry, what is new today may be obsolete tomorrow; good equipment is often scrapped to be replaced by more economical methods. Some of the present-day trends of modern petroleum refining are: larger units with continuity of operation, improved and wider application of fractionating towers, better design of pipe stills, vapor-phase cracking, vacuum distillation, varied acid treatment, dewaxing by new methods and at lower temperatures, automatic control of operations, and hydrogenation of residues.

Table IV shows the specific gravities, analyses, and heating values of the most important fuels manufactured from crude petroleum; also the theoretical air required for perfect combustion per pound of the liquid fuels listed.

It is interesting to note that the chemical analyses, air required for combustion, and to some extent the heating values are about the same for these fuels. Hence the above-stated properties are not generally specified in the selection of a liquid fuel for use in an internal-

TABLE IV

TYPICAL ANALYSES OF LIQUID FUELS

Properties and Analyses	Gasoline	Kerosene	Crude Oil	Fuel Oil
Degrees API * at 60 F.	60.5	40.3	28.2	32.2
Specific gravity at 60 F.	0.737	0.824	0.886	0.865
Carbon, per cent by weight.	83.30	83.00	84.24	83.04
Hydrogen, per cent by weight.	14.45	12.40	11.94	11.58
Oxygen, per cent by weight.	1.08	1.60	1.92	2.82
Nitrogen, per cent by weight.	1.09	2.77	1.08	1.40
Sulfur, per cent by weight.	0.08	0.23	0.82	1.16
Higher heating valve, Btu per lb, calorimeter	20,300	19,600	19,500	18,800
Theoretical air, cu ft per lb at 60 F.	191	182	181	178

* American Petroleum Institute; deg API $= \frac{141.5}{\text{Sp gr}} - 131.5$.

combustion engine. Specifications for liquid fuels include some of the following properties: gravity, flash point, viscosity, water content, sediment and residue after burning, and volatility and "anti-knock" properties. Volatility may be defined as that quality of a liquid fuel by reason of which it vaporizes on exposure to air at ordinary temperatures.

There has never been any doubt but that some time the petroleum supply of the United States would approach exhaustion. But long before the extreme condition of exhaustion is reached a shortage of petroleum will be experienced. There is considerable evidence to show that a shortage of petroleum sufficient to necessitate a marked increase in imports and an increase in petroleum prices may be expected within the next ten years. Though a shortage or an inadequate supply of petroleum products may be experienced in this country by that time, exhaustion of the petroleum resources is not likely for another fifty years.

14. Gasoline and Kerosene. Gasoline is the petroleum product in greatest demand. The inability to supply the demand by the usual distillation process, the product of which is termed straight-run gasoline, has led to the development and extensive use of cracking processes which yield a fuel suitable for modern automobile engines. Under conditions of high temperature and pressure, the heavier compounds of carbon and hydrogen break down into lighter and more volatile hydrocarbons with the deposition of free carbon and the liberation of gas. This phenomenon is known as cracking, the refinery process employing it as the cracking process, and the liquid product of proper physical properties obtained from this process as cracked gasoline. A considerable portion of the gasoline sold in this country is cracked gasoline.

The consumption of gasoline during the year 1936 was 20 billion gallons. The supply of gasoline has kept pace with growing demand by the increase in the production of crude oil, the marketing of gasoline with a wider distillation range, the developing of the natural-gas gasoline industry, the using of improved cracking processes, the hydrogenating of coal and tar in foreign countries, importing petroleum from foreign countries, and by the fact that petroleums distilled in recent years have yielded a larger proportion of gasoline. A recent development to provide an additional source of gasoline is the polymerization process, whereby some of the constituents of natural gas are combined by the action of heat, pressure, and catalysts to form the heavier constituents of gasoline. Of the crude petroleum processed in the year 1915 the petroleum industry recovered, by the various refinery methods, 20 per cent as gasoline; in 1937 the gasoline recovery averaged 50 per cent.

The trend of large-scale developments in England and Germany is toward the hydrogenation of coal. In Germany coal hydrogenation yields about 80 gal of gasoline per short ton of coal. During the year 1936 German plants hydrogenated $2\frac{1}{2}$ million tons of coal, producing 200 million gallons of gasoline in this way. In England a coal hydrogenation plant has an annual capacity of 55 million gallons of gasoline when operating on coal and tar, or about 70 million gallons when hydrogenating tar.

The fuel supplied to an Otto-cycle type of internal-combustion engine is generally a gas or atomized fuel mixed with air. Hence special devices are required to convert liquid fuels into gases or vapors. Such devices are indiscriminately known as vaporizers, carburetors, mixers, or mixing valves. The term vaporizer is usually employed when considerable heat for vaporization is required, as with fuel oil and crude oil. A carburetor [5] may be defined as a device in which a liquid fuel is atomized by being sprayed through nozzles or jets into a stream of moving air. When no special heating of the fuel is done, these devices are applicable to the more volatile fuels only, as gasoline. The name carburetor, however, may also be applied when in this device gasoline is mechanically atomized or sprayed into the current of incoming air.

The essential properties that a gasoline for use in an internal-combustion engine should possess are: ability to give easy starting of the engine and rapid acceleration, high volatility over a wide temperature range, low dilution of crankcase oil, non-corrosiveness (from the presence

[5] *Carburetors* are fully described in Article 19.

of sulfur), no tendency to vapor-lock,[6] high anti-knock value, and absence of gum-forming constituents. An ideal fuel would be one which fulfils these requirements completely; but some of the desired properties are incompatible with one another, hence the commercial product must be a compromise.

The grades of gasoline now available to the consumer represent each refiner's attempt to balance the varying requirements of present-day engines and an economical use of the stocks and crude oils available. It can be seen, moreover, that no one specified grade of gasoline can best meet the requirements of all engines. Variations in design necessitate differences in fuels to produce optimum performance with each. The refiner and the engine manufacturer are working more closely together today than ever before to modify fuels and engine designs so that a more generally satisfactory correlation of fuel and engine should result.

Fuel systems should be so designed that the maximum temperature of the gasoline in the system will not rise more than 30 F above the temperature of the atmosphere. With all fuel systems meeting this condition, the conflict between ease of starting and vapor-lock would be minimized. Improvements are also possible to a lesser degree in manifold design and its relation to distribution of the fuel-air mixture to the several cylinders. A parallel case is in a general improvement of the design of the combustion chamber so that higher compression ratios can be employed without necessitating grades of gasoline of abnormally high anti-knock value. At present there are on the market engines of high compression ratios which do not require fuel of as high an anti-knock value as do other engines of lower compression ratios, indicating the importance of combustion-chamber design.

Kerosene is also distilled from petroleum as has been described previously. The specific gravity ranges from 0.80 to 0.84; other general properties of this fuel are listed in Table IV. Since kerosene has a higher flash point and is less readily vaporized than gasoline, it is a safer fuel to use and to transport; but kerosene does not make so convenient a fuel for internal-combustion engines as gasoline. Some of the disadvantages of kerosene are: a cold engine will not start readily with this fuel; kerosene is not easily vaporized at normal temperatures; and it has a tendency to "knock." Some engines are provided with two fuel tanks, the small tank with gasoline for easy starting and a

[6] *Vapor-lock* is that condition of an engine in which the temperature of the feed lines exceeds the boiling point of the gasoline to such an extent that the fuel lines become filled with vapor, preventing the engine from getting gasoline freely. This may result when the gasoline is too volatile.

larger tank of kerosene for running conditions. With compression pressures exceeding 70 lb per sq in. gage, kerosene alone may detonate (see Article 25) considerably, but when it is blended with ethyl alcohol or benzol the difficulty can be eliminated. The detonation knock may also be suppressed by adding small quantities of tetraethyl lead to the kerosene.

15. Fuel Oil. Diesel-engine builders and operators desire a supply of fuel oil for their engines which is ideally suited for best performance. It is natural that, in the development of the Diesel engine, the ideas of different manufacturers regarding what constitutes a suitable fuel for a particular engine should vary widely. At present some manufacturers are endeavoring to build engines which will operate on heavy oils; these engines will not operate satisfactorily on the lighter grades of oil, such as are considered more desirable for other engines. Each manufacturer seems to have a different idea of the grade of fuel oil required for his type of engine. Hence, ever-increasing demands are being made upon the petroleum industry to supply various grades of fuel oil for the many types of Diesel engines now in use.

In recent years a number of engineering societies (American Society of Mechanical Engineers, American Society of Testing Materials, and Society of Automotive Engineers) have undertaken to prepare specifications for standard Diesel fuel oils for the mutual advantage of engine designers and manufacturers, operators, and the petroleum industry. These standardized fuel oils should be well defined as to detrimental factors, and still impose no hardships on the refiner or necessitate an increased price for the fuel as the result of too strict specifications. The engine builder would then know within what range of fuels his engine would be called upon to operate, and he could govern the design accordingly; however, the engine owner should be spared the trouble of buying fuel to too rigid specifications.

The tentative classification of Diesel fuel oils shown in Table V, covering five grades of Diesel fuel, has been issued by the American Society of Testing Materials; it was prepared by Technical Committee C on Fuel Oils of Committee D-2 on Petroleum Products and Lubricants. The intention of the committee is that this classification should be used not as purchase specifications, but rather as information in order that fuels purchased may be correlated according to the proposed classification and thus determine its usefulness. Resultant criticisms and comments should aid in developing a permanent series of Diesel fuel-oil specifications. The diversity of opinions regarding specification limits and the relative importance of the various items indicate the need for continued cooperative work between engine builders, engine users and oil refiners.

TABLE V

TENTATIVE DIESEL FUEL-OIL CLASSIFICATION [7]

Grade of Diesel Fuel	Flash Point, F	Water and Sediment, per cent by volume	Viscosity, Seconds			Carbon Residue, per cent by weight	Ash, per cent by weight	Pour Point, F	Sulfur, per cent by weight	Ignition Quality
			Saybolt Universal at 100 F		Saybolt Furol at 122 F					
	Min.*	Max.	Min.†	Max.	Max.	Max.	Max.	Max.‡		
1-D	145 or legal	0 05	35	50	....	0.2	0.02	35	§	¶
3-D	150	0.10	35	70	......	0.5	0.02	35	§	¶
4-D	150	0.60	.....	250	....	3.0	0.04	35	§	¶
5-D	150	1.00	......	......	100	6.0	0.08	......	§	. . .
6-D	Limits for No. 6-D fuel are the same as for No. 6 burner oil in the A.S.T.M. Tentative Specifications for Fuel Oils [8] (D-396), unless modified by special negotiations between buyer and seller.									

* Minimum flash point, as stated or required by local fire regulations, fire underwriters, or state laws.

† For viscosities below 35 sec at 100 F Saybolt Universal, other methods than that by Saybolt Universal viscosimeter at 100 F may be used and the results converted.

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

§ So far as known, sulfur content need not be considered as regards combustion characteristics. However, when sulfur content is of consequence—as, for instance, in fuel for engines in intermittent service—the following limits are suggested:

Fuel 1-D, 1.5 per cent max. Fuel 4-D, 2.0 per cent max.
Fuel 3-D, 1.5 per cent max. Fuel 5-D, 2.0 per cent max.

¶ See Table VI and discussion in text.

TABLE VI

IGNITION QUALITY OF DIESEL FUEL OILS *

Ignition Characteristics	Grade of Diesel Fuel Oil		
	1-D	3-D	4-D
Cetane number, delay, minimum	45	35	30
Diesel index number, minimum	45	30	20
Viscosity-gravity number, maximum	0.86	0.89	0.91
Boiling-point gravity number, maximum	188	195	200

* The values in Table VI are equivalent within the limits of error of the test methods used. As the correlations between indexes, and between any index and engine operation in the field, are not completely established as yet, these values should be used in an approximate way only.

[7] "Petroleum Products and Lubricants," *A.S.T.M. Standards*, September, 1936.
[8] *Ibid.*

The properties listed in Tables V and VI are determined in accordance with the following methods of testing.

Flash Point. In the absence of legal or local requirements, the minimum flash point shall be determined in accordance with the Standard Method of Test for Flash Point by the Pensky-Martens Closed Tester;[9] A.S.T.M. Designation, D-93.

Water and Sediment. Standard Method of Test for Water and Sediment of Petroleum Products;[10] A.S.T.M. Designation, D-96.

Viscosity. Standard Methods of Test for Viscosity of Petroleum Products and Lubricants;[11] A.S.T.M. Designation, D-88.

Carbon Residue. Standard Method of Test for Carbon Residue of Petroleum Products[12] (Conradson Carbon Residue); A.S.T.M. Designation, D-189.

Ash. Place 10 grams of the fuel oil in a weighed uncovered platinum crucible of a capacity not greater than 100 ml and heat gently with a Bunsen burner until the oil fires and continues to burn when the flame of the burner is applied to the surface of the oil. Allow the oil to burn without further application of heat. It will continue to burn until practically all of it has disappeared; the time required will be about 50 min. Then place the platinum crucible in a gas or electric furnace and heat to a bright red heat until all carbon is consumed. Cool, weigh, and report the residue as ash expressed as a percentage of the original 10-gram sample.

Pour Point. Standard Method of Test for Cloud and Pour Points;[13] A.S.T.M. Designation, D-97.

Sulfur. Standard Method of Test for Sulfur in Petroleum Oils by Bomb Method;[14] A.S.T.M. Designation, D-129.

Ignition Quality.[15] According to present opinion, combustion knock in a Diesel engine is caused by the accumulation of fuel in its cylinders prior to ignition. Even under conditions that insure ignition, the fuel may not ignite instantaneously but only after a definite delay. The longer this delay, the more fuel accumulates in the cylinders, which then burns unevenly, finally causing audible knock. If the delay is extreme, the engine will fail to operate with the fuel. It is significant that every factor which tends to aggravate knock in spark-ignition (Otto-cycle) engines tends to suppress it in Diesel engines. Fuels of paraffin base, consisting chiefly of saturated straight-chain hydrocarbons, give the smoothest combustion in general, while naphthenes and aromatics burn unevenly at low compression ratios. It is known that highly cracked products give trouble and also that the amount of difficulty is increased as the speed of the engine is increased. In general, slow-speed engines are not very sensitive to changes in ignition quality.

It would be impossible to reduce all types and designs of engines to a common rating for fuel-burning characteristics. Engine factors, such as combustion-chamber design, turbulence, spray characteristics, compression ratio, injection timing, and engine speed, vary greatly in different makes of engines. Ignitibility tests on engines of various types have no theoretical significance but have provided practical data for the types and sizes of engines on which the tests were made. Fuel characteristics are based on a study of actual engine performance, correlated with laboratory ratings

[9] *Ibid.*

[10] *Ibid.*

[11] *Ibid.*

[12] *Ibid.*

[13] *Ibid.*

[14] *Ibid.*

[15] See also "Diesel Fuels—Significance of Ignition Characteristics," J. R. MacGregor, *Society Automotive Engineers Journal*, page 217, June, 1936.

of various test fuels. Ignition quality, one of the most important properties of a fuel from the standpoint of satisfactory engine operation, has been expressed by such characteristics of performance as cetane number, Diesel index number, viscosity-gravity number, and boiling-point gravity number.

Cetane Number.[16] Cetene number was first proposed by Boerlage and Broeze [17] and is the percentage of cetene in a blend of cetene and alphamethylnaphthalene which has the same ignition quality, measured in a given engine under a definite set of engine conditions, as the fuel under test. It has been found that more reliable results may be possible by using cetane rather than cetene. For this reason, the A.S.T.M. classification lists the cetane number, which is determined and expressed in the same manner as the cetene number, namely, as the percentage of cetane in a blend of cetane ($C_{16}H_{34}$) and alphamethylnaphthalene ($C_{11}H_{10}$).

Diesel Index Number. The Diesel index number is calculated from the aniline point and A.P.I. gravity, by

$$\text{Diesel index number} = \frac{(\text{A.P.I. gravity at 60 F}) \times (\text{Aniline point in degrees F})}{100} \tag{6}$$

For the determination of A.P.I. gravity see the "Standard Method of Test for Gravity of Petroleum and Petroleum Products by means of the Hydrometer,"[18] A.S.T.M. Designation, D-287. Aniline point is the lowest temperature at which equal parts by volume of freshly distilled aniline and the test sample of oil are completely miscible. It is determined by heating such a mixture in a jacketed test tube to a clear solution and noting the temperature at which turbidity appears as the mixture is cooled.

Viscosity-gravity Number. The viscosity-gravity number (a measure of paraffinicity) was first proposed by Hill and Coats in 1928, and a modification was proposed by Moore and Kaye [19] in 1934. The number is determined by the equation,

$$G = 1.082A - 0.0887 + [(0.776 - 0.72A) \times \log \log (KV - 4)] \tag{7}$$

where G = specific gravity at 60 F.
A = viscosity-gravity number.
KV = kinematic viscosity in millistokes at 100 F.

Boiling Point-gravity Number. The boiling point-gravity number was first proposed by E. A. Jackson [20] in 1935. The number is determined by the following equation:

$$G = B + (68 - 0.703B) \times \log t_c \tag{8}$$

where G = A.P.I. gravity in degrees.
B = boiling point-gravity number.
t_c = 50 per cent boiling point on A.S.T.M. distillation curve in degrees C.

[16] See also "Cetane Rating of Diesel Fuels," Schweitzer and Hetzel, *Society Automotive Engineers Journal*, page 206, May, 1936; "The Development of Diesel Fuel Testing," Hetzel, *Penn State Engineering Experiment Station Bulletin* 45, 1936.

[17] "Ignition Quality of Diesel Fuels as Expressed in Cetene Numbers." Boerlage and Broeze, *Society Automotive Engineers Journal*, July, 1932.

[18] "Petroleum Products and Lubricants," *A.S.T.M. Standards*, September, 1936.

[19] "Practical Evaluation of the Ignition Characteristics of Diesel Fuel Oils," Moore and Kaye, *Oil and Gas Journal*, page 108, November 15, 1934.

[20] "Boiling Point-Gravity Constant Is Index of Lubricating Oil Characteristics," E. A. Jackson, *Oil and Gas Journal*, page 16. March 21, 1935.

Generally speaking, the larger the number, the more naphthenic the oil; and the lower the number, the more paraffinic the oil. For example, a mid-continent distillate has a 50 per cent boiling point of 215 C and an A.P.I. gravity of 44.2; the boiling point-gravity number is found to be 179 by equation 8.

A recent survey of 37 Diesel engine manufacturers [21] indicates that the majority of this group considers ignition quality, cleanliness, and viscosity to be the most important Diesel-fuel qualities. This applies to the high-, medium-, and low-speed engines and to the four classifications of combustion-chamber design (direct-injection, precombustion, separate chamber, and air cell). Some manufacturers attach primary importance to flash point, distillation, hard asphalt, and sulfur; and less importance to corrosive action, gravity, combustion residue, ash, and Conradson carbon residue.

16. Shale Oil. Probable high prices for petroleum in the future, along with a knowledge of the limitations of our oil reserves, have created an interest in the possible commercial development of our great oil-shale resources. The U. S. Geological Survey and other investigators have studied the shale deposits in Colorado, Wyoming, and Utah. It has been estimated that the shales in this region are capable of producing the surprising total of 100 billion barrels of crude shale oil. Considerable experimentation has been done in the laboratory and in full-sized retorts; the price ranges of petroleum in the United States have not been high enough to stimulate any attempt to retort oil shale on a large commercial scale. Thus it is difficult to predict at what price level, for crude petroleum, shale oil can begin to compete; $2.50 per barrel (42 gal) for average mid-continent crude at the well may be an approximate figure.

Technical, economic, and social problems must be solved before shale oil can be produced on a large scale. The production of shale oil would involve the cheap mining and handling of large quantities of shale, the reduction of the shale in retorts so as to produce a maximum yield of shale oil in good quality, and finally the refining of this oil into gasoline, kerosene, and other commercial products. The development of the shale-oil industry would depend upon attracting large amounts of capital and would necessitate the building of communities in what are now practically barren regions. It has been estimated that the mining of sufficient shale to produce a quantity of oil equal to the present output of petroleum would require the services of a labor army equal to that now engaged in the coal-mining industry.

17. Alcohol. The possibility of the extensive use of ethyl alcohol as a fuel for internal-combustion engines has been frequently discussed,

[21] "Standard Diesel Fuel Specifications Sought in Survey of Manufacturers," Hubner and Murphy, *National Petroleum News*, August 14, 1935. Also "A Study of Diesel Fuels," Hubner and Egloff, *A.S.M.E. Oil and Gas Power Meeting*, August, 1937.

especially by agricultural and political interests. Its attractiveness consists in what might be termed its potential availability. In an emergency alcohol could be produced in practically unlimited quantities from vegetable products, and it is not dependent upon an exhaustible source of supply as are petroleum products. This would probably reduce the amount of available food products, and its economic soundness is open to question.

Alcohol, especially alcohol manufactured from potatoes, has already found considerable application in Europe (notably Germany) as an engine fuel. One of the strongest pleas for alcohol has been that it can be manufactured from a number of by-products available in large quantities and at present of comparatively small value. Such by-products are " black-strap " (cane) molasses, waste liquor from sulfite paper-pulp manufacture, sawdust, straw, and cornstalks. Alcohol can also be manufactured from coal by way of calcium carbide. That is, where power is available at a low price, coal can be converted into a liquid fuel with the aid of water and limestone.

All the above are not merely suggested possibilities but actual processes, and the majority are practiced somewhere on a commercial basis. The question is simply whether in the United States any of these possibilities should be expected to yield alcohol in quantities sufficient to make it a material factor in relieving a future gasoline shortage, at a price which would not seem unreasonable during the transition period from the use of gasoline to alcohol. The high price of gasoline in South America, Central America, the West Indies, the Philippines, and other localities in which there is abundant raw material from which to produce alcohol, but no developed petroleum reserve, has led to an increased use of alcohol or alcohol-gasoline mixtures as an engine fuel. Composite fuels containing 30 to 40 per cent of alcohol, an equal or somewhat larger percentage of gasoline, along with small percentages of benzene or ether can be vaporized by the ordinary carburetor and used in engines of the present types as satisfactorily as gasoline.

Alcohol is only 60 per cent as rich in heat units as gasoline and is 15 to 20 per cent heavier. Commercial alcohol contains approximately 10 per cent of water by weight and from 11,000 to 13,000 Btu per lb; the specific gravity varies from 0.80 to 0.84 at 60 F; and its range of distillation temperatures is 160 to 175 F.

Some of the difficulties encountered in the use of alcohol as a fuel in the usual gasoline engines are: blending difficulties because of stratification; hygroscopic properties (absorption of moisture from atmosphere); fewer miles per gallon (with automobiles); decrease in power with consequent difficulties under heavy loads; difficulties in starting with engine cold; solvent or chemical action of the fuel on tanks and fuel lines, with consequent clogging of fuel pipe or carburetor; rusting and wear of cylinders, ascribed to the lack of proper lubrication; and lack of a reliable standard fuel which may contain variable quantities of water or diluents.

The difficulties encountered in the use of alcohol as a fuel for internal-combustion engines are by no means insurmountable, but are such as might be expected whenever a fuel is used for which the engine has not been primarily designed. Alcohol is in some respects a better fuel than gasoline, but it requires somewhat different handling and some minor changes in design and construction to produce the best results. These include: a considerable increase in the compression of the engine with resultant increase in thermal efficiency, obtainable either by the use of longer pistons, or in the case of an engine with a separate cylinder head by using a shallower combustion chamber; provision for supplying more heat than is ordinarily used to vaporize the charge; slight changes in carburetor adjustment (usually richer mixture); and provision for preventing or minimizing the corrosive or solvent action of the alcohol on metal and other parts. Alcohol produces no carbon in the engine cylinders and is especially adapted as a fuel for air-cooled engines, since the temperature of the engine cylinder may rise much higher before pre-ignition occurs than when gasoline is used. Difficult starting may be overcome by the addition of ether, by priming the cylinders, or by using special fuels for starting the engine.

18. Significance of Exhaust Gas Analysis. An analysis of the exhaust gases of an internal-combustion engine provides a useful check upon the fuel-air mixture and subsequent combustion conditions. If the analysis of the fuel supplied is known, the weight of air needed for complete combustion may be determined from the equivalent combining weights of air (the oxygen in it) with carbon and hydrogen. From the analysis of the resultant exhaust gases, as tested in an Orsat or similar apparatus, the approximate quantity of air that was supplied may be determined.

It has been shown in Art. 7 that, for ideal combustion, 1 volume of hydrocarbon C_nH_m combines with $\left(n + \frac{m}{4}\right)$ volumes of oxygen to form n volumes of CO_2 and $\frac{m}{2}$ volumes of water vapor. Since the composition of air by volume is approximately 79.1 per cent nitrogen and 20.9 per cent oxygen, it follows that 4.78 cu ft of air contain 1 cu ft of oxygen and 3.78 cu ft of nitrogen. Hence, 1 volume of C_nH_m will require $4.78\left(n + \frac{m}{4}\right)$ volumes of air for complete combustion, and if the water formed were considered as occupying no appreciable volume, the volume of the products of combustion would be

$$n + 3.78\left(n + \frac{m}{4}\right)$$

or

$$4.78n + 3.78\frac{m}{4}$$

If, however, as is actually the case, no condensation within the cylinder occurs during the expansion stroke, then an increase in volume due to combustion will occur in the ratio

$$\frac{\text{Initial volume}}{\text{Final volume}} = \frac{4.78\left(n + \frac{m}{4}\right) + 1}{4.78n + \frac{m}{2} + 3.78\frac{m}{4}}$$

hence,

$$\text{Volume ratio} = \frac{4.78n + 1.19m + 1}{4.78n + 1.44m}$$

The conditions obtained with a typical hydrocarbon may be illustrated with sufficient accuracy by a consideration of the combustion of octane, C_8H_{18}.

$$C_8H_{18} + (8 + \tfrac{18}{4})\,O_2 = 8CO_2 + \tfrac{18}{2}\,H_2O$$

$$\text{Initial volume} = 4.78\,(8 + \tfrac{18}{4}) + 1 = 60.75 \text{ cu ft of mixture}$$

$$\text{Final volume} = 4.78 \times 8 + \tfrac{18}{2} + 3.78 \times \tfrac{18}{4}$$

$$= 64.25 \text{ cu ft of products}$$

Hence 1 cu ft of octane vapor will require for complete combustion 59.75 cu ft of air, and the volume of the exhaust products, including that of the water vapor, will be 64.25 cu ft.

Therefore

$$\frac{\text{Initial volume}}{\text{Final volume}} = \frac{60.75}{64.25}$$

so that the volume is about 5.9 per cent greater after combustion than before. The *volume* of the constituents of the products of combustion per cubic foot of octane vapor is as follows:

CO_2	$= 8 \times 1$	$= 8$	cu ft,	12.45 per cent
H_2O	$= 18 \div 2$	$= 9$	cu ft,	14 per cent
N_2	$= 59.75 - 12.5$	$= 47.25$	cu ft,	73.55 per cent
		64.25	cu ft,	100.00 per cent

From the reaction

$$C_8H_{18} + (8 + \tfrac{18}{4})\, O_2 = 8CO_2 + \tfrac{18}{2}\, H_2O$$

it is evident that $\dfrac{12\tfrac{1}{2} \times 32}{(8 \times 12) + (18 \times 1)} = \dfrac{400}{114} = 3.51$ lb of oxygen are required for the complete combustion of 1 lb of octane. This weight of oxygen is equivalent to $3.51 \times 4.31 = 15.13$ lb of air or 198.2 cu ft at 14.7 lb per sq in. and 60 F. The *weight* of the constituents of the products of combustion per pound of octane is as follows:

$$\begin{array}{rlrr}
CO_2 = \tfrac{352}{114} & = & 3.09\text{ lb}, & 19.15\text{ per cent} \\
H_2O = \tfrac{162}{114} & = & 1.42\text{ lb}, & 8.80\text{ per cent} \\
N_2 = 15.13 - 3.51 & = & 11.62\text{ lb}, & 72.05\text{ per cent} \\
\hline
 & & 16.13\text{ lb}, & 100.00\text{ per cent}
\end{array}$$

In an actual engine, a series of fuel-air mixtures may be used, but there is only one value, or ratio, for which complete combustion occurs, the other mixtures being weaker (in fuel) or richer than this one correctly proportioned mixture.

For the weaker mixtures there is an excess of air and it is evident that the exhaust products will contain free oxygen as well as carbon dioxide. On the other hand, with rich mixtures, there will not be enough air to burn the fuel completely. When a hydrocarbon is only partly burned carbon monoxide is formed as well as carbon dioxide and free hydrogen. For the several types of fuel-air mixtures the theoretical resultant exhaust gas analysis will be as follows:

1. *Weak mixture*; CO_2, O_2, and N_2.
2. *Theoretically correct mixture*; CO_2 and N_2.
3. *Rich mixture*; CO, CO_2, N_2, H_2, and traces of C_nH_m.
4. *Very rich mixture*; same products as in 3, with sooty deposits of free carbon.

Figure 3 illustrates the results of a number of exhaust-gas analyses for an engine using gasoline as a fuel. The curves show the volumetric percentages of the exhaust gases for various fuel-air mixtures over the entire operating range of the engine. With a theoretical proportion of about 14.5 parts (by weight) of air to 1 part of gasoline the exhaust gases contain about 13 per cent CO_2 by volume and about 0.5 per cent each of CO and O_2. If the air is present in excess (weak mixture) there is more free oxygen and less carbon dioxide in the exhaust; if the mixture is richer, there is no free oxygen and the amount of carbon monoxide increases while the carbon dioxide decreases. Although the analysis of the exhaust gas may vary somewhat with the composition of the gasoline, the curves shown in Fig. 3 may be considered representative of the commercial grades of gasoline.

Combustion in the actual internal-combustion engine is not perfect, although the incompleteness, as evidenced by the presence of carbon monoxide in the exhaust gases, is not very marked. In addition, chemical tests on the condensed water vapor from the exhaust often reveal the presence of traces of aldehydes (CH_nO compounds) and other partly oxidized organic materials. For this reason, the air-fuel ratio supplied to the engine cannot be computed from the exhaust-gas com-

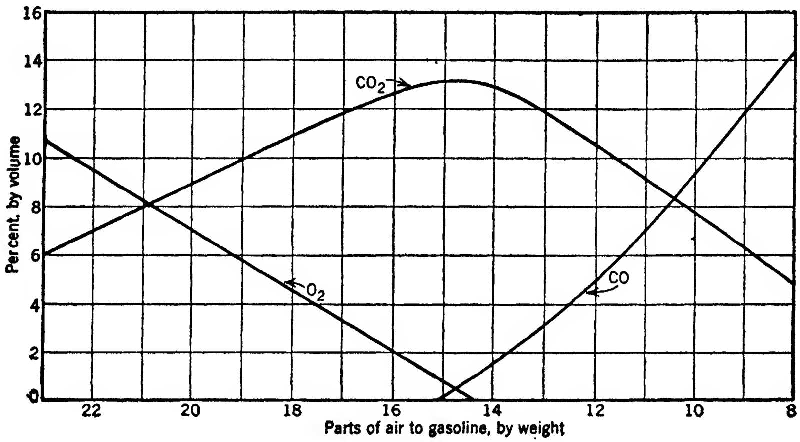


FIG. 3. Composition of exhaust gases from a gasoline engine for various fuel-air mixtures.

position with any great accuracy. In general, it is found that the discrepancy becomes greater as the supplied air-fuel ratio is made larger. For detailed information on calculations based on exhaust-gas analyses and their relation to the supplied air-fuel ratios the reader is referred to " Exhaust Dry Analysis Calculations " by E. H. Lockwood, *S.A.E. Journal,* Vol. 21 (1927), page 571; and Vol. 23 (1928), page 314: also " Report on Air-Fuel Ratio Tests " by H. W. Best, *S.A.E. Journal,* Vol. 25 (1929), page 533.

PROBLEMS

1. Given a gaseous fuel analysis, percentages by volume: H_2, 30; CO, 37; and N_2, 33. (*a*) Determine the higher heating value in Btu per cubic foot and per pound of fuel at 14.7 lb per sq in. abs and 60 F. (*b*) Find the theoretical air required for combustion in cubic feet per cubic foot of fuel, pounds per cubic foot of fuel, and pounds of air per pound of fuel. *Ans.* (*b*) 2.3 lb air per lb fuel.

2. Using the fuel of problem 1, determine: (*a*) The partial pressures of the products of combustion. (*b*) The temperature at which the water vapor formed begins to condense. (*c*) The apparent volume of the products of combustion, per cubic foot of fuel, at 14.7 lb per sq in. abs and 60 F. (*d*) The volume of the products of combustion as measured by an Orsat apparatus (with water vapor condensed), at 14.7 lb and 60 F.

3. Given a fuel analysis, percentages by volume: H_2, 53; CO, 6; CH_4, 35; CO_2, 2; and N_2, 4. (*a*) Determine the higher heating value per cubic foot and per pound of fuel. (*b*) Find the volume of air supplied and the products of combustion formed per cubic foot of gas when the gas is burned with 20 per cent excess air. *Ans.* (*a*) 20,050 Btu per lb. (*b*) Products, 6.40 cu ft.

4. A gaseous fuel has the following analysis, percentages by weight: H_2, 8; CO, 43; CH_4, 5; C_2H_4, 7; CO_2, 14; and N_2, 23. (*a*) Find the volume of air required for combustion, at 14.7 lb and 60 F, per pound of fuel. (*b*) What would be the volume of the products of combustion, at 14.7 lb and 60 F, per pound of fuel if 50 per cent excess air were supplied? *Ans.* (*b*) 129.8 cu ft.

5. A four-stroke cycle, single-cylinder, single-acting internal-combustion engine uses natural gas of the following analysis, percentages by volume: CH_4, 92.6; C_2H_6, 4.3; CO_2, 0.6; and N_2, 2.5. The engine delivers 80 brake horsepower and operates at 200 rpm; thermal efficiency based on ihp, 22 per cent; mechanical efficiency, 82 per cent; and volumetric (charging) efficiency, 88 per cent. (*a*) Find the higher heating value per cubic foot of gas. (*b*) Calculate the cubic feet of air supplied per cubic foot of gas, assuming 25 per cent excess air. (*c*) What is the displacement of the piston in cubic feet per minute? (*d*) Determine the cylinder size (bore and stroke) if the stroke is 1.5 times the diameter. *Ans.* (*d*) 16 × 24 in.

6. Determine the volume of the products of combustion, at 15 lb per sq in. abs and 450 F, when 1 cu ft of methane is burned with 25 per cent excess air.

7. A fuel oil consists of 83 per cent carbon and 12 per cent hydrogen, by weight. Calculate the percentage of excess air, by weight, when the dry products of combustion are, percentages by vólume: CO_2, 10; O_2, 10; and N_2, 80.

8. A gaseous fuel has the following percentage composition by volume: H_2, 50; CO, 5; CH_4, 30; CO_2, 2; O_2, 3; and N_2, 10. Calculate the minimum volume of air required for the combustion of 1 cu ft of the gas. *Ans.* 4.04 cu ft.

9. The gaseous fuel in problem 8 has a heating value of 508 Btu per cu ft and is mixed with air in the ratio of 1 to 7.03, by volume. (*a*) Determine the heating value per cubic foot of the fuel-air mixture. (*b*) What percentage excess air was supplied? (*c*) Find the volumetric composition of the exhaust gases.

10. If a gas engine were supplied with the fuel-air mixture of problem 9, find the diameter and stroke for a single-cylinder engine to develop 60 hp at 100 explosions per minute. Assume a stroke-bore ratio of 1.5, a thermal efficiency of 28 per cent, and a volumetric (charging) efficiency of 80 per cent. *Ans.* $13\frac{3}{4} \times 20\frac{3}{4}$ in.

11. A gas engine uses producer gas of the following analysis, percentages by volume: H_2, 8; CO, 24; CH_4, 3; C_2H_4, 0.5; CO_2, 5; O_2, 0.5; and N_2, 59. The gas is burned with 30 per cent excess air; the pressure and temperature at the beginning of compression are 14 lb per sq in. abs and 120 F, respectively; the pressure at the end of compression is 140 lb per sq in. abs. Assuming compression and expansion according to $PV^{1.35} = C$, find: (*a*) The amount of heat generated per cubic foot of fuel-air mixture in the cylinder. (*b*) The thermal efficiency of the cycle.

12. Using the data given in problem 11, find: (*a*) The cubic feet of producer gas at standard conditions (14.7 lb and 60 F) used per ihp. (*b*) The mean effective pressure of the cycle.

13. What is the weight of the products of combustion when 8 lb of kerosene are burned with 20 per cent excess air?

14. What is the volume of the products of combustion at 14.7 lb per sq in. abs and 60 F when 8 lb of gasoline, C_8H_{18}, are burned with 20 per cent excess air?

15. Find the theoretical and actual volumes of the products of combustion at 15 lb per sq in. abs and 400 F when 1 lb of kerosene is burned with 20 per cent excess air.

16. Determine the partial pressures of the products of combustion at 14.7 lb per sq in. abs when 1 lb of kerosene is burned with 20 per cent excess air.

17. Determine the pounds of air required for the complete combustion of 1 lb of alcohol, C_2H_6O. Find the probable volumetric composition of the products of combustion.

18. A mixture of 60 per cent (by weight) alcohol, C_2H_6O, and 40 per cent gasoline, C_8H_{18}, is used as a fuel; the alcohol contains 10 per cent (by weight) of water. (*a*) Find the pounds of air necessary for complete combustion of 1 lb of this fuel. (*b*) Determine the volume of the products of combustion, for 1 lb of fuel, at a pressure of 15 lb abs and a temperature of 400 F.

19. The vapor pressure of heptane, C_7H_{16}, at 60 F is 6.2 in. of mercury. Calculate the degree of saturation of a fuel-air mixture, consisting of 1 part of heptane to 18 of air by weight, at 14 lb per sq in. abs and 60 F.

20. Assuming gasoline to be octane, C_8H_{18}, calculate independently (*a*) the volume of air required per cubic foot of gaseous fuel, and (*b*) the weight of air per pound of fuel for complete combustion. (*c*) Also determine *b* from *a*.

21. Determine the higher and lower heating values, Btu per cubic foot at 14.7 lb per sq in. abs and 32 F, of a *mixture* of octane and air of correct proportion for complete combustion. Assume higher heating value of octane to be 20,300 Btu per lb.

22. (*a*) What weight of octane will saturate 1 lb of air at 14.7 lb per sq in. abs and 32 F? (*b*) Find the percentage of saturation of the correct octane-air mixture in problem 21. (*c*) Find the density in pounds per cubic foot of the saturated mixture at 14.7 lb and 32 F. Assume the vapor pressure of octane at 32 F to be 5.2 in. of mercury. (NOTE. One cubic foot of air at any temperature is said to be saturated with liquid-fuel vapor when that vapor exerts its full pressure at that temperature.) *Ans.* (*a*) 0.83 lb. (*b*) 7.94 per cent. (*c*) 0.122 lb.

23. The fuel used in a Diesel engine has the following percentage composition by weight: C, 86.68; H_2, 11.7; O_2, 0.14; and S, 1.48. The volumetric analysis of the dry exhaust gas shows a percentage of CO_2, 8.14; CO, 0.02; O_2, 10.24; and N_2, 81.6. Find: (*a*) The theoretical weight of air for complete combustion. (*b*) The pounds of dry exhaust gases per pound of fuel. (*c*) The pounds of water vapor and sulfur dioxide formed during the combustion of 1 lb of the fuel. (*d*) The percentage of excess air. *Ans.* (*b*) 26.3 lb. (*c*) 1.053 and 0.03 lb.

24. (*a*) Find the partial pressure of the steam present in the exhaust gases of problem 23, neglecting the sulfur dioxide and assuming the total pressure to be 14.7 lb per sq in. abs. (*b*) Calculate the actual total heat, Btu per lb, of the steam at 500 F and compare it with the value obtained on the usual assumption that the pressure of the steam is 14.7 lb per sq in. abs. Use 29.7 as the mean molecular weight of the dry exhaust gases.

25. Calculate the percentages of excess air when 1 lb of pentane, C_5H_{12}, is mixed with 18 lb of air. Estimate the probable volumetric composition (in per cent) of the products of combustion.

26. A gasoline engine is supplied with the fuel-air mixture of problem 25. If the thermal efficiency is 25 per cent and the volume of mixture drawn in during the suction stroke is 80 per cent of the piston displacement, determine the diameter and stroke for a four-cylinder, four-stroke cycle, single-acting engine to develop 30 hp at 2200 rpm. Use a stroke-bore ratio of 1.25 and heating value of pentane as 19,200 Btu per lb.

CHAPTER III

EXPLOSION AND COMBUSTION

When a gaseous or atomized liquid fuel is mixed with air (or oxygen) in certain proportions, the resulting mixture is explosive and the presence of a small flame or spark will cause the elements of the mixture to combine chemically with more or less violence. The fuel-air mixture for Otto-cycle engines is a gas or fuel-vapor mixed with air. Hence, with liquid fuels, special devices are required to atomize the fuel and subsequently mix the finely divided particles of fuel with air. Atomization is effected by spraying the liquid fuel through a small nozzle or jet into a stream of moving air. The air movement is produced by the suction stroke of the piston and the resultant reduction of the pressure in the cylinder. See Fig. 11. The higher atmospheric pressure causes outside air to flow through the carburetor and manifold into the cylinder. In order to increase the volumetric efficiency of an engine, especially at high altitudes, air may be forced into the cylinder by a " supercharger "; rotary compressors or centrifugal blowers are generally used for this purpose.

In Diesel engines with solid injection a stream of finely atomized liquid fuel is injected into the highly compressed and highly heated air contained in the engine cylinder. Problems connected with the high-pressure injection of liquid fuels include accurate measuring of small quantities of oil, timing of fuel injection, rate of fuel injection, fineness and form of oil spray, and distribution of the atomized oil in the combustion space of the cylinder.

19. Carburetors, Carburetion, Supercharging, and Manifolding. A carburetor is a device in which the correct fuel-air mixture is formed, at all loads and speeds of the engine, by subdividing or atomizing a liquid fuel and mixing the finely divided particles of fuel with air. Better results would be obtained with complete vaporization of the fuel, but evaporation of the fuel is not necessary if the liquid particles are finely divided and thoroughly mixed with air. An ideal fuel-air mixture for the intake manifold of an engine should be a homogeneous mixture throughout to permit development of the maximum possible power and should have composition or strength to develop maximum economy under each condition of engine operation. In order to accomplish this the carburetor should do the following things: correctly meter the liquid fuel to the incoming air supply, offer low frictional resistance which results in high volumetric capacity, supply the necessary means of enrichment of mixture for cold starting and warming up, and supply suitable instantaneous mixture for engine acceleration.

Figure 4 shows a line diagram of a double-tube *updraft* carburetor. The tip of the nozzle G is placed slightly higher than the normal liquid level in the float chamber in order to avoid leakage of fuel when there is no air flow. When air is flowing through the carburetor the resulting vacuum in the manifold of the engine causes the fuel to rise to the top of tube G and discharge into the stream of air. The fuel-air mixture flows past the throttle valve T (when open) and then through the mani-

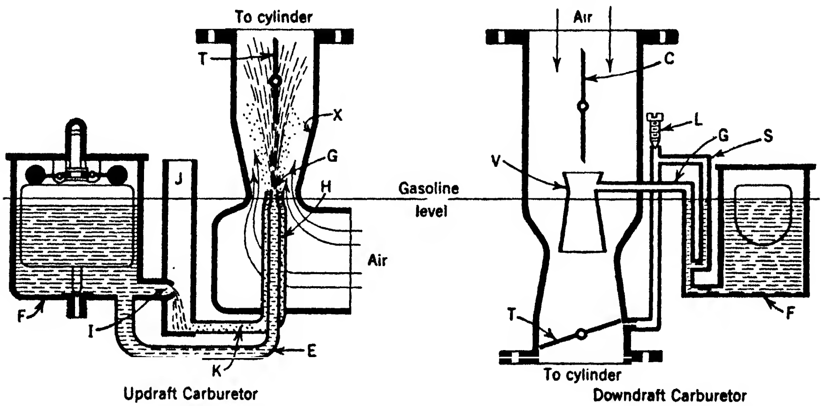


FIG. 4. Diagrammatic sketches of updraft and downdraft carburetors.

fold to the engine cylinder. The level of gasoline is maintained constant in float chamber F by means of a float which operates the needle valve. As the level falls, the needle valve opens, allowing more gasoline to enter the float chamber. Atmospheric pressure exists on top of the gasoline surface in the float chamber and pressure around the top of the tube causes the gasoline to flow into the air stream.

The outer tube H, updraft carburetor, Fig. 4, connects to well J which receives its fuel through the orifice I. For sudden fuel demands the level in J falls below the orifice I. As soon as the level at J falls below the level in the float chamber the flow of fuel through orifice I will increase, and will become a maximum when the level is below the orifice. However, there is sufficient fuel in the well J to provide for sudden demands of short duration. After the level in J is below the orifice, air will be drawn in with the fuel through K, as shown in Fig. 4. This mixture of air and fuel is further broken up and atomized after it reaches the main air stream above G. The reservoir J is sometimes referred to as an accelerating well since it supplies fuel for acceleration of the engine speed. A carburetor of this type would give a "richer" mixture as the engine speed or air flow is increased. Commercial carburetors are designed to overcome this tendency, and to give the desired fuel-air mixture over the entire range of engine speed.

For some years engine designs included a vacuum tank as a part of the fuel-feed system, and the carburetor float chamber was filled by gravity from this tank. This made the location of the carburetor below

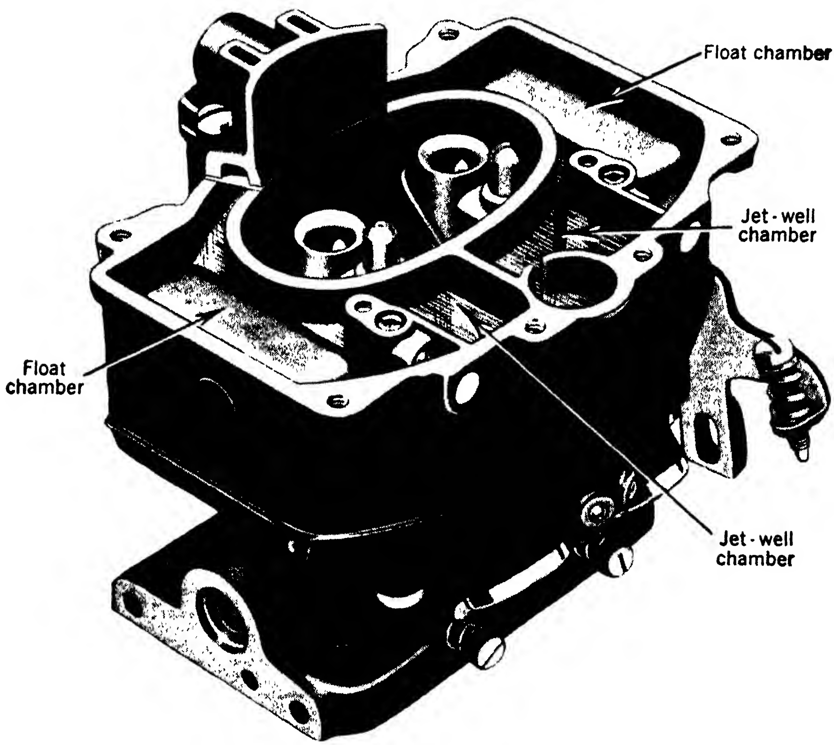


Fig. 5. Open view of lower half of Stromberg Aerotype duplex downdraft carburetor, showing two float chambers and two jet chambers.

the intake manifold a necessity if sufficient head was to be maintained on the float chamber. In recent years the vacuum tank has been replaced by a mechanical fuel pump operated from some engine shaft. With this development, the possibility of putting the carburetor on top of the manifold was conceived and *downdraft* carburetion has practically displaced the updraft principle. A diagrammatic sketch of a downdraft carburetor is shown in Fig. 4. The point at which the fuel joins the air stream in the venturi V is above the fuel level in the float chamber F. Fuel for low or idle speeds is taken through idle tube S, where it is mixed with air from the air bleeder L at the top of the idle tube. When the throttle valve T is closed, the fuel-air mixture is drawn only from the idle discharge hole owing to the high suction at this point. When the throttle is slightly opened, less fuel enters through S and suction is also placed on the high-speed tube G to feed additional fuel. Further opening of the throttle T accelerates the flow of fuel in G and stops fuel discharge from the idle discharge hole.

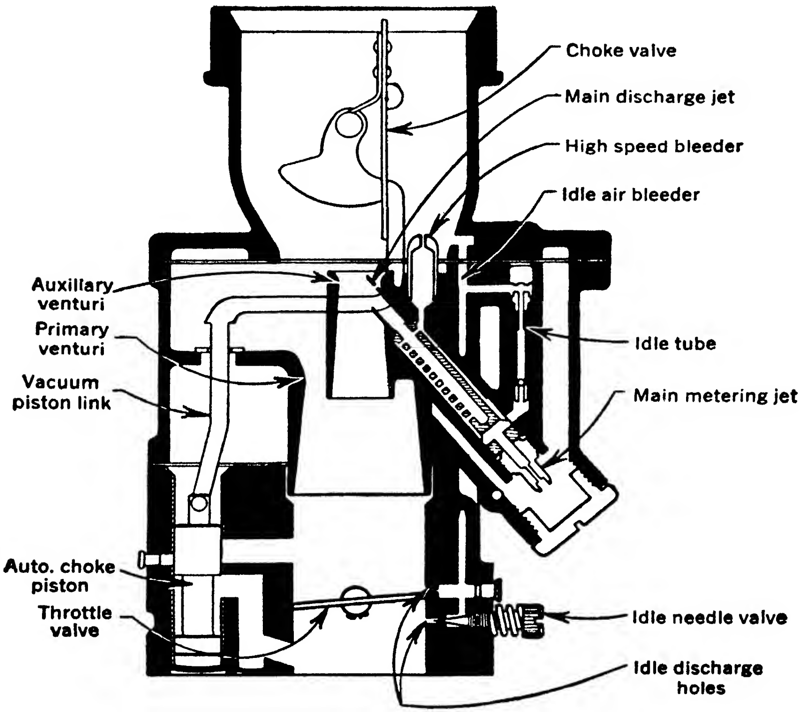


FIG. 6. Cross section, Stromberg Aerotype duplex downdraft carburetor, showing one metering jet and automatic choke control (see also Fig. 8).

With downdraft carburetion it is generally possible to obtain high volumetric (charging) efficiency and thus more power by bringing a cooler mixture into the cylinder. Difficulties with starting have been largely eliminated by the adoption of the automatic choke. The Stromberg Aerotype duplex is a typical downdraft gasoline carburetor provided with air-bled jets to insure a lean mixture at normal speeds and to supply the richer mixture necessary for maximum power and high speed. Fuel enters the carburetor through the gasoline inlet and flows through the needle valve seats into the two float chambers. See Figs. 5 and 7. The fuel chamber completely surrounds the entire body; this feature enables the fuel to be maintained at the proper level under varying operating conditions. Another feature in the float-chamber design is the baffles which help to control surging of the fuel on sharp, sudden turns. The float needle valve is attached to the float lever by a clip, see Fig. 7, which makes it positive acting at all times. The floats automatically maintain the correct fuel level in the carburetor jet-well chambers. The fuel then flows through the main metering jet,

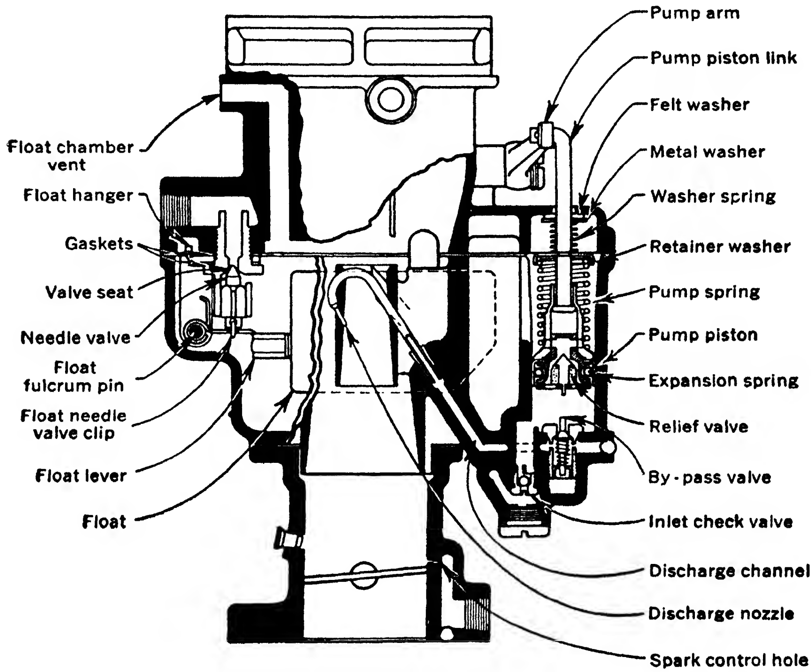


FIG. 7. Cross section, Stromberg Aerotype duplex downdraft carburetor, showing float, fuel inlet control, accelerating jet, and accelerating pump.

Fig. 6, to the main discharge jet or to the idle tube and idle discharge holes, depending on the position of the throttle valve. At speeds less than 12 miles per hour the throttle valve is practically closed and the fuel-air mixture enters through the idle discharge holes. When the throttle valve is closed, a high vacuum is created in the manifold of the engine and fuel is thereby lifted up the idle tube, mixed with air from the idle air bleed, and subsequently drawn through the idle discharge holes into the engine manifold.

At speeds from 12 to 20 miles per hour the fuel is delivered into the path of the air stream through both the idle discharge holes and the main discharge jet. Above 20 miles per hour virtually all the gasoline is delivered through the main discharge jet. The main discharge jet has its outlet at the center of the throat of the small venturi tube; this location insures high air velocity at the tip of the nozzle and effective atomization of the fuel. Atomization of the fuel is aided by supplying air through the high-speed air bleed opening into the central passage of the main discharge jet through a series of small holes located below the fuel level; this additional air is chiefly a compensating means to overcome the tendency for mixtures to become richer at high air velocities. The small venturi tube has its outlet end near the throat of a large venturi; since there is an increased velocity and reduction of pressure at the throat of a venturi tube, it will be seen that considerable mixing and atomizing of the fuel will take place with this arrangement.

For maximum power or high-speed engine operation a leaner mixture is desired than that necessary for normal throttle opening. For part throttle opening, fuel is supplied through the main metering jet to approximately 75 miles per hour. At and above this speed the by-pass valve, Fig. 7, is forced down by the action of the accelerating pump piston and fuel is forced through the pump discharge channel and nozzle into the small venturi tube. Fuel is supplied continuously through this nozzle with the throttle valve wide open. The pump arm is actuated by a rod which is under the control of the throttle lever. For rapid acceleration and flexibility, it is necessary to supply momentarily an extra amount of fuel when the throttle is opened quickly. On the upstroke of the accelerating pump piston, actuated by a spring, fuel is drawn into the lower part of the pump chamber through the inlet check valve; on the downstroke of the pump piston, forced down by the pump rod, the pressure on the liquid closes the check valve and forces open the by-pass valve. The fuel is then discharged through the pump discharge nozzle into the carburetor barrel. When the throttle is suddenly opened part way, only a small amount of fuel is discharged from the pump. In this way it is possible to retain satisfactory fuel economy at normal speeds without sacrificing acceleration, flexibility, and power at higher speeds.

The choke valve must be fully closed for starting and partly closed during the period of warming up the engine. The Stromberg automatic choke control is a device that eliminates all methods of hand-operated chokes; it may be built completely into the carburetor, as shown in Fig. 8. A thermostatic spring is connected to the carburetor choke valve stem; the tension of the thermostat is controlled by an electric heat unit connected directly to the choke valve stem, making the entire unit intact. The vacuum piston used to open the choke valve, when the engine begins to fire, is also built into the carburetor. In this carburetor the fast idle feature which supplies fuel during the cranking period and keeps the engine from stopping while warming up, is accomplished by a fuel passage entering the throttle barrel below the throttle valve to feed the additional fuel with closed throttle. Figure 8 shows the automatic choke control and fast idle with choke valve partially open and the engine firing.

As the engine becomes cold, the thermostatic spring also cools and gradually gains tension to close the choke valve. When the ignition switch is closed and starting of the engine is desired, a current of about 1 ampere flows through the electric heat unit; in this way the choke-valve action is responsive to both an initial temperature and a time component. During cranking, there is very little depression beyond the throttle, and the thermostatic spring is set with enough tension to insure a very rich cranking mixture at low temperature. As soon as the engine starts to fire and speeds up, the resulting depression beyond the throttle operates on the vacuum piston to overcome partially the thermostatic spring effort, thus decreasing the mixture enrichment. As the engine warms up the thermostat tension becomes less, decreasing the mixture enrichment; but all through this period, if the throttle is opened wide at low speed, the release of manifold depression tends to give a slightly richer mixture than at partial throttle opening. Finally, of course, the thermostat warms up enough to hold the choke valve wide open without the help of the vacuum piston.

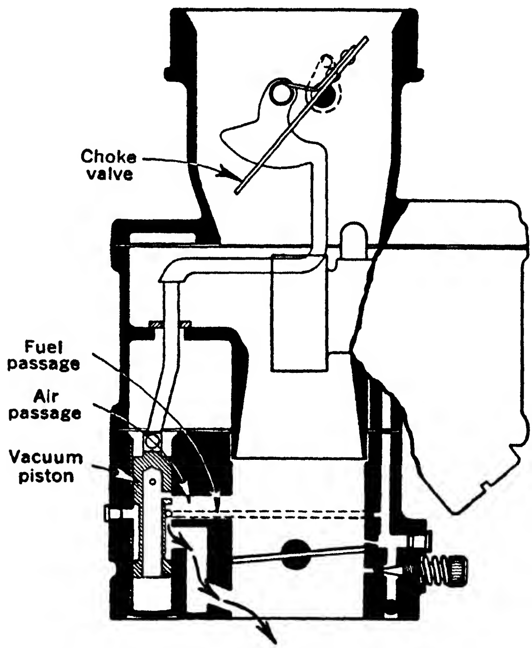


FIG. 8. Cross section, Stromberg Aerotype duplex downdraft carburetor, showing one metering barrel and automatic choke control.

A material improvement in control of the mixture during this warming-up period has been obtained with this carburetor by making the vacuum piston the means of augmenting the throttle opening, and by having its motion also uncover an auxiliary fuel passage, Fig. 8. The metering suctions are quite low in modern carburetor practice, which means that accurate control of the mixture by choke position is very difficult, the scale of accuracy needed for this running into thousandths of a pound-inch torque.

The thermostat is adjusted so that in the warming-up period, the choke is so nearly open that its own enriching effect is negligible; a predetermined fast idle and mixture enrichment being obtained instead by air and fuel orifices uncovered by the vacuum piston. This gives a powerful starting air and fuel charge, along with a smooth warming-up mixture, not unduly rich—all without being sensitive to small variations in choke setting.

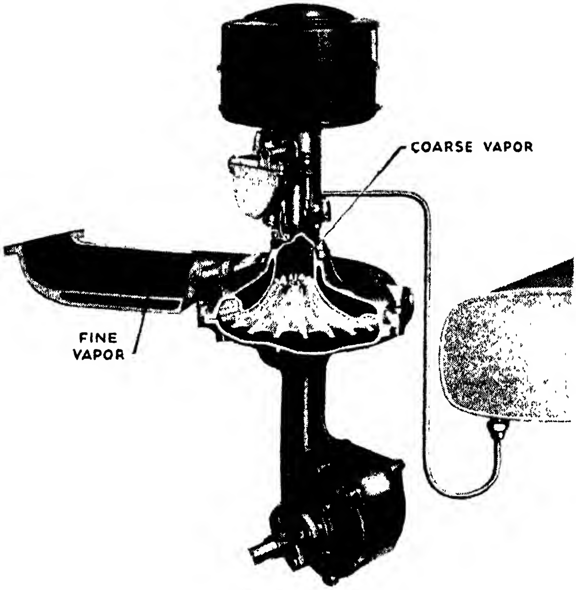


FIG. 9. Graham supercharger.

Some carburetors are provided with an auxiliary air valve so arranged that at high engine speeds the valve will open and admit additional air to dilute the mixture. For low speeds, the mixture which gives good results is 12 to 14 parts of air to 1 part of gasoline by weight. At high engine speeds, 15 to 17 parts of air to 1 part of gasoline are usually required for maximum thermal efficiency and reduced fuel consumption.

The *supercharger* is a device, such as a blower, compressor, or pump, for increasing the weight of the air charge for an internal-combustion engine over that which would normally be drawn in through the pumping action of the pistons. It is used to compensate for the lower density of air in altitude operation of aircraft engines or for the deficiency of air charge in high-speed automotive operation. The Graham supercharger, Fig. 9, is mounted on the side of the engine between the downdraft carburetor and intake manifold. The swiftly moving vanes of the supercharger rotor draw the fuel-air mixture through the carburetor. This mixture is whirled and more intimately commingled and then forced through the manifold into the cylinders at a pressure of about 5 lb per sq in. The supercharger insures a thoroughly atomized fuel-air mixture for each cylinder and distributes the mixture equally to each cylinder.

The extent to which supercharging can be utilized on any engine depends upon a number of factors, such as the octane number of the fuel, piston compression ratio, fuel-air mixture strength, mixture temperature, ignition advance, valve and port areas, and the shape and size of the combustion chamber. The purpose for which the engine is to be used must be taken into consideration, and although it may be true generally that for the same increase in over-all compression ratio, more power can be obtained by increase in supercharging pressure rather than by increase of piston compression ratio, such a division of over-all compression ratio adversely affects the fuel consumption.

Superchargers have not come into general use with automotive engines, as their utility seems to be confined to high-speed operation. The simpler means of raising the engine compression ratio provides greater efficiency in the low- and medium-speed ranges, together with a reduction of fuel consumption and a lowering of operating temperatures without the added first cost or maintenance of a high-speed unit. The supercharger is quite generally employed, however, in conjunction with maximum permissible compression ratios, as a means of getting added horsepower from aircraft engines. In this case its use seems advantageous, as the problem of manifolding a radial engine to get maximum power without supercharging is not easy.

It has frequently been claimed that by means of supercharging it is possible to obtain an increase in power output and improve the fuel economy. It is not possible to make good both claims. If the additional air were used for anything like a proportional increase in power, and if the maximum pressure were limited as it must be in the actual engine, then the effect would be to curtail the ratio of expansion, and therefore to lower somewhat the efficiency. If it were used to increase the air-fuel ratio, then it would provide an improvement in efficiency by increasing somewhat the ratio of expansion, but not in power output. Nature offers an unlimited supply of air at a pressure of about 14.5 lb per sq in.; the engine is bound to take what is offered, but it must be prepared to pay extra if it declines to accept what is available. The engine must pay for the additional air either by the power required to drive the mechanically driven centrifugal blower or by accepting the inconveniences and limitations of an exhaust-driven turbo-compressor type of supercharger.

Upon leaving the carburetor the fuel-air mixture passes through a branched pipe connection called the intake *manifold*, see Fig. 10, which distributes the mixture to each of the cylinders of a multi-cylinder engine. To obtain good efficiency and capacity, the density and strength of the mixture reaching all the cylinders should be the same. In order to insure these results the amount of fuel entering, and the pressure drop between the carburetor and the inlet valves of the engine, should be equal in all the branches of the manifold. If the fuel is not completely vaporized before reaching the branching of the manifold it may be difficult to obtain proper distribution of the fuel; this difficulty increases as the volatility of the fuel decreases.

There is no specific manifold design that may be considered to be better than others, but important characteristics to be desired are: a maximum air volume supplied without undue resistance; the air uniformly apportioned at the various division points to insure equal volumetric efficiency in all cylinders; the fuel as a liquid evenly distributed, especially when starting and warming the engine and when the

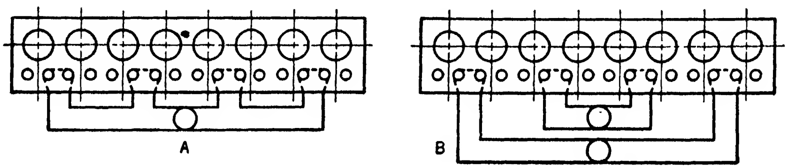


FIG. 10. Single- and double-manifold arrangements for an eight-cylinder engine.

fuel is partially vaporized under normal operating conditions; a mixture distribution providing equal temperatures in each leg of the manifold, irrespective of exterior conditions; easy starting of the engine when cold, made possible by proper flooding and atomization; the fuel heated and vaporized only sufficiently for quick warm-up, then the mixture temperature properly controlled to assure satisfactory operation with a particular fuel; and the design not too complicated for mass production and low cost.

After vaporization of the fuel the temperature of the mixture is less than that of the entering air because the latent heat of vaporization of the fuel is supplied by the air; this interchange of heat is called adiabatic saturation. The latent heat of vaporization of gasoline is about 140 Btu per lb and the specific heat about 0.5; for an air-fuel ratio of 8 to 1, complete vaporization would produce a temperature drop of about 58 F, and for a ratio of 14 to 1 a drop of about 36 F. This decrease of temperature in the manifold may result in the freezing of the moisture in the air, but the temperature drop would be less if only part of the fuel were vaporized. If the air supplied were cold it would not be possible to vaporize the fuel completely by heat absorbed from the air, because the air would become saturated before vaporizing all the fuel. Complete vaporization of the fuel may be effected by utilizing some of the heat of the cooling water or of the exhaust gases; however, preheating is generally objectionable because it decreases the density of the charge and thereby reduces the power output of the engine. Preheating methods used are: heating the air before it enters the carburetor, heating the mixture in the manifold, and heating the manifold locally (by a hot spot) at some place where the drops of fuel impinge so as to apply heat only to the liquid.

The suitability of a manifold for a certain engine can be determined only by tests. Subsequent changes may then be made to obtain more nearly the desired design characteristics enumerated above. To assure more equal distribution of the mixture to all cylinders and to obtain more power, a duplex carburetor and a double manifold are generally used for engines having eight or more cylinders. In a duplex carburetor, fuel is supplied from one float chamber and there is only one air entrance but there are a set of venturi tubes, a main metering jet, an idle system with an adjustable needle, a throttle valve, and a pump discharge nozzle for each half of the carburetor. The suction strokes overlap with the single manifold arrangement, *A* in Fig. 10, and all the cylinders do not receive the same amount of charge. The double manifold, *B* in Fig. 10, consists of two independent induction systems, one of which supplies cylinders 1, 2, 7, and 8, while the other supplies cylinders 3, 4, 5, and 6.

20. Mechanism of the Combustion of Gaseous Hydrocarbons.[1] Three methods have been advanced to explain the way in which hydrocarbons burn: (*a*) that oxygen combines with the hydrogen in preference to or before the carbon; (*b*) that oxygen combines with the carbon in preference to or before the hydrogen; and (*c*) that there is an initial association of oxygen with the hydrocarbon molecule, forming intermediate "hydroxylated" compounds, which, in turn, burn or are broken down thermally.

Before the year 1892, it was generally believed that hydrogen was the more combustible part of the hydrocarbon and, upon the burning of a hydrocarbon, was "singed-off" the molecule, leaving free carbon, as indicated by the reaction

$$2C_2H_2 + O_2 = 4C + 2H_2O$$

In the year 1884, one experimenter, speaking of the combustion of hydrocarbons, said: "There is a race for the oxygen of the air between the

[1] For a complete treatment of this subject, see "Combustion, Flames, and Explosions of Gases," Lewis and von Elbe, 1937, Cambridge University Press (American agents, The Macmillan Company, New York). Also "Kinetics of Homogeneous Gas Reactions," Kassel, 1932, Chemical Catalog Company, New York.

two constituents of ethylene (for example), and hydrogen, being the fleeter of the two, gets to the oxygen first, and is burnt to water."

In the year 1892, the older concept of the preferential combustion of hydrogen had to be discarded, since it was found that a mixture of equal volumes of ethylene and oxygen yields during combustion twice the volume of the mixture, the products being hydrogen and CO, according to the equation

$$C_2H_4 + O_2 = 2H_2 + 2CO$$

The explanation advanced was that the carbon of the hydrocarbon was preferentially burned to CO, leaving the hydrogen thus liberated to combine with excess oxygen. This theory received additional proof when it was found that hydrogen is present in the inner parts of aerated hydrocarbon flames.

Neither of the above preferential-combustion concepts, however, explains all the known facts regarding the combustion of hydrocarbons. A number of investigations have been carried out which indicate that the combustion of gaseous hydrocarbons takes place by an initial interaction of the hydrocarbon with oxygen, forming an intermediate, unstable, " oxygenated " or " hydroxylated " compound. These investigations were of three general types: *static experiments*, in which the gas or hydrocarbon was heated in bulbs with varying proportions of oxygen, at temperatures of 480 to 660 F, the contents of the bulbs after definite periods of time being cooled and analyzed; *dynamic experiments*, in which the gaseous mixtures were recirculated over a surface of porous porcelain kept at a constant temperature in a furnace, cooling and scrubbing arrangements being provided for the removal of intermediate products; *explosion experiments*, in which the hydrocarbons were exploded with varying amounts of oxygen (also mixtures of oxygen and hydrogen or oxygen and CO) and the products analyzed. The fuels used in these experiments included the paraffin hydrocarbons (C_nH_{2n+2}) from methane to hexane, the olefines (C_nH_{2n}), acetylene, and kerosene.

The following is a summary of the results obtained in the experiments outlined in the previous paragraph. There is a combination of the hydrocarbon with oxygen preliminary to final combustion. Hydrocarbons combine with oxygen to form alcohols and aldehydes (CH_nO compounds) as a preliminary step to burning to CO, CO_2, and H_2O; this was found to be true at all temperatures. Contrary to the ideas expressed in earlier theories, there is no selective combustion of either the carbon or the hydrogen of the hydrocarbon. The initial addition of oxygen to the hydrocarbon molecule forms an alcohol with which, in turn, more oxygen reacts, forming an aldehyde. This aldehyde, in many cases breaks down into the intermediate combustible gases, CO and H_2, or the aldehyde may burn completely to CO_2 and H_2O. This process is termed hydroxylation because the first addition of oxygen to the hydrocarbon molecule gives the hydroxyl grouping (OH) characteristic of alcohols.

More than 90 per cent of the theoretical formaldehyde possible from the oxidation of ethane has been isolated, thus showing that the primary mechanism of hydrocarbon combustion is essentially as given above. When sufficient oxygen is present to transform all the hydrocarbon to formaldehyde, combustion may be completed without the formation of soot or free carbon. The minimum amount of oxygen required to burn unsaturated hydrocarbons of the olefine type, without soot or free carbon, is given by the following reaction

$$(CH_2)_n + \frac{n}{2} O_2 = nCO + nH_2$$

It is to be noted from the above equation that the products of combustion, with the minimum oxygen that will prevent soot, are CO and H_2. On the other hand, similar mixtures $\left(C_nH_{2n+2} + \frac{n}{2} O_2\right)$ of a saturated paraffin hydrocarbon and oxygen yield carbon, CO and CO_2, H_2O; to prevent the formation of soot with paraffin hydrocarbons, it is necessary to use at least the following ratio of oxygen to hydrocarbon:

$$C_nH_{2n+2} + \frac{n+1}{2} O_2 = nCO + nH_2 + H_2O$$

With high flame temperatures it was found that in mixtures of hydrocarbons, hydrogen, and CO (with insufficient oxygen for complete combustion) more oxygen combines with the hydrocarbons than with either hydrogen or CO. Hence, when a mixture of methane and electrolytic gas is exploded in the proportions $CH_4 + 2H_2 + O_2$, the reaction is:

$$CH_4 + O_2 + 2H_2 = CO + H_2 + H_2O + 2H_2$$

More than 95 per cent of the oxygen reacts with the methane and less than 5 per cent combines with the hydrogen originally in the mixture. The relative rates of combustion of hydrocarbons, hydrogen, and CO are also indicated by the composition of furnace gases at various distances above the top of a bituminous coal fire. A series of analyses of these combustible gases showed that practically all the methane was consumed before the gases had traveled 1 ft from the fuel bed, the hydrogen at the end of 5 ft, whereas some carbon monoxide remained unburned at 8 ft from the surface of the fuel bed. Hence the relative rates of combustion for these three fuels are: hydrocarbons, hydrogen, and carbon monoxide; i.e., hydrocarbons burn fastest and carbon monoxide most slowly.

The following reactions are given to illustrate the steps occurring in the combustion or hydroxylation of typical hydrocarbons. The oxygen required for each successive hydroxylation is not shown. At temperatures below the ignition point, the hydroxylation of methane is:

$$\begin{array}{ccccccc}
\mathrm{H} & & \mathrm{H} & & & & \mathrm{H} \\
| & & | & & & & | \\
\mathrm{H{-}C{-}H} & \rightarrow & \mathrm{H{-}C{-}OH} & \rightarrow & \mathrm{H_2O} & + & \mathrm{H{-}C{=}O} \\
| & & | & & & & \\
\mathrm{H} & & \mathrm{H} & & & & \\
\text{methane} & & \text{methyl alcohol} & & & & \text{formaldehyde}
\end{array}$$

Depending on the amount of oxygen present, the formaldehyde thus formed may do one of the following:

a. Break down thermally to CO and H_2,

$$CH_2O \rightarrow CO + H_2$$

b. React with oxygen to form CO and H_2O,

$$CH_2O + \tfrac{1}{2}O_2 = CO + H_2O$$

c. React with oxygen for complete combustion to CO_2 and H_2O,

$$CH_2O + O_2 = CO_2 + H_2O$$

The combustion of a chain hydrocarbon is similar, i.e., one of the C_nH_{2n+2} groups is oxidized in the same manner. The following sequence shows the mechanism for the slow combustion of ethane at temperatures below its ignition point:

$$\underset{\text{ethane}}{C_2H_6} \rightarrow H_3C\cdot CH_3 \rightarrow \underset{\text{ethyl alcohol}}{CH_3CH_2OH} \rightarrow CH_3\cdot CH(OH_2) \rightarrow \underset{\text{acetaldehyde}}{CH_3CHO} +$$

$$H_2O \rightarrow CO + H_2 + \underset{\text{formaldehyde}}{H\cdot CHO} \rightarrow \underset{\substack{\text{formic}\\ \text{acid}}}{HCO\cdot OH} \rightarrow \underset{\substack{\text{carbonic}\\ \text{acid}}}{CO(OH)_2} \rightarrow CO_2 + H_2O$$

The combustion of an unsaturated hydrocarbon is similar to the process for the saturated hydrocarbons. The hydroxylation of a typical unsaturated hydrocarbon, ethylene, is:

$$\underset{\text{ethylene}}{C_2H_4} \rightarrow H_2C\cdot CH_2 \rightarrow \underset{\text{vinyl alcohol}}{H_2C\cdot CHOH} \rightarrow HO\cdot HC\cdot CH\cdot OH \rightarrow \underset{\text{formaldehyde}}{2CH_2O}$$

As in the combustion of methane shown above, the formaldehyde may decompose thermally to CO and H_2, or it may burn to CO_2 and H_2O.

Considering the petroleum products in order, natural gas, gasoline, and typical fuel oils, the hydrocarbons present are in each case increasingly heavier and of a more complex structure. In the higher hydrocarbons, in which R represents long-chain hydrocarbon radicals, the mechanism may be:

a. $RCH_2 \cdot CH_3 \rightarrow RCH_2 \cdot CH_2OH \rightarrow RCH_2CHO + H_2O$

$RCH_2 \cdot CHO \rightarrow R \cdot CHOH \cdot CHO \rightarrow RCHO + CH_2O \rightarrow RCHO + H_2 + CO$

b. $RCH_2 \cdot CH_3 \rightarrow RCH_2CH_2OH \rightarrow H_2O + \underset{\text{unsaturated}}{(R \cdot CH = CH_2)}$

The heavier hydrocarbons usually undergo reactions other than simple hydroxylation, and the same is true of the lighter hydrocarbons if conditions are unfavorable for the formation of hydroxylated compounds. The heavy hydrocarbons may be cracked to give saturated and unsaturated lighter hydrocarbons, or may decompose completely into carbon and hydrogen. The light saturated hydrocarbons are cracked more slowly than the heavier hydrocarbons. Thus methane when heated at about 2400 F is not completely decomposed at the end of an hour according to the reaction

$$CH_4 = C + 2H_2$$

Ethane, C_2H_6, the next hydrocarbon above methane in the paraffin series, is less stable, however, and can decompose very rapidly by heating in a tube at about 2100 F.

Concerning the speed at which the still heavier hydrocarbons decompose to carbon and hydrogen, it has been found in practice that the rate doubles for each 20 to 90 F rise in temperature, and that at the end of 1 min. at 1650 F about 40 per cent of the hydrocarbon decomposes to form carbon. With these data as a basis, it has been estimated that only thirty-six thousandths of a second is required for complete decomposition of heavy hydrocarbons at 2730 F, a reasonable temperature for the actual combustion of the various fuels.

Thus, the combustion of hydrocarbons does not occur directly on collision of fuel and oxygen molecules; these molecules are rather inert toward each other. Only the presence of certain intermediate substances, usually in amounts so small as to escape detection by analytical methods, makes the combustion of the fuel possible. These intermediate substances are unstable and highly reactive. It has been established by investigations that these intermediates are atoms and radicals. The latter are simply molecules that have been deprived of one or more atoms. Their reactivity is due to their ability to add oxygen molecules and to remove atoms from other molecules.

If a fuel molecule becomes a radical by losing a hydrogen atom it will readily react with a molecule of oxygen, forming a new radical capable of depriving another fuel molecule of a hydrogen atom. In this way the formation of a single radical gives rise to a large series of independent reactions—a so-called reaction chain. This chain would go on indefinitely were it not for certain other reactions which destroy radicals. Sometimes the conditions are such that the reaction between a radical and oxygen produces more than one new radical. In this way additional chains of reactions are set up. If this occurs sufficiently often the fuel is consumed so rapidly that the reaction becomes explosive. The heat liberated during the reaction is also a factor that determines the explosion characteristics of the fuel.

In the usual combustion of hydrocarbons, there is a race between thermal decomposition and hydroxylation. If the conditions favor hydroxylation, such as the use of a preheated hydrocarbon-air mixture and the allowance of sufficient time for the entrance of oxygen into the hydrocarbon molecule, there would be no soot. However, if conditions favor cracking, as for example, whenever the hydrocarbons and oxygen of the air are not thoroughly mixed, the heat from the combustion of part of the hydrocarbon cracks the remainder.

The problem of ascertaining the nature of the active intermediates is one of the most complicated in the field of the kinetics of gas reactions. Experimental evidence has been used to determine these radicals and the most probable sequence of the chain reactions. The initiation of these reactions depends upon the presence of reaction centers, hot or actuated molecules at an energy level above the mass average, from which combustion spreads chainwise. A consideration of the absorption of energy in quanta and its correlation with the number of molecules reacting is necessary to account for the fact that combustion reactions do not follow the laws of mass action, as deduced from simple stoichiometric equations. The success of some recent analyses to explain the large number of experimental facts justifies the hope that the technology of combustion may be advanced from its present empirical state to a scientific basis.

21. Combustion Temperatures. In an Otto-cycle engine, combustion occurs at nearly constant volume and the whole of the chemical energy stored in the mixture, with the exception of that lost to the walls of the cylinder, will be devoted to increasing its internal energy. However, the theoretical method of finding the temperature rise by dividing the heat output by the specific heat at normal temperatures gives results that are far too high, in fact nearly double those that are actually attained. This discrepancy is due to: the variation of the specific heat of the cylinder gases, which increases with increase in temperature, see Art. 33; the dissociation of the products of combustion, carbon dioxide and water (with attendant absorption of heat at high temperatures) into carbon monoxide and oxygen, and hydrogen and oxygen, respectively, see Art. 33; and an appreciable loss of heat by radiation and conduction to the walls of the combustion chamber. The weakening of the mixture due to dilution with residual exhaust products and the heat put into the mixture by compression are also factors in the actual engine which modify the temperature attained by the combustion of a mixture of any given energy content.

The line-reversal method of temperature measurement has been adapted to determine the flame temperature in an internal-combustion engine.[2] When gasoline and moderate compression pressures are used, temperatures have been measured over a sufficient portion of the operating cycle to determine the maximum flame temperature during combustion for varying air-fuel ratios within the combustion limits. The flame temperature found from a line-reversal measurement is characteristic of thermal equilibrium, established throughout the gases in the cylinder early in the process of burning and maintained during the subsequent expansion. The maximum flame temperatures for a compression ratio of 3.86 to 1 have been found to vary from 3750 F abs, with the richest and leanest air-fuel mixtures, to 4450 F abs with an air-fuel ratio of 13.9 to 1. The maximum flame temperatures observed and the corresponding temperatures calculated were found to be in closest agreement for the normal operating range of air-fuel ratios between 12 to 1 and 14 to 1. Throughout this range the calculated values are approximately 600 degrees higher than the observed values. With either richer or leaner mixtures the difference increases, reaching a maximum of 1000 degrees at the lean combustion limit.

Some of the discrepancy between the calculated and the observed temperatures by line-reversal measurement is undoubtedly due to the inability or difficulty to measure the following: the heat loss and work done up to the position of maximum temperature, the effect of NaOH solution, the effect of humidity of the air, the effect of the finite shutter opening, the effect of the fogging of cylinder window, and the effect of chemical equilibrium conditions. These experimental errors were considered by Hershey and Paton and the upper limits estimated. After increasing the observed values by these amounts they were still from 200 to 600 F below the calculated temperatures. It would therefore seem probable that this remaining difference is due to an inadequate analysis of the actual combustion process. The concentrations of the gases in an engine cylinder may be calculated when the temperature and pressure are known, hence the measurement of temperature independent of the other thermodynamic variables makes it possible to study the progress of the chemical reactions and equilibrium at high temperatures.

Complete cyclic measurements of manifold and cylinder pressures

[2] "Flame Temperatures in an Internal-Combustion Engine Measured by Spectral Line Reversal," A. E. Hershey and R. F. Paton, *University of Illinois Engineering Experiment Station Bulletin* 262, 1933.

during charging of the cylinder, of cylinder pressures with and without combustion, and the flame temperatures were made with the engine used in the above study for an air-fuel ratio of 13.7 to 1 and the results plotted in Fig. 11.[3] The temperature seems to reach a maximum value before the pressure, but during the latter part of the cycle the pressure and temperature curves parallel each other quite closely. The reader should observe the low manifold and cylinder pressures existing during the suction stroke, 360 to 240 degrees of crank angle; compare these curves with the discussion under manifolding in Art. 19.

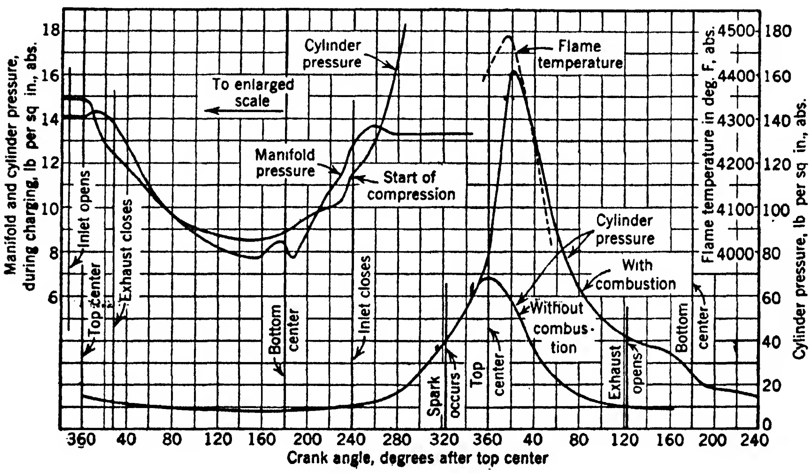


FIG. 11. Flame temperature; also manifold and cylinder pressures. Air-fuel ratio 13.7 to 1.

In a mixture of gases in which chemical action is possible, for example in a mixture of CO, O_2, H_2, N_2, H_2O, reactions may proceed with an accompanying change of temperature; but at some definite state of the mixture, a state determined by the temperature and the partial pressures of the constituents of the mixture, chemical action ceases, and the mixture is said to be in chemical equilibrium. According to the law of chemical equilibrium the mixture at the point of maximum pressure must contain a considerable amount of uncombined CO and a smaller amount of uncombined H_2. Consequently, at this point the whole of the heat of combustion based on complete reaction has not been developed. The discrepancy between the actual and calculated temperatures is due largely to the dissociation of the combustion products.

22. Dissociation. Dissociation may be defined as the breaking up of the molecules of a gas into simpler constituents. At any temperature

[3] *Ibid.*, page 31.

for a given gas, however homogeneous, the molecules at a given instant have different rates of activity. Some of the molecular activity may be so violent as to break up compound molecules into parts which later will re-combine with other parts. The probability of such disruptive activity is obviously greater the higher the temperature of the gas. In a hot gas in equilibrium, a process of dissociation and re-combination goes on continuously, to an extent depending on the temperature, with the result that at any instant a certain proportion of the gas is in the dissociated state. The proportion dissociated depends also on the pressure; at high pressure it is less than at low pressure, for the same temperature. According to Table X, page 94, the amount of H_2O dissociated, under a pressure of 147 lb per sq in., is 0.27 per cent at a temperature of 3630 F; for H_2O at the same temperature, the proportion dissociated at a pressure of 1470 lb per sq in. is 0.125 per cent. When an excess of air is provided, the influence is still smaller.

If the figures in Table X can be accepted as applicable to the conditions of a gaseous mixture after explosion in the cylinder of an internal-combustion engine, conditions which are not those of chemical equilibrium, it appears that dissociation plays no considerable part in that action. So far as it has any effect it reduces, very slightly, the chemical contraction, by substituting some molecules of H_2 and O_2 for molecules of H_2O and some molecules of CO and O_2 for molecules of CO_2. For the same reason it reduces slightly the immediate development of thermal energy, leaving a small proportion of the available chemical energy of the gaseous fuel to be developed later, as the proportion of dissociated molecules diminishes with falling temperature. The effect is therefore equivalent to a continued combustion or "after-burning." Or, if the whole thermal energy was regarded as being developed at once, and then a small portion of it as being absorbed by the breaking up of some of the molecules in consequence of their activity, the effect of dissociation would be indistinguishable from that of increased specific heat. Undoubtedly, dissociation places a barrier in the way of combustion at high temperatures and prevents the attainment of these temperatures, by combustion, which would otherwise be possible.

23. Combustion, Explosion, and Flame Propagation. Assume a homogeneous explosive air-gas mixture to be ignited at any point. A flame spreads in all directions from the point of ignition, traveling at a rate which depends on a number of factors, so that each portion of the mixture ignites in turn, the most distant portions igniting last. When the initial pressure is atmospheric, the velocity of flame propagation may be only 5 ft per sec, even in a rich air-gas mixture. In a large gas engine the available time for good explosion is about 1/10 sec and the maximum distance the flame must travel is about $1\frac{1}{2}$ ft; this gives a mean velocity of 15 ft per sec. In an automobile engine operating at 3000 rpm, assuming the time of explosion as about 1/6 revolution, the time of explosion is about 1/300 sec; if the flame travels 2 in. the average rate of flame propagation would be 50 ft per sec. The term explosion is generally used where the velocity of such flame propagation is high, but there is no definite line of demarcation between explosion and slow burning. The velocity of flame propagation in an explosive mixture depends on the kind of fuel, strength of the mixture, pressure, temperature, volume or mass of the mixture, dimensions of the containing vessel, method of ignition, amount of inert gases present, turbulence, quality and duration of ignition, and several other factors.

The portion of the air-gas mixture which is first ignited, adjacent to the point of ignition, burns at nearly constant pressure as it is surrounded by a large elastic cushion of unignited gas. Combustion is virtually completed before the pressure rises; the spread of the flame then brings more of the gas into action, the pressure rises, and the portion which was first burned is compressed. This compression is nearly adiabatic; its effect is to increase the temperature of the first burned portion much above its original combustion temperature and above the temperature reached in combustion by the outlying parts of the mixture. The effect of this continuous increase in pressure and temperature of the burned portion is to increase the rate of flame propagation to the unburned parts of the mixture. If the velocity of flame propagation is continually accelerated during the combustion of an air-gas mixture, the process is called *explosion*. When gas is burned with a Bunsen burner the velocity of flame propagation remains constant and combustion is not explosive. The flame is stationary, but as the gas is moving the flame is also moving relative to the gas, in the opposite direction and with the same velocity.

Experiments on explosions in closed vessels and actual engines have shown that the form of the combustion chamber of an internal-combustion engine is one of the most important features of design to control both the power output and the efficiency of the engine. Some of the important considerations in the design of a combustion chamber are: provision for free and unobstructed entry of the air and fuel, turbulence of the air and fuel, location of the point of ignition, and avoidance of pockets where the gases may become quiescent.

In explosion experiments with weak mixtures the velocity of flame propagation is comparatively slow, so slow that it is largely affected by convection currents set up by the ignition of the mixture nearest the ignition point. The gas in the upper part of the combustion chamber may be completely ignited while the gas in the lower part of the chamber is still unburnt. By stirring the contents of the chamber, so that the gases are in motion when ignition occurs, a much more rapid combustion of the whole charge can be obtained.

24. Turbulence. The maintenance of the turbulence set up by the gases during their entry into the combustion chamber is a very important design consideration, for upon turbulence depends the rate at which combustion takes place. If the combustible mixture were completely quiescent at the time of ignition, the flame propagation would be so slow that, even in a slow-speed engine, scarcely half the fuel would be burned before the exhaust valve opened. Regardless of what fuel is used, we must rely almost entirely on turbulence or mechanical disturbance to propagate the flame rapidly throughout the combustible mixture. Turbulence becomes more important as the density of the charge is reduced by throttling, for as the density is reduced the proportion of residual exhaust products is increased; and these, being diluents, tend to decrease the subsequent flame temperature and thereby retard the rate of flame propagation.

The effect of turbulence, in promoting rapid flame propagation of an air-gas mixture, is not so great with rich mixtures as with weak mixtures. When a fresh charge is drawn into a combustion chamber and compressed the gases will be in more or less violent motion at the time of ignition, because the shape of the combustion chamber and the velocity of the entering gases cause turbulent conditions which cannot be quieted by the time explosion begins. The great advantage of this is that combustion is more rapidly propagated throughout the charge and the maximum pressure occurs early in the expansion stroke. The flame propagation is not spherical, as in a quiescent mixture, but is by currents and eddies of burning gas which carry the flame to all parts of the combustion chamber more rapidly than is possible with spherical propagation. After turbulence in a gas-engine cylinder is quieted, explosion takes about three times as long as under actual operating conditions. Also, if the explosive charge in a gas engine were not fired after the first compression but were fired after several successive compressions, so that the turbulence set up on its entry had time in part to subside, the process of combustion would be prolonged. In a high-speed engine the whole expansion stroke may take only 1/25 sec. or less, and the explosion is completed in a small fraction of that time. This would be impossible were it not for the effect of turbulence in causing the flame to spread quickly throughout the mixture.

In any internal-combustion engine cylinder there is always a lay[illegible] of the working fluid adhering to the cold cylinder walls. This par[illegible] the fluid, because of its close proximity to cool surfaces, may lose he[illegible] rapidly as to escape complete combustion. Turbulence of the w[illegible] the fluid will scour away this layer and distribute it throughout the [illegible] in tion chamber. Undoubtedly the thickness of this layer and its influence upon the power output and efficiency of the engine depend largely upon the degree of turbulence within the combustion chamber.

Turbulence has no important influence upon the tendency of a combustible mixture to detonate. If the mixture were quiescent and ignited from any point, the flame would progress from that point at an increasing velocity until it reached a speed at which the unburnt gas, compressed by the rapidly approaching flame, received heat from compression, radiation, and conduction at a rate in excess of that transmitted to the cylinder walls, with the result that the temperature of the mixture would be raised above its self-ignition temperature. It would then ignite spontaneously and almost instantly throughout its whole mass, thus setting up an explosion or detonation wave. If the combustible mixture were in a state of violent disturbance at the time of ignition, the entire process would be speeded up very much; but it is the whole process that is so speeded up and not any one part of it, with the result that the two phenomena are precisely the same, though on a different scale so far as rate of flame propagation is concerned.

A combustion chamber that has a large surface in relation to its volume is generally supposed to owe its low efficiency to a large direct heat loss to the cylinder walls but it is far more probable that this form of combustion chamber owes its low efficiency to the larger area of the layer of gases which clings to the walls and thus escapes combustion; the thickness of the layer also is greater because there is more resistance to the motion of the gases and the scouring effect of turbulence is less. Considering the surface-volume ratio of combustion chambers as a function of cylinder bore, it appears that high-speed Diesel engines are in a class separate and distinct from the larger Diesel engines from which they were developed. With a cylinder diameter of 25 in. and the piston at dead center position the combustion-chamber surface is 13 sq ft per lb of compressed air; for a 4-in. diameter high-speed Diesel the surface may exceed 100 sq ft per lb of chamber contents. For some years the Diesel engine has been proclaimed as the most advanced type of internal-combustion engine, but the world's principal demand for automotive prime movers is still being supplied by the Otto-cycle type of engine because the latter is more flexible and responds more readily to the acceleration and speed demands of most users. The large ratio of surface to volume in the small Diesel engine required for automotive work pr held largely responsible for its failure to assert itself in this field. Con-
conerable progress has been made in the direction of overcoming the
ignitil chamber handicap by subdividing it and making it function as a
may becumulator rather than as a heat dissipater. The success already
is still ul with high-speed Diesel engines of the precombustion-chamber

type justifies the expectation that the subdivided chamber will replace the simple-chamber small-bore engines, regardless of results attained with the simple chambers of the large-bore engines.

25. Detonation and Knock Testing. The highest compression ratio consistent with no pre-ignition of the charge should be used in order to obtain high thermal efficiency; see Tables VII and VIII. There is, however, one characteristic of volatile fuels which makes serious modification of this statement necessary; and that is the tendency of these fuels to detonate. *Detonation* occurs when the nearly simultaneous explosion of all parts of an air-gas mixture is effected by means of a compression wave, as distinguished from an explosion in which ignition spreads from one part to another in a manner like that of slow combustion. Detonation is quite distinct from pre-ignition; pre-ignition is the firing of the charge at some point during the compression stroke *before* the point of ignition, but detonation follows ignition. It occurs through the development of a compression wave whose pressure heats the still unburnt portions of the charge to a temperature at which they suddenly combine, without waiting to be reached by the igniting flame. Thus detonation depends upon the flame temperature and rate of increase of pressure and temperature during combustion; there are, therefore, degrees of detonation, depending upon the relative rates of propagation and of temperature increment due to the compression wave. Other contributing factors are: strength of air-gas mixture, engine speed, location of point of ignition, time of ignition, and form of combustion chamber.

Detonation or " pinking " may be distinguished from normal explosion by its sharp, metallic, hammerlike ringing sound, produced when the high-pressure compression wave strikes and rebounds from the cylinder walls. Detonation not only is disagreeable to the ear but it also causes overheating, decreases the power developed, increases the fuel required, and increases the stresses and wear of the engine. Means of avoiding or reducing detonation are: keeping the compression ratio below a certain limit, depending upon the fuel to be used; enriching fuel-air mixtures; retarding the time of ignition; improving cooling; special designing of the combustion chamber; and, for engines with higher compression pressures, the using of anti-knock fuels, such as gasoline blended with benzol or with small amounts of anti-knock compounds. A definite knock-suppressing effect may be obtained by adding 2 to 7 cc of tetraethyl lead fluid, $Pb(C_2H_5)_4$, to one gallon of gasoline; the amount of tetraethyl lead added depends upon the quality of the gasoline and the effect desired.

The apparatus developed by Midgley and Boyd for measuring the relative intensity of detonation is the bouncing pin indicator shown in

Fig. 12. Essentially it consists of a slender hollow barrel having locked in place at the lower end a diaphragm which supports a long pin free to bounce and close electrical contact points C at the upper end. The normal burning of the fuel in the combustion chamber produces a rate of pressure rise which is not sufficient to cause a sudden deflection of the diaphragm, hence the bouncing pin remains in contact with the diaphragm. Detonation, however, so alters the characteristics of the combustion that the rate of pressure rise is tremendously increased and there is a consequent large and sudden deflection of the diaphragm

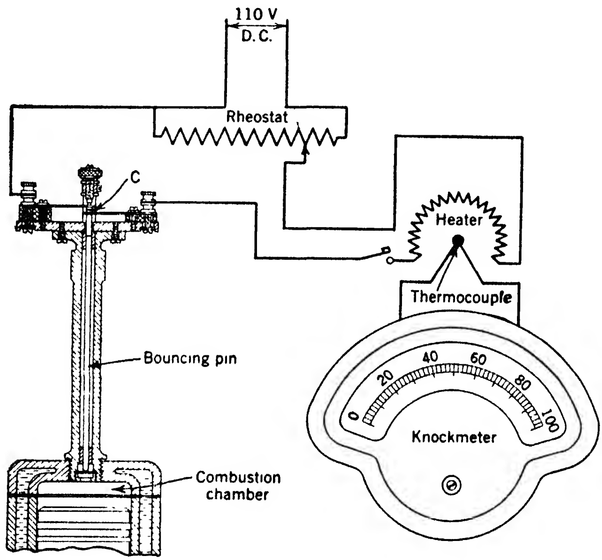


FIG. 12. Apparatus for measurement of detonation, bouncing pin circuit with indicating knockmeter (millivoltmeter).

which projects the bouncing pin upward to close the contact points and the electric circuit.

The direct-reading knockmeter in Fig. 12 has a heater element which is connected in series with the bouncing pin and the source of direct current. The intermittent current pulsations through the bouncing-pin circuit heat the resistance wire. A portion of the heat generated in the resistance wire is conducted to the hot junction of the thermocouple assembly; this heat raises the temperature of the hot junction and generates a current of electricity in the thermocouple circuit; the intensity of this current is indicated by the knockmeter (millivoltmeter). The " number of millivolts " indicated on the knockmeter is a relative measure of the detonation of the fuel. Detonation has been studied by the use of a stroboscope, also with a microphone and oscillograph to register engine sounds.

A more recent method of expressing the detonating property of a volatile fuel is by stating its *octane number*. This is determined by taking a highly detonating hydrocarbon, heptane, and adding to it sufficient iso-octane to give the mixture the same detonating property as the

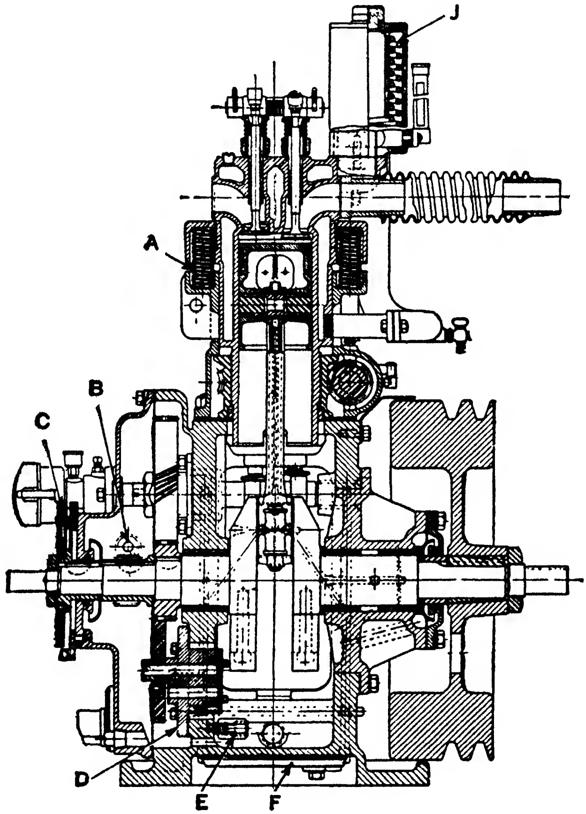


Fig. 13. Sectional view of Waukesha ASTM-CFR fuel testing engine. A, cylinder back-lash springs; B, tachometer drive shaft and worm; C, spark position indicator; D, gear-type oil pump; E, oil pressure relief valve; F, lube oil heater; J, condenser cooling coil.

volatile fuel. Iso-octane has very good anti-detonating properties and the percentage present in the equivalent octane-heptane mixture is the octane number of the volatile fuel under test.

The standard ASTM-CFR testing unit provides a means for rating gasoline by octane number through a comparison of its knock intensity with knock intensities for fuels of known octane rating. It is a one-cylinder, variable compression, overhead valve engine; see Fig. 13. A study of the factors which affect knock ratings (exclusive of the knock qualities of the fuel itself) shows them to include: engine speed, compression ratio, combustion chamber shape, mechanical uniformity, charge density, spark timing, mixture ratio, mixture temperature, jacket temperature, knock intensity, and calibration of the knock indicating instruments. In the ASTM-CFR engine, all of these factors can be controlled.

This standard test engine is belted to a special induction motor with constant, synchronous speed characteristics to maintain an engine speed of 900 rpm at all times, either idle or loaded. A three-tank, adjustable-level carburetor permits regulation of the mixture ratio and quick change from one fuel to another. Changes of the compression ratio may be made while the engine is operating by a hand crank turning a worm gear to raise and lower the cylinder as necessary. Ratios from 3 : 1 to 10 : 1 are possible, and they may be accurately measured by a micrometer attached directly to the cylinder. In order to maintain uniform operating conditions, mixture and jacket temperatures are fully controlled. Evaporative cooling is used so that the water jacket temperature is held at the boiling point of the coolant. A condenser, *J* in Fig. 13, cooled by city water and rigidly connected to the cylinder jacket, converts the vapor into a liquid and returns it to the reservoir of the cooling system. Fuel-mixture temperatures are controlled by a carbon-pile rheostat on the switchboard panel connected to a bayonet-type heater in the intake manifold.

As previously mentioned, octane ratings of fuels by the ASTM-CFR method are accomplished by comparing a sample fuel against fuels of known octane rating.[4] A preliminary knockmeter reading, Fig. 12, of the test fuel at a predetermined compression ratio enables the operator to compute an approximate octane rating of fair accuracy. The operator then prepares two reference fuels which have known octane ratings, one higher than the estimated rating of the test sample and the other lower. Then with the carburetor mixtures set for maximum knock and the engine compression ratio held constant, accurate knockmeter readings are taken of all three specimens. When the reference fuels give knockmeter readings that are above and below the reading given by the test specimen, simple interpolation permits an accurate computation of the test-fuel octane rating.

In gasoline engines, not only the power output but the entire general design depends primarily upon the octane number of the fuel; an engine designed to utilize 100-octane gasoline may develop twice the power output of one designed and operated on 70-octane gasoline. Gasoline

[4] "Tentative Methods of Test for Knock Characteristics of Motor Fuels," A.S.T.M. Designation: D 357–36 T. American Society of Testing Materials, 260 S. Broad Street, Philadelphia, Pennsylvania.

engines operate normally with a relatively low ratio of compression, hence on the steep (rapidly changing) part of the air-cycle thermal efficiency curve, see Fig. 19; any increase in octane number permits operation further up on the efficiency curve and allows additional supercharging. In Diesel engines, the ratio of compression is high, hence they operate on the nearly flat portion of the curve and the thermal efficiency is practically constant over a wide range, whereas supercharging is unlimited by the possibility of detonation. The limit is determined only by the allowable maximum cylinder pressure. If the Diesel engine will run at all it will give practically the same power output and efficiency on any fuel which it will burn; the kind of fuel is likely to influence such secondary factors as ease of starting and silence of operation.

26. Combustion in Diesel Engines. In Diesel engines, detonation frequently occurs because the fuel is not burned as rapidly as it is injected. The pressure under conditions of detonation may exceed 1000 lb per sq in. where the injection has been started very early. Ignition takes place from the surface of many liquid droplets traveling in definite directions, and it is desired to bring air to these droplets at as great, or greater, rate as that at which the air is being consumed by the burning fuel. This requires methodical movement of the air across the fuel stream; turbulence, as desired in the gasoline engine, is of little help because as much as possible of the air within the cylinder must pass across the stream of burning droplets. The desired air movement can best be obtained by setting the air within the combustion chamber into a unidirectional flow, more or less at right angles to the fuel flow.

Combustion in Diesel engines may be divided into three stages: first, a delay period during which the surface of each individual droplet of fuel is surrounding itself with an envelope of vapor, the outer surface of which must retain a temperature high enough to bring about self-ignition; second, a period of very rapid burning during which the many droplets, which have accumulated and become surrounded by air during the delay period, burst into flame in rapid succession; third, a period of controlled burning when, because of the high temperatures within the cylinder, the last droplets of fuel burst into flame almost as they enter the combustion chamber. During the third period there is almost complete control of the rate of burning, and therefore pressure, since these depend almost entirely upon the manner in which the fuel is admitted to the cylinder. Some authorities call the period of after-burning a fourth stage in Diesel combustion. The more unburned fuel that accumulates during the delay period the more violent will be the rate of pressure rise during the second period and the greater will be the Diesel knock. Also, if the fuel were fairly volatile with a relatively high self-ignition temperature, there would be the possibility of small pockets of vaporized combustible mixture being formed; this is undesirable as in the gasoline engine. To eliminate this "detonation" in Diesel engines, it is necessary to reduce to a minimum the delay in the ignition of a Diesel fuel; see discussion under Ignition Quality, page 33.

The ignition delay period in Diesel engines may be reduced by: increasing the ratio of compression; adding heat to the air during compression; reducing the size and increasing the dispersion of the fuel droplets during injection; using a fuel of high cetane value, see Art. 15, or low self-ignition temperature. Contrary to what happens in carburetor-type engines, an increase in compression materially reduces detonation, owing to the fact that the delay in ignition is reduced and the high temperatures in the cylinder counteract the effect of cooling at the time of injection. There is a limit to the increasing of the compression ratio and the adding of heat to the air during compression because of practical considerations; and beyond a certain point nothing is gained in thermal efficiency and there is an appreciable loss in power output. Reducing the size and increasing the dispersion of the droplets insure a more rapid combustion because of the increased concentration of oxygen surrounding the finely divided particles of fuel. In actual practice, however, the reduction in the size of the fuel droplets involves reduction of penetration, and in most combustion chambers maximum penetration is desired, more especially so if a high compression ratio is used. There are those who believe that in the Diesel engine, as well as in the gasoline engine, the fuel should be vaporized prior to combustion, this being facilitated by finer atomization. By using a fuel with a high cetane number it is possible to reduce ignition delay by increasing the differential between the air temperature and the fuel self-ignition temperature. Although it is desirable to reduce the delay period as much as possible, a delay period of sufficient duration is needed to allow the fuel to penetrate into the engine cylinder before combustion begins.

A fairly high cetane value of a Diesel fuel oil is advantageous because it shortens the delay period and thus allows better control over the rate of combustion and pressure rise. The cetane value of fuel oils depends primarily upon the origin, but can be varied by changing the distillation range or by adding amyl nitrite, ethyl nitrate, or certain peroxides. In general, any source which yields a low-octane gasoline will yield a high-cetane fuel oil, and vice versa. Within limits, slow volatility of a fuel oil is desired, because too volatile a fuel is likely to cause detonation and produce vapor troubles in the fuel injection system. If the volatility were too low, however, a further delay period would be introduced, because of the greater time required to form a gas envelope on the outside surface of the fuel droplets.

In order to reduce the delay period of fuel injection in Diesel engines many working cycles employing auxiliary chambers have been developed in recent years to modify the timing of the combustion of the fuel and to reduce the rate of pressure rise produced by "detonation." All of these auxiliary-chamber processes are being developed with the aim of initiating combustion in a chamber under conditions of deficiency of air, and keeping away from the main engine structure the high-peak pressures. In the precombustion-chamber process the fuel is injected into a chamber, ignited there, and then blown into the main cylinder; engines using this method of injection are shown in Figs. 69, 82, and 83. Experiments and the performance of actual engines clearly indicate the desirable action of the precombustion chamber, especially in high-speed fuel-injection engines. Another auxiliary-chamber design is the so-called air-storage process, see Fig. 87, in which a substantial part of the fuel is injected into the main chamber; subsequently the compressed air in the air-storage cavity flows into the cylinder for purposes of insuring slow and complete combustion of the fuel.

CHAPTER IV

THERMODYNAMICS OF INTERNAL-COMBUSTION CYCLES

In an internal-combustion engine the fuel that is used to supply the heat energy forms a part of the working substance; its combustion takes place in the cylinder where the gases do direct work by expansion. This essential principle is shared equally by the gas engine, the gasoline engine, and the oil engine.

27. Cycles of Combustion. There are two distinct methods by which the combustion of a fuel may be accomplished in an internal-combustion engine:

a. Air, intimately mixed with a suitable quantity of fuel in gaseous or vaporized form, is compressed to a moderate extent in the cylinder, and is then ignited by an electric spark, the resultant rise of pressure being practically instantaneous, with *nearly* constant volume.

b. Air only is compressed in the cylinder, the compression being carried to a high pressure so that the resultant temperature is sufficient to ignite the fuel. At the instant this high compression pressure is attained, the fuel is sprayed into the cylinder, the piston at the same time beginning to move downward on the power stroke. Subsequent combustion of the fuel tends to produce a further increase of temperature and so maintains a *nearly* constant pressure during the first portion of the downward stroke of the piston. The period of fuel injection can be prolonged, and the pressure maintained nearly constant, until the available oxygen has been consumed.

The essential difference between *a* and *b* will be at once apparent. Engines of class *a*, which include all gas and gasoline engines besides some oil engines, get their power impulse from a sudden rise of pressure, so rapid that the engines are called *explosion, constant-volume,* or *Otto-cycle engines.* Engines of class *b* depend upon the maintenance of nearly constant pressure during the period of fuel injection and consequent combustion. This type is practically confined to heavy fuel oils; the engines are commonly known as *constant-pressure* or *Diesel engines.*

28. Cycles of Operation. The process of introducing the working substance into the cylinder, compressing, igniting, and driving it out of the cylinder after having done work, will now be considered. This cycle of events may be accomplished in one of two ways: by the four-stroke cycle or by the two-stroke cycle. Other cycles have been attempted, but none have met with any measure of success.

Four-stroke Cycle. In the most usual type of internal-combustion engine, the cycle of operation (or mechanical cycle) is completed in four strokes of the piston or two revolutions of the crankshaft. (1) The charge of gas and air may be drawn into the cylinder, *A* in Fig. 14, the intake valve being open on the downward (suction) stroke of the piston. (2) At the beginning of the return (compression) stroke, *B* in

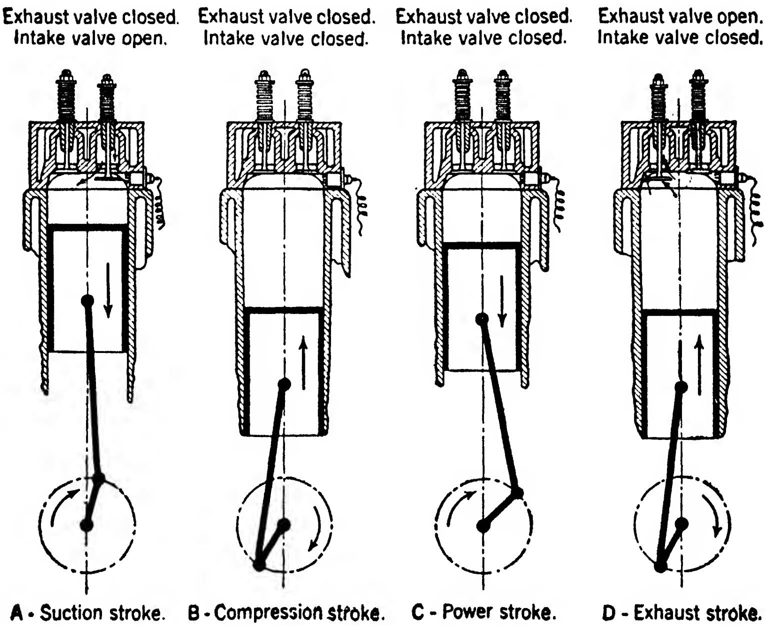


FIG. 14. Cycle of operation of an Otto four-stroke cycle engine.

Fig. 14, intake and exhaust valves being closed, the mixture of air and gas is compressed into the clearance space, which may be from one-fourth to one-eighth of the original volume. Near the top dead center the compressed mixture is ignited by an electric spark, heating the charge to a very high temperature and consequently raising the pressure. (3) During the next downward stroke the fired mixture expands, *C* in Fig. 14, doing work on the piston which, through the connecting rod and crank, creates a turning effort of the crankshaft, as shown. (4) As the piston moves upward on the fourth stroke, the products of combustion are forced out of the cylinder, *D* in Fig. 14, the exhaust valve being open. A small quantity of the burnt mixture remains in the clearance space and is mixed with the next charge of fuel, unless special means are used to remove these inert and non-combustible gases by scavenging.[1] After this cycle the inlet valve is again opened, and the cycle of operations is repeated. In the four-stroke cycle there is only one power stroke out of every four, which seems retrogressive when compared with steam-engine practice; nevertheless, the four-stroke cycle has entirely justified itself.

The indicator diagram of the four-stroke cycle is shown in Fig. 15. During the suction stroke 0–1, the pressure is less than atmospheric; and during exhaust 4–0, the pressure is above atmospheric. The area of the loop 0–1–5 represents the work done *on* the gases during the cycle and is due to the "pumping action" of the piston in forcing out the exhaust gases (high pressure at 4) and drawing in the charge during suction (low pressure along 0–1). The curve 1–5–2 represents compression of the charge, V_2 being the volume of the clearance space into which the charge is compressed. Line 2–3 is the rise of pressure caused by the explosion, and 3–4 is the expansion which constitutes the effective working stroke. The area 2–3–4–5 represents work done *by* the gases during the cycle, and the net work developed during the four-stroke cycle is, therefore, (2–3–4–5) minus (0–1–5).

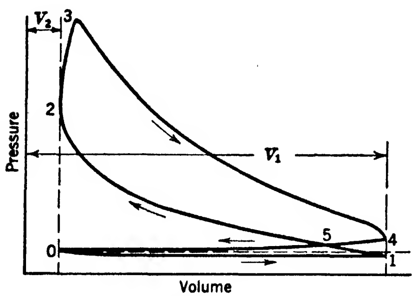


FIG. 15. Indicator diagram of the Otto four-stroke cycle.

It is to be noted that the four-stroke principle can be used for engines operating on the Diesel cycle, as well as for Otto-cycle engines (as described above).

Two-stroke Cycle. Figure 16 illustrates the action of an internal-combustion engine in which one cycle of operation is completed in two strokes of the piston. At *A* the scavenging valves in the cylinder head are open, and a charge of pure air (compressed by an air compressor) blows into the cylinder, forcing out the exhaust gases as shown. The scavenging valves close, and as the piston moves upward the exhaust ports are covered, as at *B*. Continued upward motion compresses the air charge until upper dead center is reached. The clearance volume is small, and the maximum or final compression pressure rises to about 500 lb per sq in. The work done on the air charge during compression causes the temperature to rise to 1000 F or more. When the piston

[1] *Scavenging* is generally considered as any action in which the products of combustion are forced out of the cylinder of an internal-combustion engine with the admission of fresh air.

reaches top dead center, a charge of fuel is injected into the cylinder, and combustion occurs as the piston starts downward, as at B. The piston continues to the end of its stroke under the influence of the

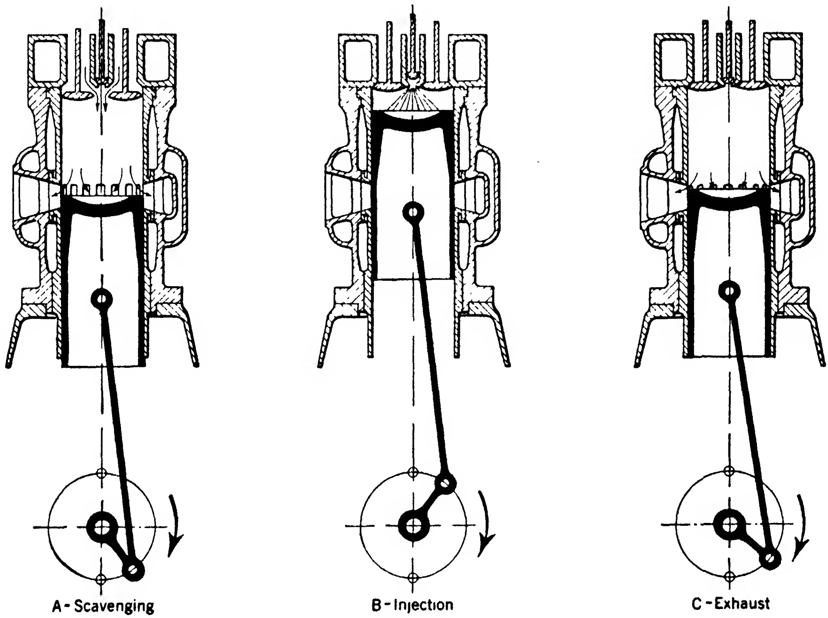


FIG. 16. Cycle of operation of a Diesel two-stroke cycle engine.

expanding gases. Before the completion of the downward stroke, as at C, the piston uncovers passages or ports in the side of the cylinder through which the exhaust gases pass. By this means one explosion is obtained every revolution of the engine shaft when using one single-acting cylinder.

An indicator diagram of a two-stroke cycle engine is shown in Fig. 17. A charge of pure air is admitted under pressure along 0–1; compression of this air occurs along 1–2 with a resultant high pressure and temperature. Injection of fuel and combustion at nearly constant pressure are represented by line 2–3. Expansion of the gases takes place along 3–4. From 4 to 0, exhaust occurs and scavenging of the burnt gases in the cylinder may be effected.

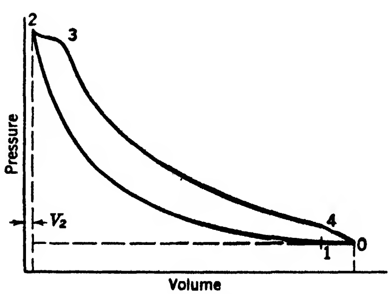


FIG. 17. Indicator diagram of the Diesel two-stroke cycle.

Although the two-stroke cycle has been described for an engine operating on the Diesel principle, the two-stroke cycle is also applicable to Otto-cycle engines.

29. Ideal Action with Combustion at Constant Volume. In any real engine the cycles of operation are complicated by exchanges of heat, through conduction and radiation, between the working gas and the walls of the cylinder, and also by the fact that the process of explosive combustion of the charge is not instantaneous, but takes an appreciable time to be completed. It is convenient, however, to consider an ideal action in which (1) there is no exchange of heat between the gas and the walls, and (2) all the heat of combustion is generated at a particular instant, namely when the volume is constant at the end of the compression stroke (before expansion begins). Such an ideal action affords a useful standard for comparison with the performance of an actual engine.

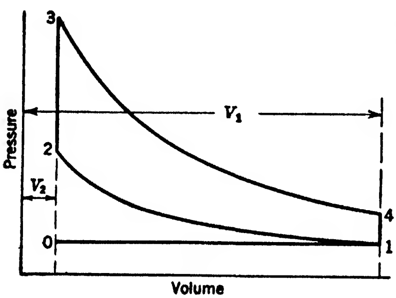


FIG. 18. Indicator diagram of the ideal Otto cycle.

Consider an ideal engine in which there is no transfer of heat between the working substance and the metal, and in which the combustion occurs only when the piston is at top dead center. The indicator diagram of this ideal engine, working on the four-stroke cycle, is shown in Fig. 18. The distance V_2 is the clearance volume, which is the volume occupied by the mixture during combustion, and 0–1 is the volume (piston displacement) swept through by the piston. The line 0–1 represents admission at atmospheric pressure; 1–2 is the compression, which by assumption is adiabatic; 2–3 is the rise of pressure caused by the explosion; 3–4 is the expansion, also adiabatic, which constitutes the working stroke. At 4 the exhaust valve opens, with the result that part of the gas at once escapes and the pressure falls to that of the atmosphere. Line 1–0 represents the exhaust stroke by which the cylinder is (except for the clearance) emptied of combustible products preparatory to receiving a fresh charge in the first stroke of the next cycle.

If the engine were operating on the two-stroke cycle, the lines 0–1 (and 1–0) would be omitted from the indicator diagram in Fig. 18, which would then consist simply of the figure 1–2–3–4–1. From 2 to 3 (in either cycle) the whole energy due to the chemical reaction goes to heat the mixture, for by hypothesis none is lost to the walls. The increase of pressure from 2 to 3 can be readily calculated when the rise of temperature resulting from this accession of heat is known, provided we also know the change in specific volume due to the change in chemical constitution brought about by the explosion. With the mixture used in gas engines there is very little change in specific volume; that is, if the products of combustion were brought to the same pressure and temperature they would occupy very nearly the same volume as before combustion took place. The specific volume of a mixture of illuminating gas and air is reduced after explosion by about 3 per cent; with some explosive vapors the specific volume is slightly increased. The changes being usually small, it is convenient in considering an ideal engine to ignore them, and to treat the working substance as if it were a gas whose specific volume is constant.

The largest constituent of the charge is air, and nitrogen is the largest constituent of the products of combustion; the specific heat of nitrogen is, for equal volumes, the same as that of air. Hence for the purpose of obtaining a simple standard with which actual engines may be compared, a practice has been adopted of treating the working substance as if it were air, to which between 2 and 3 there is imparted a definite quantity of heat. The quantity of heat may be calculated when the composition of the explosive mixture and the heats of combustion of its various constituents are known. Having the temperature at 2, we could calculate the rise of temperature and consequently the pressure at 3, if the average specific heat (at constant volume) were known between 2 and 3.

30. Ideal Otto Cycle with Constant Specific Heat. Air Standard. In the study of the ideal cycle, as shown in Fig. 18, the working substance is supposed to have a constant specific heat and the action is based on what is generally called the *air standard* for comparison with the performance of actual internal-combustion engines. The assumptions made are that there is no transfer of heat between gas and metal, that there is instantaneous complete combustion, that there is no change of specific volume, and that the specific heat is constant.

This last assumption is by no means true of air, and is even less applicable to the mixed gases in the cylinder of a gas engine. It is known that the specific heat increases with the rise of temperature to an extent which greatly affects the action of an engine. The specific heat of the air-gas mixture is much greater at high temperatures than at low temperatures; hence the "air standard" is an unreasonably high criterion to apply to any actual performance. The efficiency of a real engine must be less than this standard, not only because of such more or less avoidable losses as occur through radiation and conduction of heat to the cylinder walls, but also because the standard assumes, on the part of the working substance, an essential quality that is widely different from the quality of the real gases of which it is composed. Even if there were no loss of heat, the limit of temperature which the gases reach after explosion must be much lower than that which would be reached if the specific heat were constant. However much the heat losses are minimized, the hypothesis of constant specific heat and no dissociation makes the air standard an impossible ideal.

It is instructive, nevertheless, to study the air standard, with constant specific heat, as a means of examining some of the effects that follow from varying the working conditions. For instance, the air standard may be used to show how the thermal efficiency of the gas-engine cycle is improved by increasing the compression.

Let T_1 be the absolute temperature at which the air-gas mixture is taken in, T_2 the temperature to which it is compressed, T_3 the temperature after explosion, and T_4 the temperature after expansion. Figure 18 shows the cycle, with the stages numbered to correspond with these subscripts. The lines 1–2 and 3–4, Fig. 18, are assumed to be adiabatics; all the heat must therefore be supplied along line 2–3 and rejected along 4–1. If c_v, the specific heat at constant volume, is treated as constant for the purposes of this calculation, the *heat supplied* along 2–3 is

$$ {}_2H_3 = Mc_v(T_3 - T_2) = \textit{input} \tag{9} $$

in which M = weight of the charge, pounds.
c_v = specific heat at constant volume, 0.173 for air.
T_3 = absolute (F) temperature at 3.
T_2 = absolute (F) temperature at 2.

The *heat rejected* along 4–1 is

$$ {}_4H_1 = Mc_v(T_4 - T_1) \tag{10} $$

in which T_4 = absolute (F) temperature at 4.
T_1 = absolute (F) temperature at 1.

The net work is

$$ {}_2H_3 - {}_4H_1 = Mc_v(T_3 - T_2) - Mc_v(T_4 - T_1) = \textit{output} $$

The air-standard thermal efficiency of the Otto cycle is

$$ \frac{\textit{output}}{\textit{input}} = \frac{{}_2H_3 - {}_4H_1}{{}_2H_3} = \frac{Mc_v(T_3 - T_2) - Mc_v(T_4 - T_1)}{Mc_v(T_3 - T_2)} $$

$$ = \frac{(T_3 - T_2) - (T_4 - T_1)}{T_3 - T_2} $$

$$ e_a = 1 - \frac{T_4 - T_1}{T_3 - T_2} \tag{11} $$

From the laws of perfect gases

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2} = \frac{P_3V_3}{T_3} = \frac{P_4V_4}{T_4}$$

also

$$P_1V_1^n = P_2V_2^n \quad \text{and} \quad P_3V_3^n = P_4V_4^n$$

Combining the above equations, $$\frac{T_1}{T_2} = \left(\frac{P_1}{P_2}\right)^{\frac{n-1}{n}} \tag{12}$$

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{n-1} \tag{13}$$

and

$$\frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{n-1} \tag{14}$$

But $V_3 = V_2$ and $V_4 = V_1$, as shown in Fig. 18. Hence

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{n-1} \quad \text{and} \quad \frac{T_4}{T_3} = \left(\frac{V_2}{V_1}\right)^{n-1}$$

therefore

$$\frac{T_1}{T_2} = \frac{T_4}{T_3} \quad \text{or} \quad \frac{T_4}{T_1} = \frac{T_3}{T_2}$$

Then

$$\frac{T_4}{T_1} - 1 = \frac{T_3}{T_2} - 1 \quad \text{and} \quad \frac{T_4 - T_1}{T_1} = \frac{T_3 - T_2}{T_2}$$

or

$$\frac{T_4 - T_1}{T_3 - T_2} = \frac{T_1}{T_2} \tag{15}$$

Substituting equation 15 in equation 11,

$$e_a = 1 - \frac{T_1}{T_2} \tag{16}$$

Thus from equation 12,

$$e_a = 1 - \left(\frac{P_1}{P_2}\right)^{\frac{n-1}{n}} \tag{17}$$

And from equation 13,

$$e_a = 1 - \left(\frac{V_2}{V_1}\right)^{n-1}. \tag{18}$$

Equation 16 shows that the air-standard thermal efficiency of a gas engine employing the Otto cycle depends upon the ratio of the absolute temperatures before and after compression. From equation 18, the smaller the clearance, V_2, of the engine, the larger is the efficiency. Equation 17 is important as showing the beneficial influence of compression, the efficiency increasing as P_2 increases, since the pressure P_1 remains nearly constant.

Example. In an engine operating on the ideal (air-standard) Otto cycle the suction pressure is 14.5 lb per sq in. abs, temperature at the beginning of compression 100 F, and the highest pressure attainable (at the end of combustion) is 300 lb per sq in. abs at a temperature of 2000 F. Find: (*a*) the pressure at the end of compression, (*b*) the temperature at the end of compression, (*c*) the air-standard thermal efficiency, and (*d*) the clearance volume in per cent of the stroke. Assume that $n = k = 1.40$.

Solution. (*a*) From equation 12,

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

$$T_2 = 560\left(\frac{P_2}{14.5}\right)^{\frac{0.4}{1.4}} \qquad (A)$$

From the laws of perfect gases,

$$\frac{P_2V_2}{T_2} = \frac{P_3V_3}{T_3}$$

Since $V_2 = V_3$ (see Fig. 18),

$$T_2 = 2460\left(\frac{P_2}{300}\right) \qquad (B)$$

Putting T_2 of (A) in place of T_2 in (B)

$$560\left(\frac{P_2}{14.5}\right)^{2/7} = 2460\left(\frac{P_2}{300}\right)$$

Hence $P_2 = (31.77)^{1.4} = 127$ lb per sq in. abs.

(*b*) $T_2 = 2460\left(\frac{127}{300}\right) = 1041.4$ abs or 581.4 F.

(*c*) $e_a = 1 - \frac{T_1}{T_2} = 1 - \frac{560}{1041.4} = 1 - 0.538 = 0.462$ oı 46 2 per cent.

(*d*) Since $P_1V_1^n = P_2V_2^n$, $\frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^n$.

Take the piston displacement $= 1.00$; then $V_1 = V_2 + 1$.
Thus

$$\frac{127}{14.5} = \left(\frac{V_2 + 1}{V_2}\right)^{1.4}$$

and $V_2 = 0.27$ or 27 per cent of the stroke.

With increased compression ratios $\left(\frac{V_1}{V_2}\right)$, the air-standard thermal efficiency of the ideal Otto cycle increases as shown in Table VII, taking

TABLE VII

EFFECT OF COMPRESSION ON AIR-STANDARD THERMAL EFFICIENCY OF IDEAL OTTO CYCLE

Clearance Volume V_2 in per cent	Compression Ratio $\frac{V_1}{V_2}$	Air-standard Thermal Efficiency	P_1 in lb per sq in abs	P_2 in lb per sq in. abs	$\frac{P_2}{P_1}$
50.0	3	0.356	14.7	69	4.7
33.3	4	0.426	14.7	102	7.0
20.0	6	0.512	14.7	180	12.2
14.7	8	0.564	14.7	270	18.4
11.1	10	0.602	14.7	369	25.1
9.1	12	0.630	14.7	477	32.4
7.7	14	0.652	14.7	590	40.1
6.7	16	0.670	14.7	713	48.5
5.9	18	0.685	14.7	840	57.1

$n = k = 1.4$. Table VII shows that there is at first a rapid gain of thermal efficiency with increased compression, but as the compression ratio becomes higher the rate of increase of this thermodynamic advantage decreases; see also Fig. 19. The efficiencies actually obtained in tests of engines are considerably lower, usually 40 to 60 per cent less, owing to heat losses and to the combustion not being instantaneous, as well as to the fact that the specific heat is not constant but increases at high temperatures.

With all engines of this class there is a practical limit to the amount of compression. The pressure must not be so great as to cause preignition by unduly raising the temperature of the mixture during the compression stroke, nor so great as to give to the explosion, when it does occur, the peculiar characteristic known as detonation.[2] The compression pressures for engines operating on the Otto cycle vary from 50 to 250 lb per sq in., differing with various kinds of fuels; for example, being low when there is much hydrogen and comparatively high for alcohol. For gasoline automobile engines, the pressures range from 80 to 180 lb per sq in. Difficulty may be experienced with automobile engines designed for the higher compression pressures when operated with ordinary gasoline. With producer gas as a fuel, compression pressures of 100 to 160 lb per sq in. are allowable; and with blast-furnace gas, pressures as high as 200 lb per sq in. may be used. When air alone is compressed, as in the Diesel engine, and the fuel is injected only when combustion is intended to occur, the same considerations do not hold; compression may then be usefully carried much higher.

[2] *Detonation* is discussed on pages 65 to 69.

Comparison of Figs. 15 and 18 shows the variation of the actual Otto cycle from the theoretical. In the actual engine, combustion is not instantaneous but begins before the piston reaches the end of the compression stroke, at 2 in Fig. 15. The exhaust valve is generally opened before the end of the expansion stroke, in order that the pressure in the cylinder may be practically atmospheric before the piston starts on the exhaust stroke. The exhaust stroke, Fig. 15, is at a pressure slightly

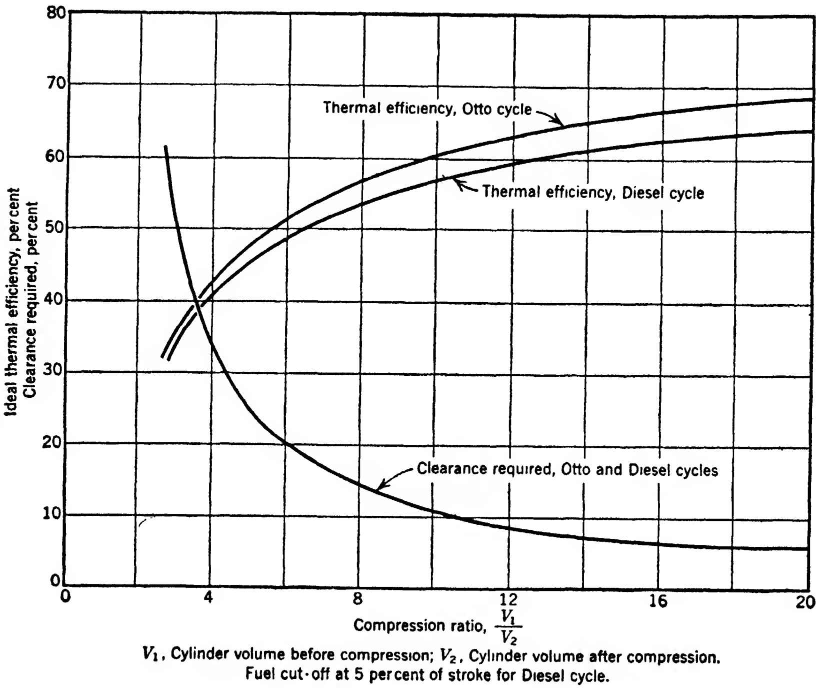


FIG. 19. Increase of thermal efficiency (air standard) with increase of compression ratio. Suction pressure assumed as 14.7 lb per sq in. abs. See Tables VII and VIII for accompanying data.

above atmospheric, and during the suction stroke the pressure in the cylinder is below atmospheric. The area of this loop, representing a loss of work, is usually small, especially if ample valve areas are provided and the engine is operated at a relatively low speed. The compression and expansion curves of the actual card are not true adiabatics, principally because of the heat loss to the water jacket and variable specific heat; the curves, however, approximate very closely the polytropic curve, $PV^n = C$. For a definite value of n, the discussion and equations given on the preceding six pages apply for any gas. The value of n for gasoline mixtures is generally taken as 1.32 for both the compression and expansion curves.

The thermal efficiency curves in Fig. 19 indicate that for equal ratios of compression there would be no thermodynamic advantage in substituting a Diesel engine for an Otto-cycle engine. But by avoiding any commingling of the fuel and air before compression, the use of a higher compression ratio becomes practicable and consequently a higher efficiency could be obtained with the Diesel cycle.

31. Ideal Diesel Cycle with Constant Specific Heat. Air Standard. Gas engines, automobile engines, and many oil engines are of the constant-volume (Otto-cycle) type of internal-combustion engine. If the pressure of the working substance does not change while combustion is taking place, the cycle is called the constant-pressure or Diesel cycle. Suppose that air alone is compressed into the clearance space of an engine cylinder before any fuel is admitted and that the fuel (generally oil) is then forced into the cylinder, burning as it enters, while the piston begins its outward (expansion) stroke. By suitably regulating the rate of admission of the fuel, the pressure is theoretically kept constant until combustion is completed, 2–3, Fig. 20. By avoiding any admixture of fuel with the air in the cylinder, before and during compression, a high ratio of compression is practicable with the Diesel cycle with no possibility of pre-ignition, and consequently a high thermal efficiency is generally obtained.

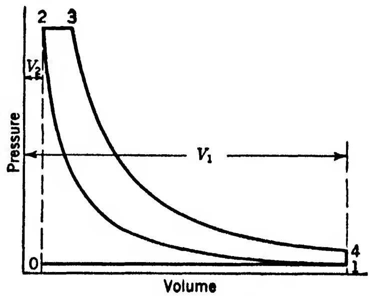


FIG. 20. Indicator diagram of the ideal Diesel cycle.

In the ideal Diesel cycle, Fig. 20, air enters the cylinder along 0–1; the air is compressed on the return stroke of the piston along the adiabatic curve 1–2. The air is compressed to about 500 lb per sq in. or more before the fuel is admitted. At 2 the fuel is forced into the cylinder, and, owing to the high temperature of the compressed air, the fuel ignites. The supply of fuel continues as the piston moves back, causing combustion at constant pressure. The curve 3–4 represents the adiabatic expansion of the products of combustion. At point 4 the exhaust valve opens and the pressure drops to 1 (atmospheric). On the fourth stroke, along 1–0, the products of combustion are forced through the exhaust opening; some of the products, however, remain in the clearance space and are mixed with the incoming air on the first stroke of the next cycle.

To simplify the analysis, certain conditions are assumed for the ideal air-standard cycle of the Diesel engine:

1. That the pressure of the air during the suction stroke is atmospheric.
2. That compression is adiabatic, which necessitates the assumption that the cylinder walls are non-conductors of heat.
3. That combustion of the fuel is at constant pressure, that is, the line 2–3, Fig. 20, is horizontal.
4. That a liquid fuel is completely vaporized before entering the cylinder.
5. That the fuel is injected into the cylinder without admixture of air, and that it enters the cylinder at the temperature T_1.
6. That expansion is adiabatic.
7. That the products of combustion are exhausted at atmospheric pressure, and that no exhaust gases are left in the clearance space.
8. That the specific heat is constant throughout the cycle (independent of temperature as well as pressure).

Since the lines 1–2 and 3–4, Fig. 20, are assumed adiabatics, all the heat of the cycle must therefore be supplied along line 2–3 and rejected along 4–1. The *heat supplied* along 2–3 is

$$ {}_2H_3 = Mc_p(T_3 - T_2) \tag{19} $$

in which M = weight of the air in the cylinder, assumed as constant since the fuel is generally oil and the increase in weight of the charge is small.

c_p = specific heat at constant pressure, 0.242 for air.

T_3 = absolute (F) temperature at 3.

T_2 = absolute (F) temperature at 2.

The *heat rejected* along 4–1 is

$$ {}_4H_1 = Mc_v(T_4 - T_1) \tag{20} $$

in which c_v = specific heat at constant volume, 0.173 for air.

T_4 = absolute (F) temperature at 4.

T_1 = absolute (F) temperature at 1.

The net work done is

$$ ({}_2H_3 - {}_4H_1) = Mc_p(T_3 - T_2) - Mc_v(T_4 - T_1) $$

The air-standard thermal efficiency of the *Diesel cycle* is

$$ \frac{{}_2H_3 - {}_4H_1}{{}_2H_3} = \frac{Mc_p(T_3 - T_2) - Mc_v(T_4 - T_1)}{Mc_p(T_3 - T_2)} $$

$$ = 1 - \frac{c_v(T_4 - T_1)}{c_p(T_3 - T_2)} $$

but ratio of specific heats $\frac{c_p}{c_v} = k$, hence

$$e'_a = 1 - \frac{(T_4 - T_1)}{k(T_3 - T_2)} \tag{21}$$

From the laws of perfect gases

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2} = \frac{P_3V_3}{T_3} = \frac{P_4V_4}{T_4} \tag{22}$$

also

$$P_1V_1^n = P_2V_2^n \quad \text{and} \quad P_3V_3^n = P_4V_4^n \tag{23}$$

Since 2–3 is at constant pressure and 4–1 at constant volume from equation 22,

$$T_3 = T_2\left(\frac{V_3}{V_2}\right) \tag{24}$$

and

$$T_4 = T_1\left(\frac{P_4}{P_1}\right) \tag{25}$$

From equation 23,

$$\frac{P_1}{P_2} = \left(\frac{V_2}{V_1}\right)^n \quad \text{and} \quad \frac{P_4}{P_3} = \left(\frac{V_3}{V_4}\right)^n$$

by dividing, and knowing that $P_2 = P_3$ and $V_1 = V_4$,

$$\frac{P_4}{P_1} = \left(\frac{V_3}{V_2}\right)^n \tag{26}$$

Substituting (26) in (25),

$$T_4 = T_1\left(\frac{V_3}{V_2}\right)^n \tag{27}$$

Substituting (27) and (24) in (21),

$$e'_a = 1 - \frac{T_1}{T_2}\left\{\frac{\left[\left(\frac{V_3}{V_2}\right)^n - 1\right]}{k\left(\frac{V_3}{V_2} - 1\right)}\right\} \tag{28}$$

Since T_1 and T_2 are unknown in most cases a ratio of volumes and pressures will usually be more convenient for finding the air-standard efficiency of the Diesel-engine cycle. From equations 22 and 23,

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{n-1} = \left(\frac{P_1}{P_2}\right)^{\frac{n-1}{n}}$$

Hence

$$e'_a = 1 - \left(\frac{V_2}{V_1}\right)^{n-1} \left\{ \frac{\left[\left(\frac{V_3}{V_2}\right)^n - 1\right]}{k\left(\frac{V_3}{V_2} - 1\right)} \right\} \tag{29}$$

and

$$e'_a = 1 - \left(\frac{P_1}{P_2}\right)^{\frac{n-1}{n}} \left\{ \frac{\left[\left(\frac{V_3}{V_2}\right)^n - 1\right]}{k\left(\frac{V_3}{V_2} - 1\right)} \right\} \tag{30}$$

Equation 30 shows that the thermal efficiency of the Diesel cycle depends not only upon the compression, but also upon the ratio of the volume V_3 at cut-off and the clearance volume V_2. The efficiency increases as this ratio $\left(\frac{V_3}{V_2}\right)$ decreases.

Example. Determine the air-standard efficiency of an engine operating on the Diesel cycle when the suction pressure is 14.5 lb per sq in. abs and the fuel is injected for 5 per cent of the stroke. The clearance volume is 8 per cent of the stroke. Assume that $n = k = 1.4$.

Solution. From equation 29,

$$e'_a = 1 - \left(\frac{0.08}{1.08}\right)^{0.4} \left\{ \frac{\left[\left(\frac{0.13}{0.08}\right)^{1.4} - 1\right]}{1.4\left(\frac{0.13}{0.08} - 1\right)} \right\}$$

$$e'_a = 1 - \frac{0.973}{2.832 \times 0.875}$$

$$= 0.607 \text{ or } 60.7 \text{ per cent}$$

Comparison of Figs. 17 and 20 shows the deviation of the actual Diesel card from the theoretical cycle. In the actual engine, compression approaches the adiabatic curve since practically pure air is compressed in the cylinder, and the value of exponent n is usually between 1.35 and 1.40. In general, the fuel cannot be injected rapidly enough to maintain a constant pressure, and line 2–3, Fig. 17, shows a slight decrease during combustion of the fuel. The value of n for the expansion curve of the actual engine is about 1.35 or less.

With increased compression ratios $\frac{V_1}{V_2}$, the air-standard efficiency of the ideal Diesel cycle increases as shown in Table VIII, taking $n = k = 1.4$ and assuming fuel injection during 5 per cent of the piston stroke. Table VIII and Fig. 19 show that there is at first a rapid gain

TABLE VIII

EFFECT OF COMPRESSION ON AIR-STANDARD THERMAL EFFICIENCY OF IDEAL DIESEL CYCLE

Clearance Volume V_2 in per cent	Fuel Cut-off, per cent Stroke	Compression Ratio $\frac{V_1}{V_2}$	Air-standard Thermal Efficiency	P_1 in lb per sq in abs	P_2 in lb per sq in abs	$\frac{P_2}{P_1}$
50.0	5	3	0.343	14.7	69	4.7
33.3	5	4	0.410	14.7	102	7.0
20.0	5	6	0.489	14.7	180	12.2
14.7	5	8	0.535	14.7	270	18.4
11.1	5	10	0.569	14.7	369	25.1
9.1	5	12	0.594	14.7	477	32.4
7.7	5	14	0.612	14.7	590	40.1
6.7	5	16	0.628	14.7	713	48.5
5.9	5	18	0.639	14.7	840	57.1

of thermal efficiency with increased compression, but as the compression ratio becomes higher the *rate of increase* of the thermodynamic advantage decreases.

32. Comparison of the Ideal Cycles of Combustion. Experience has indicated that, if the performance of an internal-combustion engine at a given compression ratio were known, its performance at other ratios of compression could be calculated by a consideration of air-cycle efficiencies for similar conditions. The actual thermal efficiency of an engine at two ratios of compression is in almost exact proportion to the thermal efficiency of the ideal air cycles of the same compression ratios. For example, a certain Otto-cycle engine with a 4 to 1 compression ratio is known to have an output of 75 hp and its probable output is desired at a compression ratio of 6 to 1. The thermal efficiency of an Otto air-standard cycle at 4 to 1 compression ratio is 42.6 per cent, whereas the efficiency of the same cycle at 6 to 1 compression ratio is 51.2 per cent; see Table VII and Fig. 19. The ratio of efficiencies is 1.2, hence the horsepower of the actual engine at 4 to 1 compression ratio may be multiplied by 1.2 to obtain the probable output of the engine at 6 to 1 compression ratio; this would be 90 hp.

Figures 21 to 26, inclusive, show a number of ideal cycles and illustrate the various possibilities of engine improvement. Comparing the actual thermal efficiency of an engine with the efficiency of the corresponding ideal cycle gives an indication of the margin of improvement for reducing the thermodynamic losses that occur in the operation of the real engine. In all cases, the ideal thermal efficiency is higher than the actual efficiency; the difference is caused by imperfect combustion, variable specific heat, wire-drawing of the charge through the induction system, work required to expel the exhaust gases, heat losses, friction, and other practical limitations.

The displacement for each cycle, Figs. 21 to 26, was made 100 cu in. The initial pressure and temperature of each cycle were taken as 14.7 lb per sq in. and 68 F, respectively; compression and expansion curves were assumed to be adiabatics, with $n = k = 1.4$. The heat added was to be 1000 Btu per lb of air for each cycle. The displacement volume plus the clearance volume gives the total cylinder volume; thus in Fig. 22, $V_1 = 100 + 20 = 120$ cu in.

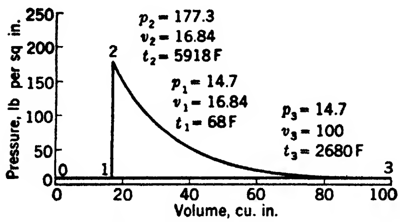


FIG. 21. Ideal Lenoir cycle. Expansion ratio, 5.93 to 1; thermal efficiency, 38 per cent; net work, 216 ft-lb.

Lenoir's 1860 engine, as described on page 4, employed no precompression of the gaseous charge. As the piston moved away from dead center, the charge was drawn into the cylinder. At the completion of part of this stroke, the intake valve closed and the charge was exploded with an electric spark. Expansion could be continued to atmospheric pressure; on account of the absence of compression, the efficiency and output of the engine were very low. Figure 21 shows the theoretical indicator card for an ideal Lenoir engine. The line 0–1 represents admission, 1–2 explosion, 2–3 expansion, and 3–0 exhaust.

Referring to Fig. 21, the *heat supplied* along 1–2 is

$$ {}_1H_2 = Mc_v(T_2 - T_1) \tag{31} $$

the *heat rejected* along 3–1 is

$$ {}_3H_1 = Mc_p(T_3 - T_1) \tag{32} $$

and the air-standard thermal efficiency of the *Lenoir cycle* is

$$ e_L = \frac{{}_1H_2 - {}_3H_1}{{}_1H_2} = 1 - \frac{k(T_3 - T_1)}{(T_2 - T_1)} = 1 - \frac{k\left(\dfrac{T_3}{T_1} - 1\right)}{\left(\dfrac{T_2}{T_1} - 1\right)} \tag{33} $$

For the constant-pressure process, 3–1, from

$$ \frac{P_1V_1}{T_1} = \frac{P_3V_3}{T_3}, \quad \frac{T_3}{T_1} = \frac{V_3}{V_1} \tag{34} $$

for the constant-volume process, 1–2, from

$$ \frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}, \quad \frac{T_2}{T_1} = \frac{P_2}{P_1} \tag{35} $$

and for the adiabatic expansion, 2–3,

$$ \frac{P_2}{P_3} = \left(\frac{V_3}{V_2}\right)^k = \left(\frac{V_3}{V_1}\right)^k = \frac{P_2}{P_1} \tag{36} $$

because $P_3 = P_1$ and $V_2 = V_1$.

The ratio $\frac{V_3}{V_2} = r$, the *ratio of expansion;* but $V_2 = V_1$, hence $\frac{V_3}{V_1} = r$. Substituting equations 34, 35, and 36, successively in equation 33,

$$e_L = 1 - \frac{k(r-1)}{(r^k - 1)} \tag{37}$$

Example. Using the information shown in Fig. 21, check the results given in the legend; ideal Lenoir cycle with an expansion ratio of 5.93.

Solution. Substituting in equation 37; thermal efficiency,

$$e_L = 1 - \frac{1.4\ (5.93 - 1)}{(5.93)^{1.4} - 1}$$

$$= 1 - \frac{6.9}{11.1} = 0.38 \text{ or } 38 \text{ per cent}$$

Net work per cycle,

$$W_{123} = 778\ ({}_1H_2 - {}_3H_1) = 778\ [Mc_v(T_2 - T_1) - Mc_p(T_3 - T_1)]$$

$$M = \frac{P_1V_1}{R_1T_1} = \frac{14.7 \times 144}{53.35 \times 528} \times \frac{16.84}{1728} = 0.00073 \text{ lb air per cycle}$$

$$W_{123} = 778\ [0.00073 \times 0.173\ (6378 - 528) - 0.00073 \times 0.242\ (3140 - 528)]$$

$$W_{123} = 778\ (0.739 - 0.461) = 216 \text{ ft-lb}$$

The compression ratio of the ideal Otto cycle of Fig. 22 is 6 to 1; for a thermal efficiency of 51.2 per cent the clearance volume required is 20

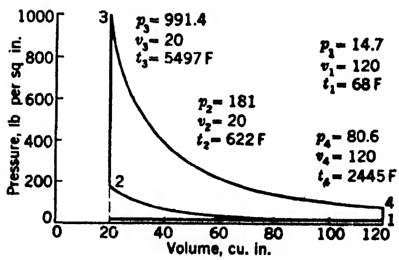


FIG. 22. Ideal Otto cycle. Compression ratio, 6 to 1; thermal efficiency, 51.2 per cent; net work, 1730 ft-lb.

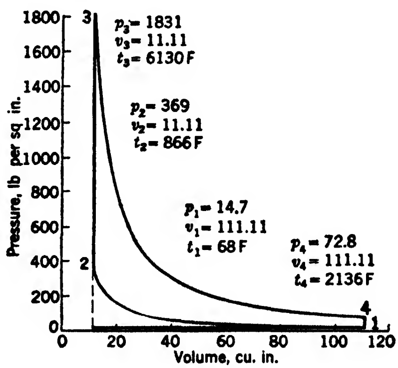


FIG. 23. Ideal Otto cycle. Compression ratio 10 to 1; thermal efficiency, 60.2 per cent; net work, 2038 ft-lb.

per cent and the maximum explosion pressure attained is 991.4 lb per sq in. Comparing these results with the Lenoir cycle of Fig. 21, expansion ratio of 5.93 to 1 and a thermal efficiency of 38 per cent, shows the advantages of compression of the charge in the cylinder before combustion. The ideal Diesel engine with a compression ratio of 6 to 1 and fuel cut-off at 5 per cent of the stroke has a thermal efficiency of 48.9 per cent; see Table VIII.

Figure 23 shows an ideal cycle for an Otto engine of 10 to 1 compression ratio; Fig. 25 shows an ideal cycle for a Diesel engine of 20.23 to 1 compression ratio. It will be noted that the efficiency and the net work of the two cycles are almost identical. Experimental internal-combustion engines with compression ratios of 20 to 1 have been built, and their possibilities will continue to be of interest to designers and engineers. Otto-cycle engines with 10 to 1 compression ratios are being used for special applications where fuel of sufficiently high " anti-knock " value is available.

An ideal indicator card for a Diesel engine is shown by Fig. 24. The compression ratio is 13 to 1 with a compression pressure of 533 lb per sq in. and temperature of 1013 F; fuel injection is for 22 per cent of the stroke. It should be noted that the efficiency of the Diesel cycle of

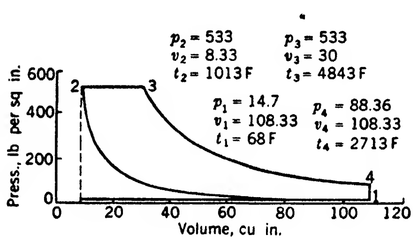


Fig. 24. Ideal Diesel cycle. Compression ratio, 13 to 1; thermal efficiency, 50.5 per cent; net work, 1707 ft-lb.

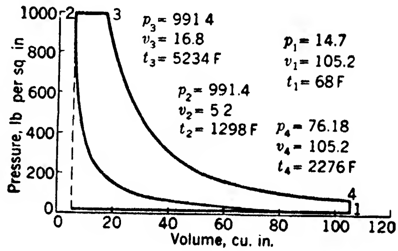


Fig. 25. Ideal Diesel cycle. Compression ratio, 20.23 to 1; thermal efficiency, 59.8 per cent; net work, 2016 ft-lb.

Fig. 24 is 50.5 per cent, which is approximately equal to that of the ideal Otto cycle of 6 to 1 compression ratio. The power output for an engine of a given size is slightly less with this 13 to 1 Diesel cycle than that of the 6 to 1 Otto cycle at the same speed. The Diesel engine depends upon high compression pressures for its efficiency, and at ratios of compression used in Otto-cycle engines it would be less efficient. With an ideal Diesel cycle of 6 to 1 compression ratio and 5 per cent fuel injection, see Table VIII, the thermal efficiency is 48.9 per cent; this is less than for the ideal Otto cycle with the same compression ratio. The Diesel engine would not operate satisfactorily with compression ratios of less than 10 to 1.

Considerable experimental work is being done on engines using the *dual* or combination *cycle* shown in Fig. 26. In engines of this type a portion of the fuel, usually about 30 per cent, is burned at constant volume whereas the remainder is burned at constant pressure. Some engines of this type have ignition systems; others depend upon the compression temperature of the charge to effect ignition.

With reference to Fig. 26, the *heat supplied* at constant volume along 2–3 is

$$_2H_3 = Mc_v(T_3 - T_2) \tag{38}$$

the *heat supplied* at constant pressure along 3–4 is

$$_3H_4 = Mc_p(T_4 - T_3) \tag{39}$$

and the *total heat supplied* along 2–3–4 is

$$H_{234} = Mc_v(T_3 - T_2) + Mc_p(T_4 - T_3) \tag{40}$$

Equation 40 assumes that the same amount of air will be in the cylinder during the entire combustion period 2–3–4. The portion of the heat

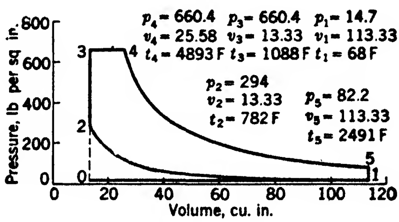


FIG. 26. Ideal dual (also called combination, duplex, mixed, and semi-Diesel) cycle; 30 per cent of heat added at constant volume and 70 per cent at constant pressure. Compression ratio, 8½ to 1; thermal efficiency, 52.7 per cent; net work, 1783 ft-lb.

added during constant volume 2–3 and the part added during constant pressure 3–4 depend upon the design of the engine. The *heat rejected* at constant volume along 5–1 is

$$_5H_1 = Mc_v(T_5 - T_1) \tag{41}$$

The air-standard thermal efficiency of the *dual cycle* is

$$e_d = \frac{H_{234} - {_5H_1}}{H_{234}} = 1 - \frac{(T_5 - T_1)}{(T_3 - T_2) + k(T_4 - T_3)} \tag{42}$$

in terms of pressures and volumes,

$$e_d = 1 - \frac{V_5(P_5 - P_1)}{V_2(P_3 - P_2) + kP_3(V_4 - V_3)} \tag{43}$$

also,

$$e_d = 1 - \frac{\left[\left(\frac{P_3}{P_2}\right)\left(\frac{V_4}{V_3}\right)^n - 1\right]}{\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\left[\left(\frac{P_3}{P_2} - 1\right) + k\left(\frac{P_3}{P_2}\right)\left(\frac{V_4}{V_3} - 1\right)\right]} \tag{44}$$

Engines operating on the dual cycle generally employ compression ratios higher than is customary with the Otto cycle but lower than the usual Diesel-cycle compression ratios. The efficiency of the ideal dual cycle shown in Fig. 26 is 52.7 per cent; comparing this result with the Otto-cycle efficiency for a compression ratio of 6 to 1 shows a gain of only 1.5 per cent, although the compression ratio of the dual cycle is considerably higher.

The *Carnot cycle* consists of four successive stages: isothermal addition of heat, adiabatic expansion, isothermal rejection of heat, and adiabatic compression to the original condition of pressure, volume, and temperature. This cycle represents the most efficient means for transforming heat into work; it is a hypothetical cycle but useful as a measure of the limiting efficiency of other heat-engine cycles. The efficiency of the Carnot cycle is generally stated in the form

$$e_c = \frac{T - T_o}{T} \tag{45}$$

where T = highest absolute (F) temperature reached in the cycle.
T_o = lowest absolute (F) temperature reached in the cycle.

Diesel's original proposal was to compress air to the highest possible temperature, followed by combustion at this temperature; thus he was following the Carnot cycle more closely than any of the other theoretical internal-combustion-engine cycles.

33. Ideal Otto Cycle with Variable Specific Heat. Gas Standard. In Article 30 the air-standard thermal efficiency for Otto-cycle engines was explained fully, and numerical results were given in Table VII. The air-standard efficiency, however, is much above what could be obtained in an actual engine even if every source of loss were removed. The reason for this is that the gaseous mixture employed in practice, although mainly air, is by no means entirely so, and consequently it is incorrect to assume that the specific heat of the working substance is constant throughout the cycle.

Experiments have shown that the gases which are present in the engine cylinder after explosion have specific heats which increase with increase of temperature. The efficiency of the cycle is decreased as a result of this increase of specific heat. For a given amount of fuel burned, the rise of temperature during explosion is decreased and the pressure P_3, Fig. 18, is less than that attained if the specific heat were constant. The expansion curve 3–4, Fig. 18, is also lowered and the work of the cycle is decreased. It is therefore necessary to make allowance for this change in specific heat and replace the air standard with a *gas standard* based on fuel-air mixtures commonly used in Otto-cycle engines.

The specific heat of a gas is a function of the temperature, usually a second degree function. The specific heats of the gases (N_2, CO_2, H_2O, etc.) generally present in the cylinder of an internal-combustion engine may be calculated from the equations of the general form

$$c_v = a + bT + cT^2 \tag{46}$$ [3]

$$c_p = a' + bT + cT^2 \tag{47}$$ [3]

where a, a', b, and c are constants, and T is absolute Fahrenheit temperature.

The working substance of the actual engine consists almost entirely of nitrogen, carbon dioxide, and water vapor. The specific heat of these three gases shows an increase with rise in temperature; and the last two gases dissociate [4] (splitting of molecules into simpler constituents) at high temperatures, especially at low pressures. Tables [5] IX and X give the mean specific heats at constant volume and the percentage of dissociation for these gases.

TABLE IX

MEAN SPECIFIC HEATS AT CONSTANT VOLUME, BETWEEN 200 F AND GIVEN TEMPERATURES, BTU PER POUND PER DEGREE F

Gas	Temperature in Degress Fahrenheit					
	930	1830	2730	3630	4530	5430
Nitrogen	0.185	0.188	0.196	0.205	0.214	0.225
Carbon dioxide	0.187	0.217	0.229	0.238	0.247	0.249
Water vapor	0.350	0.385	0.425	0.468	0.540	0.623

Calculations of the gas-standard efficiency, taking into account the variable specific heats and dissociation, are somewhat laborious and will be omitted here. However, the results of computations for the Otto cycle with gasoline (octane, C_8H_{18}) as the fuel for four compression ratios and with varying fuel-air mixtures are tabulated on pages 38 and 39, *Bulletin* 160, "A Thermodynamic Analysis of Internal-combustion Engine Cycles," Goodenough and Baker, University of Illinois Engi-

[3] Expressions for the specific heats of a number of gases are given on page 106, *Bulletin* 139, "An Investigation of the Maximum Temperatures and Pressures Attainable in the Combustion of Gaseous and Liquid Fuels," Goodenough and Felbeck, University of Illinois Engineering Experiment Station, 1924.

[4] *Dissociation* is described on pages 60 to 61.

[5] Tizard and Pye, *The Automobile Engineer*, February, 1921.

TABLE X

DISSOCIATION OF CARBON DIOXIDE AND WATER VAPOR AT VARIOUS TEMPERATURES AND PRESSURES, PER CENT

Temperature, Degrees Fahrenheit	Pressure in Pounds per Square Inch							
	1.47		14.7		147		1470	
	CO_2	H_2O	CO_2	H_2O	CO_2	H_2O	CO_2	H_2O
2730	0.104	0.043	0.048	0.02	0.0224	0.009	0.01	0.004
3630	4.35	1.25	2.05	0.58	0.96	0.27	0.445	0.125
4530	33.5	8.84	17.6	4.21	8.63	1.98	4.09	0.927
5430	77.1	28.4	54.8	14.4	32.2	7.04	16.9	3.33

neering Experiment Station, 1927. These results have been plotted in Fig. 27, included as Fig. 8 in the afore-mentioned bulletin, which shows

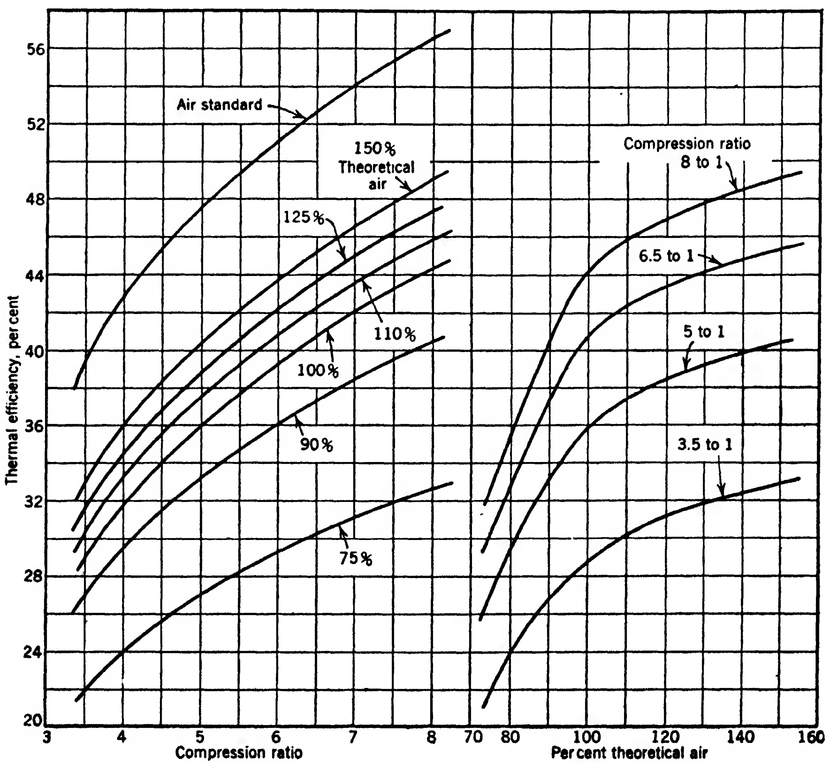


FIG. 27. Variation of efficiency with compression ratio and with mixture strength.

the variation of efficiency with the compression ratio and with mixture strength. The gas-standard thermal efficiencies increase as the mixture becomes leaner (more air), because of the decreased temperature rise with leaner mixtures and smaller change in the specific heat. With a large excess of air in the fuel-air mixture, the temperature rise approaches zero and the mean specific heat approaches a constant value, equal to the specific heat used in the air-standard cycle. Therefore the gas-standard thermal efficiency approaches the air-standard thermal efficiency as the percentage of air increases; see Fig. 27. This analysis also showed that the mean effective pressure, and therefore the power, is a maximum when the air supply is somewhat less than 100 per cent of the theoretical amount. Thus the mixture for maximum power is a mixture of relatively low efficiency.

Goodenough and Baker's analysis with gasoline as a fuel was extended to include benzene (C_6H_6) and kerosene ($C_{12}H_{26}$) to determine the effect of the fuel on the efficiency. The gas-standard efficiencies were practically the same for the three fuels with 100 per cent theoretical air and a compression ratio of 5. This result confirmed the results of other investigators, such as Tizard and Pye, that *the ideal efficiency of an internal-combustion engine is independent of the kind of liquid fuel used.*

The air-standard thermal efficiency of the Otto cycle is given by equation 18,

$$e_a = 1 - \left(\frac{V_2}{V_1}\right)^{n-1}$$

with $n = k = 1.4$, as assumed in the air-standard cycle, the above equation becomes

$$e_a = 1 - \left(\frac{V_2}{V_1}\right)^{0.4}$$

The same type of equation may be used to express the correct value of the gas-standard thermal efficiency, but the exponent, instead of being a constant 0.4, will be a variable and a function of the compression ratio and the air supply. Designating the exponent in the above equation by x, the *gas-standard thermal efficiency* of the Otto cycle may be given by the equation

$$e_g = 1 - \left(\frac{V_2}{V_1}\right)^{x} \tag{48}$$

Selecting values of the thermal efficiency in Fig. 27, the corresponding values of x for various compression ratios and fuel-air mixtures, calculated from the preceding equation, are those given in Table XI.

TABLE XI *

VALUE OF EXPONENT x IN EQUATION 48

Compression Ratio, $r = \frac{V_1}{V_2}$	Per Cent Air Supplied			
	100	110	125	150
3.5	0.2723	0.2874	0.3022	0.3189
5.0	0.2770	0.2916	0.3055	0.3214
6.5	0.2802	0.2942	0.3075	0.3232
8.0	0.2815	0.2964	0.3087	0.3248

* Table 7, University of Illinois Engineering Experiment Station, *Bulletin* 160, p. 51.

Let a denote the percentage of air supplied of the theoretical amount required; thus in Table XI, a = 100, 110, 125, 150. Then the following empirical equation gives quite accurately the values of x in Table XI,

$$x = 0.3867 - \frac{6.5}{a - 35} - \frac{0.043}{r} \tag{49}$$

Equation 49 applies for 100 per cent or more of the theoretical air. If the air supply were insufficient for complete combustion, the following equation gives an approximation for the exponent in equation 48,

$$x = 0.524 - \frac{24.6}{a} \tag{50}$$

Tizard and Pye [6] proposed the following equation for the gas-standard thermal efficiency

$$e_g = 1 - \left(\frac{V_2}{V_1}\right)^{0.295} \tag{51}$$

The exponent 0.295 was substantiated by Goodenough and Baker as indicated by the values given in Table XI for a 10 to 25 per cent excess of air. The values of x for 150 per cent air are of theoretical interest only, as such an excess of air would probably give a non-explosive mixture.

34. Ideal Diesel Cycle with Variable Specific Heat. Gas Standard. In Art. 31 the air-standard thermal efficiency for Diesel-cycle engines was explained fully, and numerical results given in Table VIII. In general, the discussion of the effects of variable specific heat on the Otto-cycle thermal efficiency, given in the first four paragraphs of Art.

[6] Tizard and Pye, *The Automobile Engineer*, February, 1921.

33, are applicable to the Diesel cycle. Calculations of the gas-standard thermal efficiency for the Diesel cycle, taking into account the variable specific heats and dissociation, are lengthy and will be omitted. However, the results of computations for the Diesel cycle with kerosene ($C_{12}H_{26}$) as the fuel for four compression ratios and with varying fuel-air mixtures are tabulated on page 43, *Bulletin* 160, University of Illinois

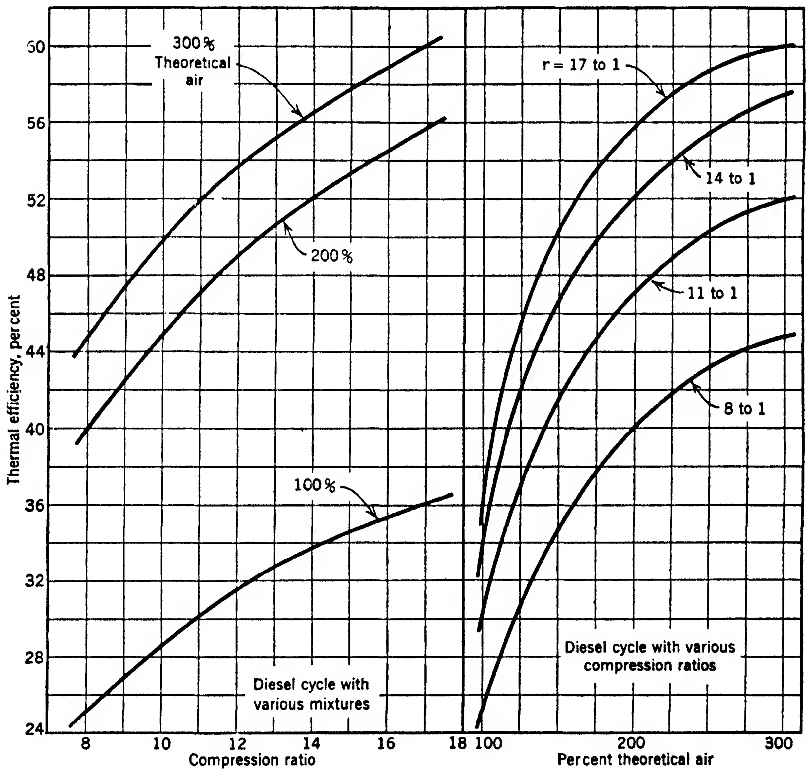


FIG. 28. Effect of compression ratio and mixture strength upon efficiency of the Diesel cycle.

Engineering Experiment Station. These results have been plotted in Fig. 28, included as Fig. 11 in above bulletin, which shows the variation of efficiency with the compression ratio and with mixture strength. For the *gas-standard thermal efficiency* of the Diesel cycle, equation 48 may also be used with values of x given by the empirical equation

$$x = 0.434 - \frac{19.5}{a - r} - \frac{0.7}{r} \tag{52}$$

In the Diesel cycle, Fig. 20, the internal energy U_1 of the air in the initial state 1 is supplemented by the energy added by the injection of the fuel oil along 2–3. The energy of the oil plus the equivalent of the work of forcing it into the cylinder may be represented by the thermal potential i_o. For the adiabatic cycle, the work is given by the equation

$$A(W) = U_1 + i_o - U_4 \tag{53}$$

where $A = \frac{1}{778}$ Btu per ft-lb.

The term U_4 represents the energy of the mixture in the final state 4, the opening of the exhaust valve. This energy consists of two parts: the chemical energy of the unburned CO and H_2, and the thermal energy of the mixture of the products at the end of expansion. With 100 per cent or more air the amount of unburned CO and H_2 is small and the chemical energy in the mixture at T_4 is correspondingly small. Therefore, the energy U_4 will depend primarily on the temperature T_4. Any conditions that will result in a decrease of T_4 will cause a reduction in U_4, a subtractive term in equation 53, and consequently an increase in the work done and an increase in thermal efficiency.

The effect of compression on the temperature at state 3, Fig. 20, is as follows: with 100 per cent air, the increase of the ratio of compression from $3\frac{1}{2}$ to 8 causes T_3 to rise from 4980 to 5150 F, or only 170 F. During the adiabatic expansion 3–4, the compression ratio of $3\frac{1}{2}$ gives a decrease of temperature from 4980 to 3980 F, or 1000 F drop, whereas the compression ratio of 8 gives a decrease of temperature during adiabatic expansion from 5150 to 3405 F, or 1745 F drop. That is, the temperature at the beginning of expansion is higher for the higher compression, but that at the end of expansion is much lower. *The improved efficiency obtained with higher compression is due not to any effect of compression on combustion but solely to the more complete conversion of the energy of the products into work as the result of more complete expansion.*[7] Evidently, if the expansion could be made still more complete by improvements of the cycle, the efficiency would be further increased. With an increased supply of air the heat of combustion per unit weight of fuel is required to raise the temperature of a larger weight of air. Therefore, the temperature (T_3) at the beginning of expansion will decrease as the excess air increases; the temperature (T_4) at the end of expansion is correspondingly decreased and the efficiency will increase. This increase of efficiency with increase of excess air is verified by the curves shown on the left side of Fig. 28.

[7] "A Thermodynamic Analysis of Internal-combustion Engine Cycles," Goodenough and Baker, *University of Illinois Engineering Experiment Station Bulletin* 160, 1927.

PROBLEMS

1. An engine working on the air-standard Otto cycle has a compression ratio of 4.5. At the beginning of compression the pressure is 14 lb per sq in. abs and the temperature 90 F. Determine the pressure and temperature at points 2, 3, and 4 in Fig. 18. Assume the heat supplied per cycle as 453 Btu per lb of charge. Find the heat rejected per pound of charge, also the thermal efficiency.

Ans. Heat rejected, 248 Btu per lb.

2. Calculate the air-standard thermal efficiency of the Lenoir cycle, Fig. 21, if the pressure at 1 is 14 lb per sq in. abs, a temperature of 90 F at 1, and an expansion ratio of 4.5.

3. Assume a temperature range of 200 F for a Carnot cycle during which 50 Btu were added to 3 lb of air at a temperature of 500 F. (*a*) What is the thermal efficiency of the cycle? (*b*) How much heat is rejected during the cycle? (*c*) Find the external work done in foot-pounds.

4. A $9\frac{1}{2}$ by 19 in. gas engine is assumed to work on the ideal Otto cycle with adiabatic expansion and compression, receiving and rejecting heat at constant volume. The clearance volume is 272 cu in. At the end of the suction stroke the pressure is 14 lb per sq in. abs, and the temperature of the charge is 210 F. (*a*) What is the air-standard thermal efficiency? (*b*) Determine the pressure and temperature at the end of the compression stroke.

5. Determine the clearance volume in percentage of the stroke for an Otto-cycle engine that gives an air-standard thermal efficiency of 52 per cent.

6. (*a*) Find the clearance volume in percentage of the stroke in an ideal Otto cycle if the pressure at the beginning of compression is 14 lb per sq in. abs and at the end of compression is 140 lb per sq in. abs; $n = k = 1.4$. (*b*) If $n = 1.32$, will the clearance be larger or smaller? Prove by calculation.

7. The piston displacement of a gas engine operating on the ideal air-standard Otto cycle is 1 cu ft, the clearance volume 0.2 cu ft, the pressure at the beginning of compression 14.7 lb abs, and the temperature at the beginning of compression 140 F. The engine receives 0.06 cu ft of gas (600 Btu per cu ft) per cycle. Find: (*a*) weight of the cylinder contents; (*b*) pressure and temperature at end of compression; (*c*) rise of temperature during explosion; (*d*) pressure at end of explosion; (*e*) pressure and temperature at end of expansion; (*f*) thermal efficiency of the cycle; (*g*) thermal efficiency of an engine working on a Carnot cycle between the same highest and lowest temperatures.

Ans. (*e*) 46 lb abs; (*f*) 51.2 per cent; (*g*) 84.4 per cent.

8. The pressure at the beginning of compression in an Otto cycle is 14 lb per sq in. abs; at the end of compression it is 112 lb per sq in. abs. (*a*) Find the clearance in percentage of the stroke when $n = k = 1.4$, when $n = 1.3$, and when $n = 1.2$. (*b*) Find the thermal efficiency of the cycle when $n = k = 1.4$, when $n = 1.3$, and when $n = 1.2$.

9. A gas is compressed isothermally to 25 per cent of its original volume; heat is then added at constant volume until the pressure increases to three times the pressure at the end of compression. This is followed by adiabatic expansion until the initial pressure is reached; rejection of heat is at constant pressure. (*a*) Find the air-standard thermal efficiency of the cycle. (*b*) Find the air-standard thermal efficiency if the compression and expansion curves were both adiabatics.

Ans. (*a*) 41.1 per cent; (*b*) 52.1 per cent.

10. In a gas engine the compression ratio is 5.5; the pressure at the beginning of compression is 13.5 lb per sq in. abs and at the end of compression 135 lb per sq in.

abs. Determine the value of the exponent n if the compression curve is of the form $PV^n = C$, a constant.

11. The compression ratio of a gasoline (Otto-cycle) engine is 6. At the beginning of compression the pressure is 14.5 lb per sq in. abs and the temperature 100 F. Find the temperatures: (*a*) at the end of compression when the pressure is 160 lb per sq in. abs; (*b*) at the end of explosion at constant volume when the pressure is 450 lb per sq in. abs; (*c*) at 95 per cent of the expansion stroke when the pressure is 48 lb per sq in. abs. (*d*) Determine the exponent n for the compression curve and for the expansion curve.

12. The pressure at the beginning of compression in an Otto cycle is 13.7 lb per sq in. abs and at the end of compression 130 lb per sq in. abs. If the clearance volume is 23 per cent, find the numerical value of the exponent n of the compression line.

13. Determine the thermal efficiency of an internal-combustion engine operating on the ideal Otto cycle for clearance volumes of 2, 5, 10, 15, 20, 30, and 40 per cent; $n = 1.32$. Plot a curve showing the variation of thermal efficiency with clearance.

14. By a method similar to that used in finding the theoretical mean effective pressure of a steam-engine cylinder, find the theoretical mep of an internal-combustion engine operating on the Otto cycle, if the suction pressure is 14.5 lb per sq in. abs. Pressure at the end of compression is 87 lb per sq in. abs; that at the end of combustion is 300 lb per sq in. abs. Use $n = 1.32$. *Ans.* 81 lb per sq in.

15. In an ideal Otto cycle with adiabatic compression and adiabatic expansion the pressure, volume, and temperature at the beginning of compression are 14 lb per sq in. abs, 1 cu ft, and 100 F, respectively. At the end of compression the pressure is 112 lb per sq in. abs, and at the end of combustion the pressure is 280 lb per sq in. abs. Find the theoretical horsepower developed for 250 complete cycles per minute.

16. Determine the clearance volume in percentage of the stroke for an Otto-cycle engine that gives a *gas-standard* thermal efficiency of 52 per cent with an air supply of 120 per cent of the theoretical amount required. Substitute in equation 51 for trial value of r, then use equations 49 and 48, successively, until correct (correlated) result is obtained.

17. (*a*) Calculate the air-standard thermal efficiency for a Diesel engine having a clearance of 5 per cent and fuel injection during 7 per cent of the stroke, $n = k = 1.4$. (*b*) Same for an Otto-cycle engine with a clearance of 5 per cent. Compare result with that of (*a*). Explain the difference briefly.

18. Determine the air-standard thermal efficiency of the following ideal internal-combustion engine cycle: (*a*) adiabatic compression from volume V_1 to volume V_2, $V_1 = 7V_2$; (*b*) heat added at constant pressure until volume $V_3 = 2V_2$; (*c*) adiabatic expansion to volume $V_4 = 1.5V_1$; (*d*) heat rejected at constant volume V_4; (*e*) heat rejected at constant pressure until initial conditions are obtained.

19. An engine operating on the constant-pressure (Diesel) air-standard cycle has a compression ratio of 4.5. At the beginning of compression the pressure is 14 lb per sq in. abs, the temperature 90 F. Determine the pressure and temperature at points 2, 3, and 4 in Fig. 25. Assume the heat supplied per cycle to be 453 Btu per lb of charge. Find the heat rejected per pound of charge, also the thermal efficiency. Compare result with that given in problem 1; explain the difference briefly. *Ans.* Heat rejected, 322 Btu per lb.

20. (*a*) Calculate the clearance in percentage of stroke for a Diesel engine that gives a pressure of 540 lb per sq in. abs at the end of compression. Pressure at the beginning of compression 14 lb per sq in. abs; use $k = 1.4$ and $n = 1.35$. (*b*) If the temperature at the beginning of compression is 185 F, what is the temperature at the end of compression? (c) If fuel injection continues at constant pressure for 5 per cent of the stroke, what is the temperature at the end of fuel injection?

21. Assume that it is desirable to obtain a temperature of 1000 F at the end of compression in a Diesel cycle. The initial conditions are 13.5 lb per sq in. abs and 120 F. Find the clearance volume in percentage of stroke and the pressure at the end of compression: (a) when compression is according to $PV^{1.4} = C$; (b) when $PV^{1.35} = C$; (c) when $PV^{1.3} = C$. *Ans.* (a) 11 per cent; 342 lb abs.

22. If fuel were injected for 6 per cent of the stroke in problem 21, find: (a) the temperature at the end of fuel injection when $n = 1.35$ for the compression curve; (b) the pressure and temperature at the end of expansion when $n = 1.35$ for the compression curve and $n = 1.3$ for the expansion curve.

Ans. (b) 32.7 lb and 940 F.

23. The pressure at the beginning of compression in a Diesel engine is 14 lb per sq in. abs, the pressure at the end of compression is 518 lb per sq in. abs, and the clearance volume is 8 per cent of the stroke. Find the exponent n of the compression line and the temperature ratio.

24. Calculate the theoretical thermal efficiency for Diesel engines having clearance volumes of 3, 6, 9, 12, and 15 per cent, with fuel injection for 8 per cent of the stroke in each case. Use $k = 1.4$ and $n = 1.35$.

25. Determine the theoretical thermal efficiency of a Diesel engine with 8 per cent clearance when the fuel is injected for 2, 6, 10, 14, and 20 per cent of the stroke. Plot a curve showing the variation of thermal efficiency with fuel cut-off. Use $k = 1.4$ and $n = 1.35$.

26. In a Diesel-engine cycle $p_1 = 14.5$ lb per sq in. abs, $t_1 = 125$ F, $p_2 = p_3 = 522$ lb per sq in. abs, and $n = 1.33$. (a) Find the required clearance in percentage of stroke. (b) Determine t_2, t_3, t_4, and p_4 for fuel cut-offs of 2, 6, 10, 14, and 20 per cent of stroke. (c) Find the mep for each of the above cut-offs. (d) Tabulate above data and results. (e) Plot curves for t_3, t_4, p_4, and mep using fuel cut-off as abscissa.

27. The compression in a Diesel-engine cylinder follows the curve $PV^{1.35} = C$. The pressure at the beginning of compression is 14.5 lb per sq in. abs, and the temperature 100 F. (a) Find the compression ratio if the pressure at the end of compression is 450 lb per sq in. abs. (b) What temperature is attained? (c) What is the temperature at the end of fuel injection if the fuel valve closes at 10 per cent of the expansion stroke and the pressure is 425 lb per sq in. absolute?

28. In problem 27, assume that the decrease in pressure during fuel injection is a straight line and that expansion follows the curve $PV^{1.35} = C$. Find the mep of the cycle. *Ans.* 82.5 lb per sq in.

29. A Diesel engine with a compression ratio of 12 is assumed to operate on the air-standard cycle. The pressure at the beginning of *compression* is 14.7 lb per sq in. abs and the temperature 140 F. The temperature at the end of adiabatic *expansion* is 640 F. During what percentage of the stroke would it be necessary to inject fuel for constant-pressure burning? *Ans.* 4.93 per cent.

30. Calculate the clearance volume in percentage of the stroke for a Diesel-cycle engine that gives a *gas-standard* thermal efficiency of 32 per cent with an air supply of 120 per cent of the theoretical amount required. Substitute in equation 51 for trial value of r; then use equations 52 and 48, successively, until correct (correlated) result is obtained.

31. An internal-combustion engine working on the dual (combination) air-standard cycle has a compression ratio of 10. At the beginning of compression the pressure is 14 lb per sq in. abs and the temperature 90 F. Assume the heat supplied at constant volume per cycle to be 200 Btu per lb of charge and that supplied at constant pressure per cycle to be 250 Btu per lb of charge. Determine the pressure and temperature at points 2, 3, 4, and 5 in Fig. 26. Find the heat rejected per pound of charge; also the thermal efficiency. *Ans.* Heat rejected, 187 Btu per lb.

32. The compression ratio of an ideal dual-combustion cycle is 10. The pressure ratio, during constant-volume combustion, is 1.8; the volumetric ratio, during constant-pressure combustion, is 1.6. Determine the thermal efficiency of the cycle if $k = 1.4$ and $n = 1.3$.

33. In a dual (combination) internal-combustion engine cycle, given $t_1 = 140$ F, $p_1 = 14$, $p_2 = 300$, and $p_3 = 500$ lb abs. Assume fuel injection for 5 per cent of the stroke and $n = k = 1.4$. (*a*) Find the clearance in percentage of the stroke. (*b*) What is the air-standard thermal efficiency? (*c*) Determine the theoretical mep of the cycle.

34. Same as problem 33 for the Diesel cycle, $p_2 = p_3 = 500$ lb per sq in. abs.

35. In a Diesel engine, the pressure at the beginning of compression is 14 lb per sq in. abs, the volume 2 cu ft, and the temperature 150 F. At the end of compression the pressure is 500 lb per sq in. abs; the volume at the end of combustion was double that at the end of compression. If $PV^{1.32} = C$ during compression and expansion, find the probable horsepower developed for 150 complete cycles per minute.

Ans. 94 hp.

CHAPTER V

TWO-STROKE CYCLE AND FOUR-STROKE CYCLE GAS ENGINES

Gas engines operate on the Otto cycle with nearly constant-volume combustion; the fuel and air are mixed before or during compression and always depend upon an outside source of heat (electric spark, hot tube, or hot bulb) for igniting the explosive charge at the proper instant, while the charge is under compression. The heat internally developed by combustion gives the working medium a high temperature and pressure; expansion of the gases occurs, and work is done as the piston advances. Various gaseous fuels mixed with air have distinct ignition temperatures and, consequently, to avoid pre-ignition, a different compression should be used for each fuel.

35. Distribution and Utility of Gaseous Fuels.[1] Early gas engines were operated on manufactured or by-product gases, as there were but few places where natural gas was available. For many years, owing to the rapid progress of gasoline and Diesel engines, it was thought that the gas engine would become obsolete, except where by-product gases were available as in steel plants. However, the development of the oil industry made available increased quantities of natural gas, and pipe lines for distributing natural gas have extended through 38 states for a total of nearly 200,000 miles of main and distribution piping. This rapid expansion of pipe lines carrying natural gas to the large industrial centers has made this convenient fuel available at low rates for power generation; in general, it is more economical to transport gas than electricity.

The pipe-line companies employ natural-gas engines to operate the gas compressors necessary to maintain the pipe-line pressure. Oil-well drillers use oil engines for power until the well starts production, then change over to the natural gas which accompanies the oil. Natural gas provides more power when used in a gas engine than under a steam boiler. Plants along the pipe lines frequently install engines that can readily be converted from a gas engine to a Diesel engine, depending upon the economics of the fuel supply. Manufactured-gas plants mix natural gas with their own product, using gas engines for boosters and distribu-

[1] The production and properties of the various gaseous fuels are given in Art. 10.

tion. One steel plant has added waste-heat boilers to absorb heat from the exhaust gases of three large by-product gas engines and thus increased the useful heat to 70 per cent of the total supplied. A number of sewage-disposal plants are using sludge gas in high-compression (6 to 1) gas engines; these provide sufficient power to operate the plant at practically no fuel cost.

36. Two-stroke Cycle Gas Engines. Early gas engine applications had a rapid growth in two fields where commercial gaseous fuels were distributed; in the cities and in the oil fields gas engines replaced small steam engines. The cities were served with expensive manufactured gas or with natural gas that became costly as supplies were threatened and conservation for domestic use became necessary. The great natural-gas supplies had not yet been discovered and no enterprise was available to put millions of dollars into gas transportation facilities. The oil fields had an excess of gas over that required for power and no market for it. The conservation methods of recent years had not been applied. Consequently the industrial centers used four-stroke cycle engines of the most economical type available, but the oil fields used primitive two-stroke cycle gas engines. Mechanically the latter were well constructed but they were by no means as economical of fuel as the four-stroke cycle engine, although far superior to the oil-field boiler and steam engine, and there was no known need of conserving the gas. The petroleum engineer did not realize that a flowing oil well is a natural gas lift, the ascending column of fluid being lightened by formation gas and gas coming out of solution as the pressure on the oil is reduced; and he had not learned to pump this gas back into its natural reservoir to maintain rock pressure, prolonging the natural flow of wells and conserving the natural gas for future use.

Oil fields with depleted gas pressures due to age or exploitation began to be short of natural gas. Whereas Eastern fields felt this shortage after a generation of conservative production, Western fields, extended to the limit to meet the demand for oil, were threatened with gas shortage in a few years. The two-stroke cycle gas engine had established itself as a reliable prime mover; however, it became desirable to improve its fuel economy without the loss of its original advantages of simplicity, compactness, low first cost, and low maintenance.

An internal-combustion engine of the two-stroke cycle type fires once each revolution and is of much simpler construction than the four-stroke cycle engine of the same power. As the two-stroke cycle engine fires twice as often as a four-stroke cycle unit, it requires smaller power cylinders, and the bearing and mean effective pressures are lower. Ports are provided for inlet and exhaust which are covered and uncovered by the piston, thus eliminating troublesome valve gearing and requiring no delicate timing and adjustment of fuel-mixing valves.

The standard two-stroke cycle gas engine of some years ago was usually of crosshead construction with the scavenging air pumped by the back end of the power piston. In Fig. 29 the crankcase is completely and securely enclosed to prevent leakage. During the upstroke of the piston, a partial vacuum will tend to form in the crankcase; thus when the piston uncovers port A a charge of air-gas mixture is drawn into the crankcase. On the downstroke, the piston compresses the charge below it, so that when the inlet port B is uncovered the fresh charge will flow into the cylinder. The great disadvantage of this type of engine is its high fuel consumption due to the gas being mixed with the air as it is drawn into the scavenging chamber and this mixture used to scavenge the power cylinder. Thus, part of the incoming gas always goes out through the exhaust port at C and is wasted.

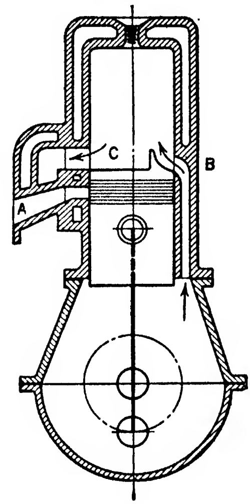


FIG. 29. Elementary two-stroke cycle gas engine with crankcase compression.

In recent years two-stroke cycle gas engines, in sizes up to 300 hp, have been used as the main source of power in oil-field pumping and recompressing plants because of the progress made toward improved scavenging. By injecting the fuel under 10 to 15 lb per sq in. pressure, after the air has been injected and the ports closed, the gas consumption of the two-cycle engine has been reduced to that of the four-cycle engine, namely 10,000 Btu per bhp-hr; see Table XIII. During the past ten years several times as many two-cycle gas engines as four-cycle engines have been installed in sizes up to 300 hp.

Figure 30 shows a section through a vertical two-stroke cycle gas engine cylinder utilizing fuel injection and connected at right angles with a horizontal compressor cylinder. The design of this engine is advantageous from the standpoint of reduced foundation and building requirements. A separate scavenging pump is provided; the compressor crosshead and scavenging pump are combined, as shown. The exhaust and inlet ports are uncovered by the engine piston and loop scavenging is used. The gas injection mechanism consists of a small poppet valve, located in the cylinder head, the opening of which is effected by a rocker arm and push rod operated by a cam on the crankshaft. The valve is arranged to open as the piston is covering the exhaust ports on the compression stroke, so that no gas is lost through the exhaust ports. Governing of the engine is accomplished by a throttle valve controlled by the governor, which regulates the gas allowed to pass to the injection valve according to the speed and load required.

Some of the advantages of the gas-injection two-cycle engine over the standard type of two-cycle engine are, lower fuel consumption and greater power output for a given displacement. The lower fuel consumption is due mainly to the prevention of loss of gas through the scavenging

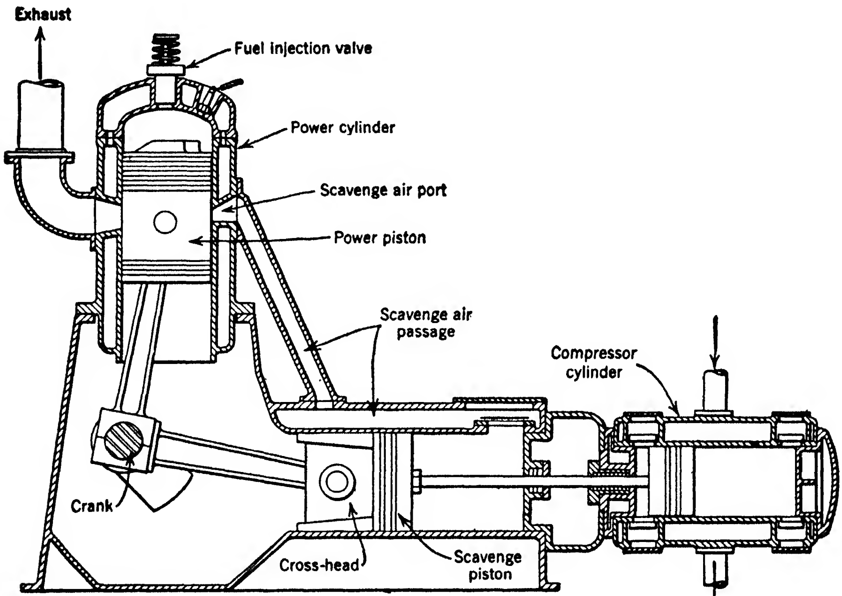


FIG. 30. Clark two-stroke cycle right-angle gas engine driven compressor, $13\frac{1}{2} \times 14$ in., 100 hp per cylinder at 300 rpm.

(exhaust) ports but is, no doubt, helped by the relative turbulence between the gas and air, thus giving more rapid and complete combustion than is possible when the gas and air are drawn in together. As there is no gas mixed with the scavenging air coming into the cylinder, about 10 per cent more air is available for combustion. These two things (greater turbulence and more air) make it possible to obtain at least 10 per cent more power with gas injection than without.

In order to compare the gas consumption and general heat balance of a four-cycle and a two-cycle engine with and without gas injection, tests were conducted on two engines that were identical in every respect except the power cylinder and camshaft mechanism. The engines had a cylinder size of $12\frac{1}{2} \times 20$ in., and were rated at 40 hp as a four-cycle and 60 hp as a two-cycle engine, both operating at a speed of 200 rpm.

The heat balances obtained from the tests are shown in Table XII. For the two-cycle engine operating without gas injection, the item listed as " Heat to radiation, charging, and unaccounted for " is high. This is due to the large charging loss that is inherent in this type of engine, which in this instance runs as high as 30 per cent. In a comparison of the two-cycle and four-cycle engines at their half-load, three-quarters-load,

TABLE XII

HEAT BALANCES FOR SIMILAR TWO-STROKE CYCLE AND FOUR-STROKE CYCLE GAS ENGINES

Percentage of Total Heat Supplied to Gas Engine	Two-cycle, 60-bhp Gas Engine without Gas Injection			Two-cycle, 60-bhp Gas Engine with Gas Injection			Four-cycle, 40-bhp Gas Engine		
	Half load	Three-quarters load	Full load	Half load	Three-quarters load	Full load	Half load	Three-quarters load	Full load
Heat to useful brake horsepower......	10.4	15.4	16.8	17.0	21.6	25.8	16.3	21.1	24.9
Heat to friction....	1.7	1.7	1.3	2.7	2.3	2.1	4.2	3.8	3.5
Heat to compress scavenging air....	1.5	1.5	1.2	2.4	2.1	1.8	.....	....	.....
Heat to cooling water	17.7	17.6	16.2	27.8	26.7	25.4	28.2	27.6	27.4
Heat to exhaust....	26.5	28.4	24.5	41.8	39.3	37.6	44.0	39.8	38.1
Heat to radiation, charging, unaccounted for......	42.2	35.4	40.0	8.3	8.0	7.3	7.3	7.7	6.1
Btu of gas, per cent	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0

and full-load ratings, it will be noticed that the amount of heat charged to the cooling water and the exhaust was slightly less for the two-cycle engines than for the four-cycle engine in each case.

The scavenging air horsepower was unusually low for each type of the two-cycle gas engine. The friction horsepower of the four-cycle engine includes the fluid or pumping losses, which may be as much as one-third of the total; and this is one reason for the lower friction horsepower of the two-cycle engine. Also, at full-load rating the same friction horsepower is not as large a percentage of the total indicated horsepower for the two-cycle as for the four-cycle engine. Both engines have high mechanical efficiencies because of the crosshead construction; this reduces the piston friction materially, which is the largest item in the total friction horsepower.

During the time of the exhaust port opening of the two-cycle engine, the pressure in the exhaust pipe, in the power cylinder, and in the scavenging chamber follow each other very closely. In this way advantage is taken of standardized exhaust conditions to increase the volumetric efficiency of the scavenging end of the cylinder, to insure good scavenging of the power cylinder, and to retain a large percentage of the scavenging

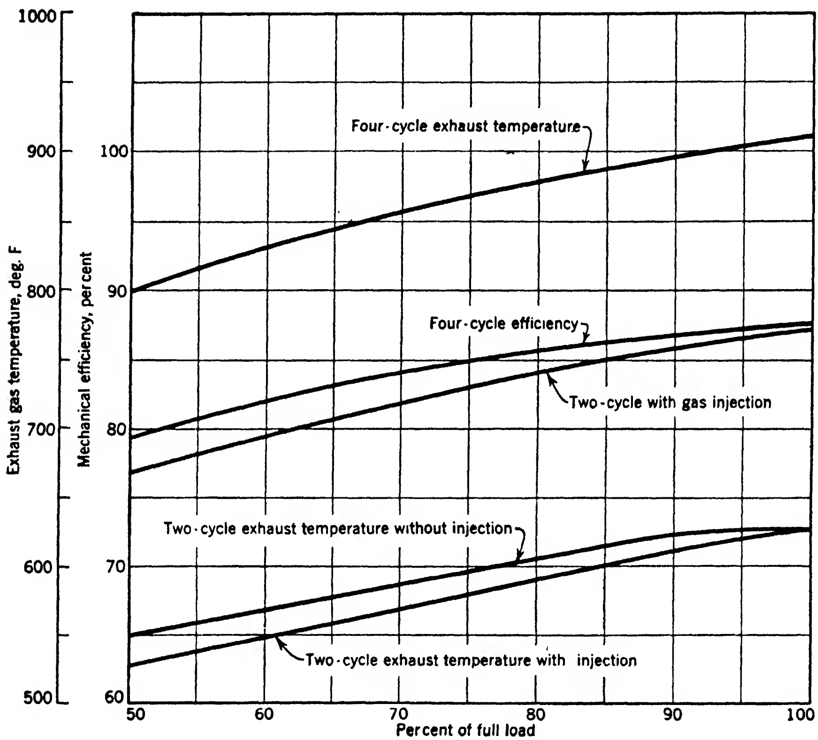


FIG. 31. Variation of exhaust gas temperature and mechanical efficiency with brake horsepower, for similar 60-bhp two-stroke cycle and 40-bhp four-stroke cycle gas engines.

air in the cylinder at the time of port closing. As this type of two-cycle engine is built only in one-cylinder (up to 125-bhp) and two-cylinder (up to 250-bhp) units it is possible to install them so that full advantage may be taken of the increased overload capacity and improved performance due to suitable exhaust piping.

The variations of exhaust-gas temperature, mechanical efficiency, brake-thermal efficiency, and gas consumption with delivered horsepower are shown in Figs. 31 and 32 for the two two-cycle engines and the four-cycle gas engine discussed on the last two pages. The difference between the exhaust temperatures of the four-cycle and the two-cycle engines forms a striking contrast. At the rated load the mechanical efficiency of the engines is very nearly the same, for reasons previously explained. The brake-thermal efficiencies and gas consumption at half-load, three-quarters load, and full-load ratings are very similar for the four-cycle and both the two-cycle engines. For the two-cycle engine without gas

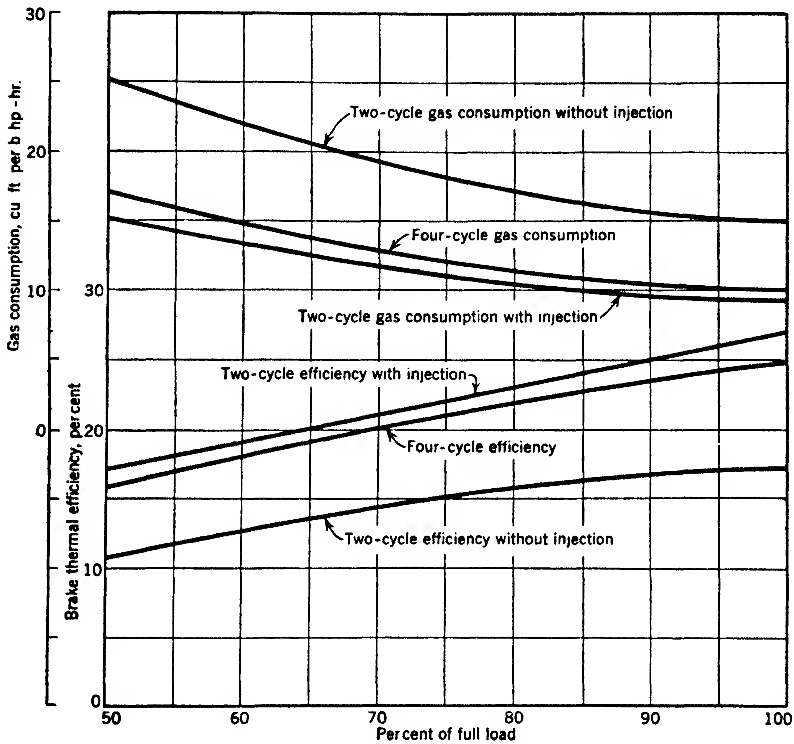


Fig. 32. Variation of gas consumption and thermal efficiency with brake horsepower, for similar 60-bhp two-stroke cycle and 40-bhp four-stroke cycle gas engines; 1000 Btu per cu ft of gas.

injection the efficiency is materially lower and the gas consumption higher, owing primarily to the charging losses, as mentioned before.

When the tests of the four-cycle engine were conducted, the mixing valve was adjusted to give the minimum practicable gas consumption for each load. This was possible because the gas was being metered continuously before entering the engine. There is no mixture adjustment that can be made to the two-cycle engine with gas injection; this insures that minimum gas consumption will be maintained under all operating conditions. The only thing that can be varied is the gas injection pressure; Fig. 33 shows the effect on the gas consumption of varying the gas pressure from 10 to 20 lb per sq in. The effect is apparently very slight, varying not more than 5 per cent. The gas consumption was about 10 cu ft per bhp-hr for this gas-injection pressure range. Figure 33 also shows the effect on the gas consumption of the four-cycle engine by setting of the mixing valve. The gas consumption was plotted against half turns of opening of the mixing valve. As shown by the curve, the gas consumption could be varied from 10 to

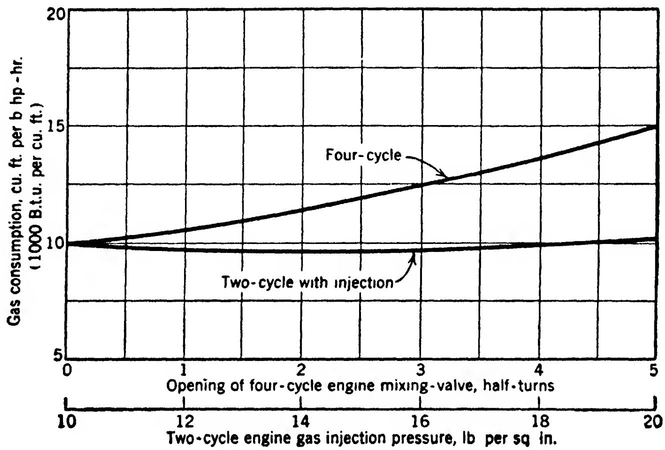


FIG. 33. Variation of gas consumption which can be effected by the operator for similar 60-bhp two-stroke cycle and 40-bhp four-stroke cycle gas engines.

15 cu ft per bhp-hr with the engine, to all appearances, running satisfactorily.

37. Four-stroke Cycle Gas Engines. The construction of a typical four-stroke cycle gas engine is shown by the vertical engine illustrated in Fig. 34. The inlet and exhaust valves are in the cylinder head. The valves are of the poppet type, mechanically opened by valve gearing and closed by a spring. The valves are operated with minimum inertia from cams on the camshaft through short balanced levers and a "roller path" motion. In this device the lever is equipped with a roller at each end, one of which rests on the valve stem and the other rides directly on the actuating cam. The lever is provided with an eccentric bushing at the fulcrum point which allows take-up for wear in the valve mechanism. The eccentric may be turned to adjust clearances while the engine is in operation.

The valves operate in removable guides. The inlet valve seats directly against the cylinder head; the exhaust valve seats against a removable and renewable seat of heat-resisting alloy. The cylinder heads, cylinders, exhaust ports, and exhaust manifold are simple single-walled castings cooled by immersion in water contained in a surrounding casting; the water overflows through pipes of adjustable height.

As in practically all such engines, the engine is single-acting and has a trunk piston, which also acts as a crosshead. The engine is made single-acting because the piston, piston rod, and stuffing box would give

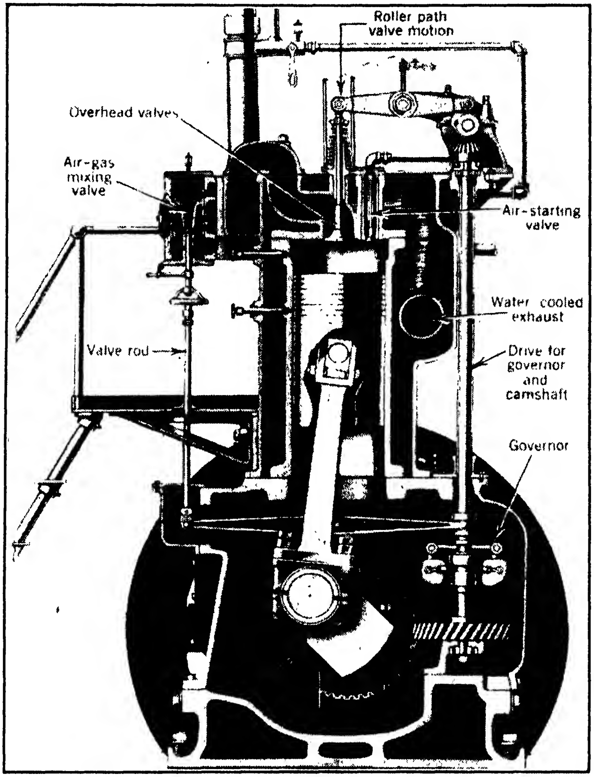


FIG. 34. Vertical section of Bruce-Macbeth four-stroke cycle gas engine.

much trouble if exposed to the high temperature of the burning gases unless the parts were water-cooled; and the water-cooling of these parts is difficult in small engines. The piston is made of heat-treated cast iron and is provided with an air-circulating space or pocket at the top to reduce the accumulation and carbonization of oil on the underside of the hot piston head. Four cast-iron piston rings are provided, the lower of which is of the oil-control type.

The main journal boxes of the engine shown in Fig. 34 rest on wedges, adjustable from the outside by bolts which extend through the engine, thus providing means for maintaining crankshaft bearing alignment. The crankcase is provided with hinged doors, giving easy access to all parts for inspection and adjustment. Stay bolts are provided to transmit the strains created by the varying pressure in the cylinders to the section of the crankcase carrying the main journals.

The governor is of the centrifugal fly-ball type and is mounted inside the crankcase on the vertical shaft which drives the camshaft. The mixing valve is of the throttling type, and is designed to provide the proper air-gas mixture at all load conditions. The turning of the mixing valve handle varies the proportion of air to gas, and when this proportion is once set it requires no further adjustment; the resulting air-gas mixture is then throttled by the governor. A hand wheel, see Fig. 34, on the mixing valve permits variation of the engine speed without stopping the engine.

Two separate and distinct electric ignition systems of the jump-spark type are provided, either one of which will start as well as operate the engine. Starting is accomplished by means of compressed air admitted to each cylinder in firing order.

These engines are built in two- and four-cylinder units, ranging in size from 40 to 400 bhp. Gas of practically any heat value may be used, from producer gas at 125 Btu per cu ft, through manufactured gas, sludge gas, and natural gas, to propane and butane at more than 3000 Btu per cu ft. The compression of the engine is varied according to the type of fuel used, and for any particular engine the heat value of the gas should be maintained near the specified (designed) value; variations of more than 15 per cent are undesirable.

The leading characteristics of representative gas engines recently manufactured in the United States are listed in Table XIII. The larger engines, 400 hp and more, operate on the four-cycle principle; for units of 300 hp and less, several times as many two-cycle as four-cycle engines have been installed in recent years. Many of the large gas engines employ double-acting cylinders to reduce weight, cost per horsepower, and floor space. These double-acting designs are more easily built as a gas engine than as a Diesel engine because the larger clearance volume makes it possible to install inlet and exhaust valves of proper size without restricting or distorting the clearance space. Many of these large gas engines are horizontal rather than vertical for accessibility and for adaption to cylinders in tandem with a simple valve gear, there being only one gear-driven crankshaft or eccentric shaft. Tandem cylinders are necessary to gain full advantage of the horizontal, double-acting type. The supply pipes for gas and air and the exhaust pipes are located beneath the floor.

The horizontal slow-speed engine is ideal for direct-connected recipro-

TABLE XIII

DOMINANT CHARACTERISTICS OF RECENT AMERICAN GAS ENGINES *

Location	Service	Brake Horse-power	Rpm	Cylinder Data			Cycle (stroke)	Gas Used	Btu per cu ft of Gas	Btu per bhp-hr
				No.	Size	Arrangement				
Haven, Kan.	Gas Comp.	1300	125	4	24 × 36	H–TT	4	Natural	1000	10,000
Washington, D. C.	Generator	1200	225	6	22 × 26	V–S	4	Sludge	600	8,400
Tulsa, Okla.	Gas Comp.	1000	125	4	$22\frac{1}{2}$ × 36	H–TT	4	Natural	1000	10,000
Dallas, Tex.	Generator	600	240	6	16 × 20	V–S	4	Natural	1000	10,000
Peoria, Ill.	Blower	535	327	6	15 × $17\frac{1}{2}$	V–S	4	Sludge	550	9,500
Marcus Hook, Pa.	Pump	500	1200	8	8 × 9	V–S	4	Refinery	1600	10,000
Tuscumcari, N. M.	Generator	420	400	8	$11\frac{1}{2}$ × $13\frac{1}{2}$	V–S	4	Natural	1000	10,000
Muskegon, Mich.	Gas Comp.	400	200	2	19 × 20	H–Tw	4	Natural	1000	10,000
Brea, Calif.	Alternator	300	500	6	9 × 11	V–S	2	Natural	1000	10,000
McConnellsburg, Pa.	Pump	300	800	8	8 × 9	V–S	4	Natural	1000	10,000
Braddock, Pa.	Pump	235	200	2	17 × 20	H–Tw	2	Refinery	1500	10,000
Tulsa, Okla.	Gas Comp.	230	200	2	18 × 20	H–Tw	4	Natural	1000	10,000
Madison, Wis.	Blower	230	514	8	9 × $10\frac{1}{2}$	V–S	4	Sludge	600	10,000
Los Angeles, Calif.	Pump	200	550	4	9 × 11	H–S	4	Sludge	600	12,000
Springfield, Ill.	Blower	177	514	6	9 × 11	V–S	2	Sludge	600	12,000
Chicago, Ill.	Pump	150	180	2	$14\frac{1}{2}$ × 20	H–Tw	2	Natural	1000	10,000
Durham, N. C.	Generator	150	360	4	$10\frac{1}{2}$ × $13\frac{1}{2}$	V–S	4	Sludge	750	11,000
Gladewater, Tex.	Gas Comp.	115	200	1	17 × 20	H–S	2	Natural	1000	10,000
Dallas, Tex.	Pump	100	300	2	11 × 15	H–Tw	2	Natural	1000	10,000
Los Angeles, Calif.	Generator	67	1000	6	5 × $5\frac{1}{2}$	V–S	4	Natural	1120	11,200
Tulsa, Okla.	Pump	40	250	1	$12\frac{1}{2}$ × 16	H–S	2	Natural	1000	10,000

* *Power*, page 327, June, 1936.
Abbreviations: H, horizontal; V, vertical; S, single; Tw, twin; TT, twin-tandem.

cating compressors, the opposed-type connection giving fewest parts and minimum floor space. The vertical cylinder arrangement is better for direct connection to electric generators because the speed is higher. The horizontal twin-tandem double-acting engine, because of large reciprocating parts, is generally used for compressor drives only and seldom runs over 125 rpm, whereas the vertical will run as high as 225 rpm in an equivalent size.

The world's largest gas engines are installed at the South Chicago plant of the Illinois Steel Company. These engines, built by the Allis-

FIG. 35. Gas compressor station with 5 four-stroke cycle twin-tandem double-acting gas engines driving single-stage compressors for boosting natural gas pressure from 165 to 450 lb. Engines use water-cooled floating pistons.

Chalmers Manufacturing Company, were installed in 1930. They are horizontal, four-cylinder, twin-tandem, double-acting, four-cycle, 11,000-bhp, and each is direct-connected to a 6600-kw, 25-cycle, 2300-volt generator running at $83\frac{1}{3}$ rpm. The cylinders are 60 × 64 in. and the engines operate on blast-furnace gas with an over-all thermal efficiency of 26.25 per cent.[2] Figure 35 shows the interior of a gas-compressor station containing five 1200-bhp, horizontal, four-cycle, twin-tandem, double-acting gas engines direct-connected to $13\frac{1}{2} \times 36$ in. single-stage compressors for compressing gas from 165 to 450 lb pressure. The engines use water-cooled floating pistons with tail guides, also cam-driven valve gear with outside lay shaft as shown. The largest vertical gas engines built in this country thus far are 1800-hp, 225-rpm units.

[2] For further details see "The World's Biggest Gas Engines," *Power*, page 736, May 13, 1930.

These twelve-cylinder engines have 6 cylinders on each side of the generator and have shown a heat economy of 9700 Btu per bhp-hr.

High-speed (more than 500 rpm) vertical gas engines are a composite of automotive and Diesel designs. They are usually four-cycle, single-acting engines with double overhead exhaust and inlet valves. Removable cylinder liners, trunk pistons, automotive-type connecting rods, full-pressure lubrication, oil filters and coolers, etc., are standard. Water jackets may be individual or cylinders are jacketed in groups; jacket water may be circulated at higher velocity at liner tops and liner supporting shoulders are moved down, both to increase the rate of heat transfer. Gas engines, both vertical and horizontal, are built in almost

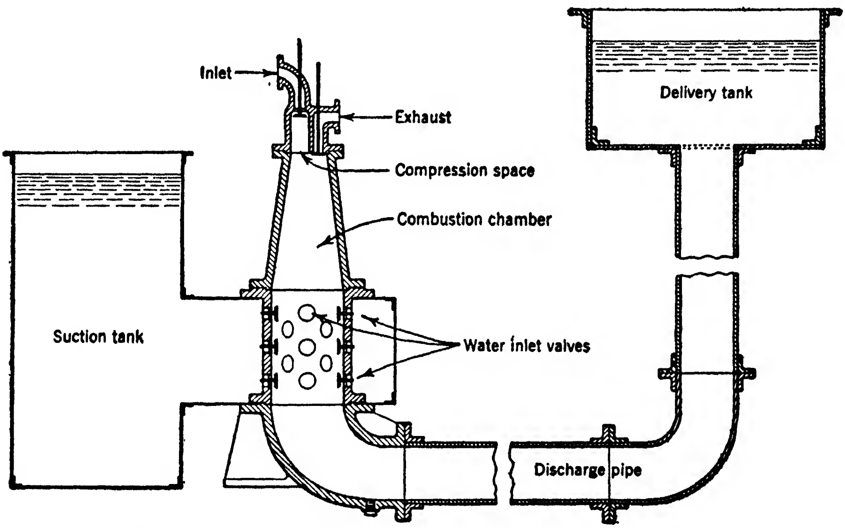


FIG. 36. Diagrammatic arrangement of the Humphrey four-stroke cycle gas pump.

any desired minimum size. These small engines usually have integral radiators and lubricating systems similar to those used in gasoline engines, and generally can be converted from gas to gasoline, and vice versa, merely by replacing the gas mixing valve with a gasoline carburetor.

38. Humphrey Gas Pump. One of the most remarkable applications of the internal-combustion principle is used in the gas pump invented by H. A. Humphrey of England in 1909. Its chief feature is the use of a moving column of water in place of a piston and connecting rod, the upper face of the water column taking the place of the top of the piston. The four-cycle pump in its original form is shown diagrammatically in Fig. 36. Suppose a charge of gas and air to be compressed at the top of the combustion chamber in the space shown and let it be fired by a spark plug, not shown in the diagram. The explosion pressure drives the water out of the combustion chamber and sets the whole water column in the discharge pipe in motion. As the gases expand the kinetic energy of the moving water steadily rises, until the gases reach atmospheric pressure. There is then no further driving force on the water column, but because of its kinetic energy continues to move forward and the pressure in the combustion chamber falls below atmospheric pressure, thus opening the exhaust valve and the water inlet valves. Water rushes from the suction tank into the combustion chamber and into the discharge pipe. When the kinetic energy of the moving column has entirely expended itself in forcing water into the delivery tank and in overcoming friction, it comes to rest and gradually starts to swing back again until it rises to the upper part of the combustion chamber and closes the exhaust valve by impact. Its further motion is stopped by the cushioning effect of the burnt gaseous products trapped in the compression space. Gradually the water column begins to move downward again, opening the inlet valve and drawing in a fresh explosive charge which on the return swing of the water is compressed, fired by the spark plug, and the cycle repeated. Thus the full thermodynamic value of the combustible charge is realized in a mechanism of extreme simplicity. Remarkable as may be the thermal efficiency of 26.63 per cent obtained on a pump delivering 28,000 gpm against a head of 27.5 ft, repeated efforts have been made to increase the capacity and discharge head of the four-cycle gas pump.

Recently an improved form of Humphrey pump using the two-stroke cycle principle was devised by A. P. Steckel.[3] It has more than doubled the output of the four-cycle type, both being of the same size. Without resorting to auxiliary pumps for scavenging and charging but merely by interposing a diaphragm in the form of a light plate piston fitting loosely in the combustion chamber between the fresh charge and the exhaust gases, the two-cycle pump can handle these gases more perfectly than the four-cycle type.

Figure 37 shows a diagrammatic arrangement of the two-cycle Steckel-Humphrey gas pump. The pump is making the compression stroke with the diaphragm and attached plunger *P* moving upward; when the diaphragm *D* reaches its uppermost position against the cylinder head, the explosive charge is compressed between the underside of *D* and the water surface. With valves closed and everything stationary, the charge is ignited; the resulting expansion imparts energy to the column of water *M*. When atmospheric pressure is reached, the

[3] "The Humphrey Gas Pump," F. du P. Thomson, *Mechanical Engineering*, page 337, June, 1934.

exhaust valves E are opened by light springs; and the diaphragm, being no longer held up by internal pressure, descends to the level of the exhaust valves, pushing the exhaust gases out below it and drawing in a new charge above it through the air and gas valves A and G. The expanding gases of the power stroke having lowered the level of the water inside the pump below that of the suction well S, the inlet valves I now open and water enters and rises to approximately the level of S. Diaphragm and water surface thus meet at the exhaust-valve belt and practically all the exhaust gases are expelled. In the meantime the moving column of water, having expended all of its energy in discharging water through the discharge valves O, comes to rest and then starts to return under the discharge head. Water now starts to go out through the exhaust valves E, which thus become clogged and close gently; compression then begins.

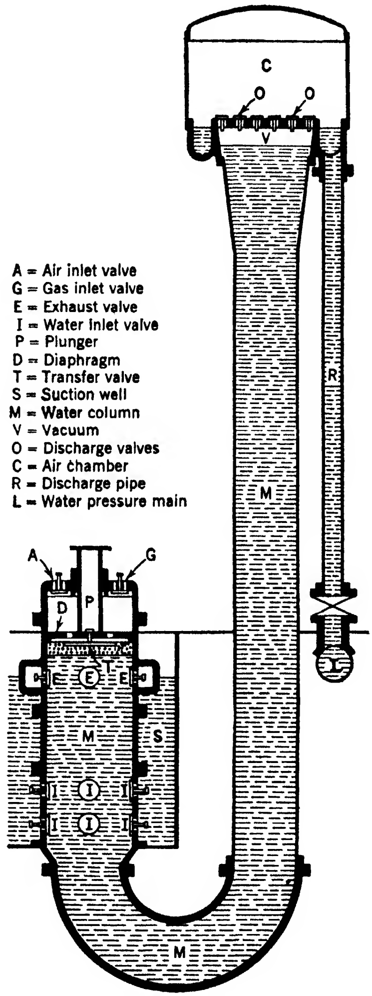


FIG. 37. Two-stroke cycle arrangement, Steckel-Humphrey gas pump.

The combined weight of plunger P and diaphragm D is sufficient to support this unit when subjected to a pressure considerably less than the static head of water now acting upon it. As the compression rises, the force available is sufficient to cause the diaphragm plunger to move upward away from the advancing water surface and transfer the air-gas charge taken in on the downstroke through the transfer valve T to the underside of the diaphragm. The diaphragm is thus accelerated while the water column is retarded, with the result that slightly before the column has come to rest the diaphragm has reached the upper limit of its stroke and is pressed firmly against the cylinder head. It is prevented from slamming against the head by a dashpot device located inside the plunger, not shown in Fig. 37. Upon completion of the upward stroke, the plunger P touches a timer, ignition takes place, and the cycle is repeated.

In order that complete expansion to atmospheric pressure may take place it is necessary that the discharge head shall not exceed the equivalent of the mean effective pressure in the gas cylinder. For this reason, when pumping against heads in excess of 60 ft, a space V is provided under the discharge valves in the form of a vacuum whose volume is equal to that of the gaseous charge when expanded to atmospheric pressure. When V has become completely filled, discharge into air chamber C takes place, and water flows through discharge pipe R to the water pressure main L. When the energy of column M has become exhausted, the water recedes from the discharge-valve deck, leaving the vacuum available for the next stroke. The height of this discharge valve deck above suction level may therefore be greater than the head required for compression by that head which is equivalent to the amount of vacuum obtained.

Gas pumps designed to operate against heads between 30 and 60 ft require no discharge valves because the mep is more than sufficient to maintain discharge during the entire expansion stroke. A set of check valves is required in the column, which, while permitting free flow of water on the power stroke, also throttles the flow on the return stroke to that head which will give the desired compression. Pumps intended to operate under heads of 30 ft or less need have no obstructions of any kind in the discharge water column. It is believed that installations of this type can be made which, for over-all cost of pumping water, will compare favorably with that of any other type now in use.

CHAPTER VI

AUTOMOBILE AND AIRCRAFT ENGINES

Automobile and aircraft engines are the most highly perfected types of internal-combustion engines. It is doubtful whether anything created by man at any time has been the subject of such intensive research or has undergone such revolutionary changes in so short a time as have the modern automobile and airplane, and those things essential for their successful operation. Since the World War we have seen the engine ratings increase about four times, along with comparable advances in fuels, economy, weight reduction, higher speed, more rapid acceleration, safer transportation, less maintenance, and many other advances. No less remarkable has been the general acceptance of the automobile and airplane as modern necessities, and the dexterity and confidence with which the average motorist and aviator use the power placed at their disposal.

39. Automobile Engines. The engine used almost exclusively for automobiles is of the vertical multi-cylinder type operating on the Otto four-stroke cycle. The peculiar demands made by the public have made necessary the building of engines that will withstand considerable abuse. Lubrication, carburetion, cooling, ignition, etc., must be simple and unfailing for long periods of time without any attention. Automobile engines are designed to run at speeds of 3000 rpm or more, with short strokes, jump-spark ignition, mechanically operated valves, using gasoline as a fuel and employing a carburetor to atomize the gasoline and mix it with the proper amount of air before the fuel enters the cylinders.

Table XIV shows the dominant design and performance characteristics of American automobile engines manufactured for passenger-car use, 1925–1936, inclusive. The figures given are the average of values of all American passenger cars listed in trade publications. This table shows the increase in maximum horsepower of American automobile engines during this time and the changes made in the engine design to produce this increased power. Greater power may be obtained by one or more of three methods: greater engine displacement, higher engine speed, or improved brake mean effective pressure. The power output per cubic inch of displacement depends entirely upon engine speed and

TABLE XIV

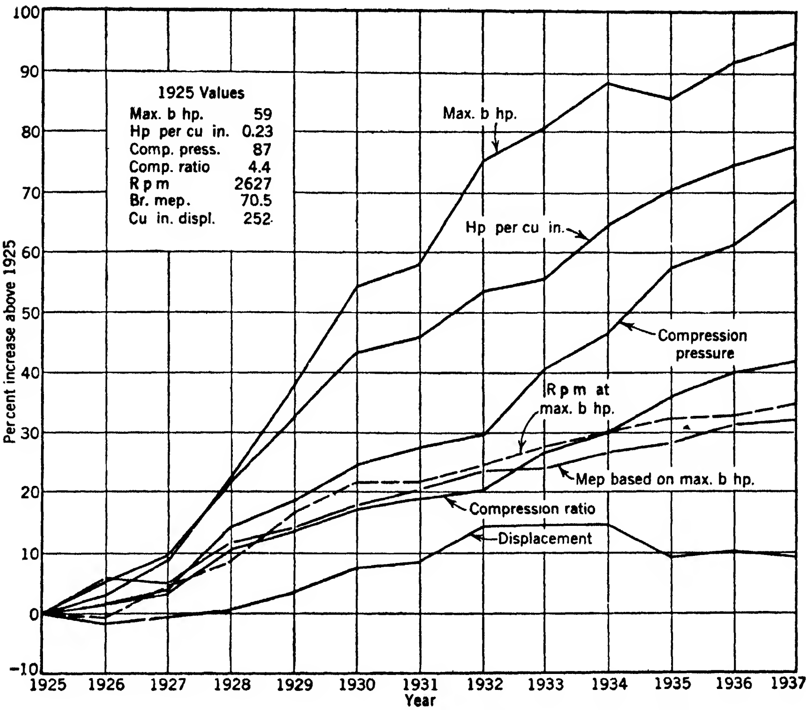
DOMINANT DESIGN AND PERFORMANCE CHARACTERISTICS OF AMERICAN PASSENGER-CAR AUTOMOBILE ENGINES *

Year	Maximum Brake Horse-power	Bhp per cu in.	Rpm at max. bhp	Mep Based on max. bhp	Cubic-inch Dis-placement	Com-pression Ratio	Com-pression Pressure
1925	59 05	0.234	2627	70.5	252.1	4.40	87.0
1926	60.67	0.246	2609	74.6	247.4	4.47	88.5
1927	64.26	0.256	2744	74.0	250.4	4.55	90.5
1928	72.18	0.285	2862	78.8	253.5	4.86	99.5
1929	81.37	0.311	3071	80.4	260.9	4.99	103.5
1930	90.97	0.336	3201	83.2	270.7	5.15	108.5
1931	93.12	0.342	3195	84.7	272.7	5.23	111.0
1932	103.50	0.359	3268	87.1	288.0	5.29	113.0
1933	106.49	0.364	3346	87.4	288.4	5.57	122.5
1934	111.21	0.384	3415	89.3	289.1	5.72	127.5
1935	109.70	0.399	3475	91.0	275.0	5.99	138.5
1936	113.20	0.408	3492	92.5	277.6	6.16	140.0

* Compiled by the Ethyl Gasoline Corporation Engineering Laboratory.

brake mean effective pressure. The average increase in horsepower since 1925 has been 88 per cent, see Fig. 38, of which increase 42 per cent has been contributed by increased engine speed, 20.7 per cent by greater displacement, and 37.3 per cent by improved brake mean effective pressure. The latter, which is a direct measure of thermal efficiency, has been effected by improved volumetric efficiency and higher compression ratios made possible by the availability of fuels of higher anti-knock value.

The continued increase of power and other dominant characteristics of passenger-car engines, as indicated by Fig. 38, are due primarily to the demand of passenger car owners for improved performance and maneuverability under a wide range of conditions. Motorists found, without perhaps an extensive analysis of the problem, that more power permitted homes to be built farther from their business activities, increased the range of vacation travel, and contributed materially to greater comfort and convenience. Competition in the passenger-car manufacturing industry has made it necessary for engineers to satisfy these desires and efficient research organizations have indicated the best methods to improve engine performance. The large increase in power has been effective with very little increase in engine displacement because of the more efficient use of metal, space, and fuel. Research along these lines is still in progress and further improvement in per-



Courtesy of *Ethyl Gasoline Corporation.*

FIG. 38. Trends of American passenger-car engine design since 1925. Average of values listed in trade publications.

formance may be expected with little or no increase in engine displacement.

Figures 39 and 40 show the essential parts of a typical eight-cylinder L-head automobile engine. The cylinders and upper crankcase are cast *en bloc* (all the cylinders in a single casting); to the bottom of the cylinder casting is bolted the crankcase or oil pan, which is a pressed steel stamping. The engine has a detachable aluminum cylinder head and employs aluminum-alloy pistons. The poppet valves (an inlet and an exhaust valve for each cylinder) are placed on the side, this being an L-head engine, and are operated by a single camshaft. The camshaft is designed to open and close all the valves at the proper time, so as to give the proper sequence of events in the eight cylinders. The head stems of the exhaust valves, to about two-thirds of the way down, are extruded from a special high heat-resisting austenitic steel; the intake valves are extruded from single pieces of chrome-nickel steel; the lower ends of the valves are case-hardened to give long life where they come in contact with the tappets.

The valve seats are at a slight angle, see Fig. 40, and slope inward to the combustion chamber about $6\frac{1}{2}$ degrees. This allows for a more compact combustion chamber and increases breathing capacity and volumetric efficiency. The head of the intake valve is fitted to a 30-degree angle valve seat in the block, for improved charging efficiency, whereas the head of the exhaust valve is fitted to a 45-degree angle seat to insure proper cooling. The tension of the valve springs is 40 lb when closed and 110 lb when open. The tappets or valve lifters are of the mushroom type. They are cast separately from a special alloy iron, which, when chilled on the lower end gives a greater degree of surface hardness where

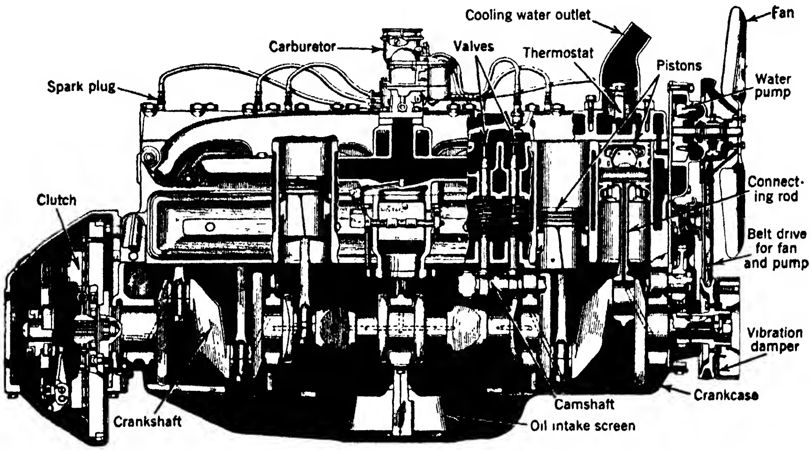


FIG. 39. Side sectional view of Packard automobile engine.

the tappets come in contact with the cam surfaces. It is to be noted that separate intake- and exhaust-valve guides of cast iron are pressed into the cylinder block.

The inlet valve of any given cylinder opens 1 degree before the piston reaches top center on the suction stroke, see Fig. 41, then as the piston moves downward the mixture of gasoline and air is drawn into the cylinder from the carburetor. The inlet valve closes 39 degrees after the piston reaches bottom center; as the piston rises the mixture is compressed with both valves closed. Compression serves to mix more evenly the gasoline vapor with the air; the contour of the inside of the cylinder head, Fig. 40, also aids this mixing. Just before the beginning of the next downward stroke the explosive mixture is ignited by an electric spark from a spark plug located in the top of the cylinder. The combustion resulting from this ignition increases the pressure from 130 to 400 lb per sq in. On the expansion or power stroke the piston is forced downward and by means of the connecting rod the driving force is transmitted to the crankshaft of the engine. As the engine has eight cylinders, the torque on the crankshaft is more uniform—there being four power impulses every revolution—than if fewer cylinders were employed. On the next upward stroke of the piston the exhaust valve is open, having opened 45 degrees before the piston reached bottom center, and the products of combustion are exhausted to the atmosphere, and the cycle of events—intake, compression, expansion, and exhaust—is repeated. The exhaust valve remains open until 5 degrees after top center of piston travel, thus the intake and exhaust valves are both open for a period of 6 degrees crankshaft angle; see Fig. 41. The energy of the exhaust gases leaving the cylinder at high velocity will create a reduction of pressure and during the interval when both valves are open will aid in cooling the cylinder, purging it of exhaust gases, and improving the volumetric efficiency.

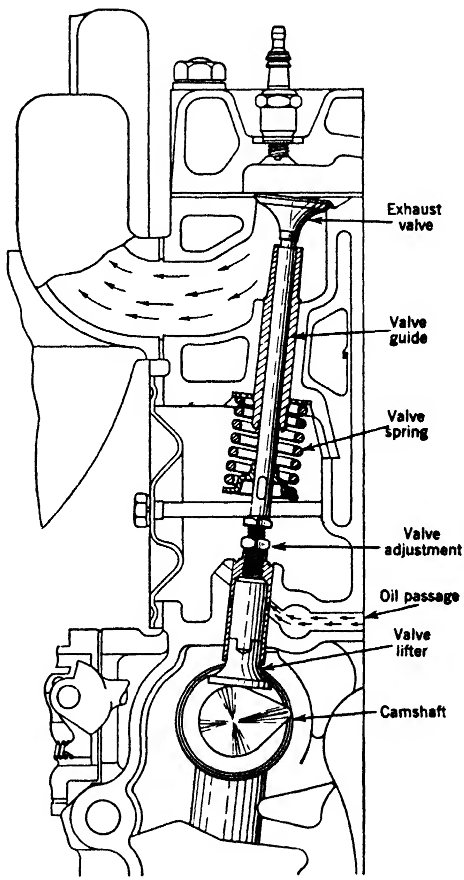


Fig. 40. Packard valve-operating mechanism.

In order to make the explosions in an engine cylinder as effective as possible, the ignition of the compressed mixture should occur a trifle before the piston reaches the top dead center. Ignition should take place, therefore, just before the end of the compression stroke. The faster the engine is running, the earlier in the stroke the ignition should occur, so as to insure complete combustion. That is, the "spark" should be further advanced (occur earlier) at high than at low speed.

The crankshaft is provided with five main bearings, $2\frac{3}{4}$ in. in diameter. Forced lubrication is supplied by means of a gear-type oil pump which has a capacity of 1600 quarts per hour at a speed of 35 mph and is driven by the camshaft. A notable feature of the lubrication system is the elimination of all possible tubing; there is only one short pipe and it leads from the inside of the oil intake screen to the oil pump; oil passages are drilled in the walls of the crankcase. Oil is forced under pressure to all main, connecting-rod, camshaft, and piston-pin bearings. The crankshaft is drilled to provide oil passages from the main bearings to the crankpin or connecting-rod bearings. The I-beam

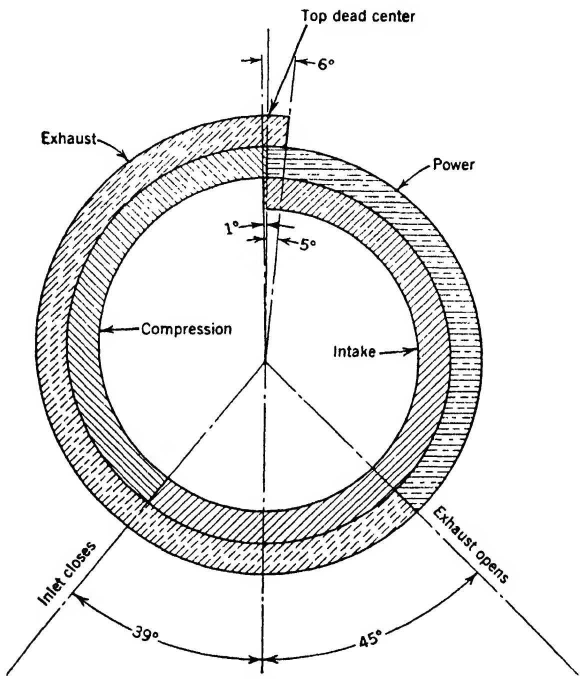


FIG. 41. Valve-timing diagram and cycle of operation for Packard automobile engine.

shaped connecting rods are rifle-drilled lengthwise from end to end so as to furnish pressure lubrication to the piston (wrist) pin bearings.

The cylinder walls are lubricated by oil thrown from the lower connecting-rod bearings. Lubrication of the cylinder walls by oil spray is quite satisfactory in warm weather or after the engine and oil become heated. However, when the engine is started in cold weather very little oil is thrown on the cylinder walls for some time and aluminum pistons may become scored. Some car builders advise running the engine fairly fast when warming up so as to throw oil on the cylinder walls as soon as possible, the contention being that if the engine is running very slowly no oil will be thrown on the cylinder walls because of reduced oil pressure and higher viscosity of the cold oil.

The engine shown in Fig. 39 uses steel-strutted aluminum-alloy pistons of light weight and long skirt. This material is used because of its light weight, which is an important factor in reciprocating parts. Lightweight pistons insure smoother operation at high speed, and allow for better acceleration with reduced bearing loads. Aluminum dissipates heat more than three times as rapidly as cast iron, hence there is less carbon formation and higher compression ratios may be employed. Each piston is fitted with two $\frac{1}{8}$-in. grooved oil-seal compression rings and one $\frac{3}{16}$-in. ventilating oil ring, all placed above the piston pin center.

A plain-tube, multiple-jet type of 1-in. dual downdraft carburetor is used, similar to the carburetor shown in Fig. 5. This carburetor is really two carburetors built into one unit: it employs a single float bowl, single air intake, single choke valve; but double venturi, double set of jets, double throttle, double idle adjusting needles, double risers, and a pump discharge nozzle for each barrel. The throttles are carried on a single shaft in the carburetor body, and the carburetor is mounted on a $\frac{3}{8}$-in. asbestos composition riser pad. This pad is an effective insulator against heat and prevents the direct conduction of heat from the hot manifold to the carburetor and float chamber, thus reducing the tendency to vapor-lock and also aiding starting when hot.

Cooling of the automobile engine is effected by a fan and water jackets. Water for cylinder cooling is circulated through the water jackets and radiator by means of a small rotary or centrifugal pump, located at the rear end of the fan shaft, Fig. 39. The cooling-water circulation may be controlled by a thermostatic valve, Fig. 39. When the engine is cold, there is no circulation to the radiator (top connection); when the water in the jackets becomes heated to 145–150 F, the valve opens and the circulation is through the radiator.

Sectional views of a typical valve-in-head automobile engine are shown in Figs. 42 and 43. The five crankshaft bearings are stepped in size from front to rear to facilitate machining the crankcase and to provide more uniform rigidity. Oil under pressure is provided by a gear-type pump driven from the camshaft through spiral gears. Oil is forced to all crankshaft bearings, connecting-rod bearings, camshaft bearings, timing chain, and rocker-arm shaft and bushings. The cylinder walls, pistons, and piston pins are lubricated by oil forced through the passages in the crankshaft and lower end of connecting rod; see Fig. 43. When the engine is in operation and these passages are in line, oil is sprayed upward against the cylinder walls, when the piston is near the top of its stroke, thereby completely lubricating pistons, rings, and pins. The surplus oil wiped from the cylinder wall by the lower oil ring is forced through a drilled passage in the piston, see Fig. 43, thus providing lubrication to the piston-pin bearings.

Both inlet and exhaust valves are located in the detachable cylinder head. The inlet valves have a clearance diameter of $1\frac{5}{8}$ in., the exhaust valves $1\frac{9}{32}$ in., valve seat angle 45 degrees, and a lift of 0.348 in. The streamlined inlet valve is made of chrome-nickel steel and the exhaust valve is made of silchrome No. 1 steel. Valves in the cylinder head require overhead rocker arms and pushrods. The rocker-arm assembly is shown in Figs. 42 and 43. The hollow rocker shaft is supported and held in place on the cylinder head by spacer and stud supports. All

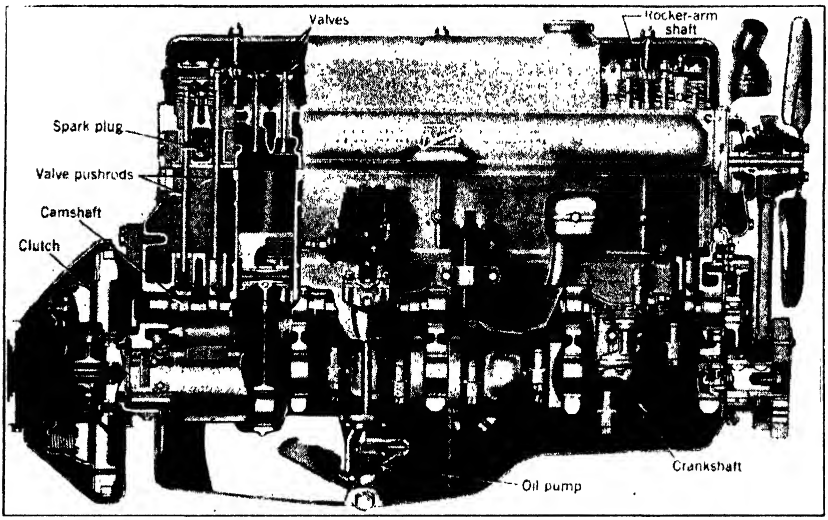


Fig. 42. Side sectional view of Buick automobile engine.

rocker arms are lubricated from the pressure system through direct tubing to the hollow rocker shaft, thus supplying not only the rocker-arm bearings with oil but also the springs and pushrod ball- and socket-surfaces. The valve tappets are adjusted by means of adjusting screw and lock nut to give a clearance of fifteen thousandths of an inch when hot. Double valve springs are used to avoid periods of spring vibration which might occur if only one spring were used.

The pistons, made of a special aluminum alloy, are given an electrolytic treatment which oxidizes the surface, thus making it hard and slightly porous so that oil adheres to it even under extreme conditions of heat or cold. The head of the piston has a special dome shape, see Fig. 43. The combustion chamber formed by the space between the piston head and cylinder head makes possible the use of higher compression ratios by affording improved turbulence of the compressed cylinder charge. Four piston-ring grooves are provided in the head of each piston with an additional heat-deflecting groove in the top land to reduce the heat carried to the top ring. The top groove of the piston

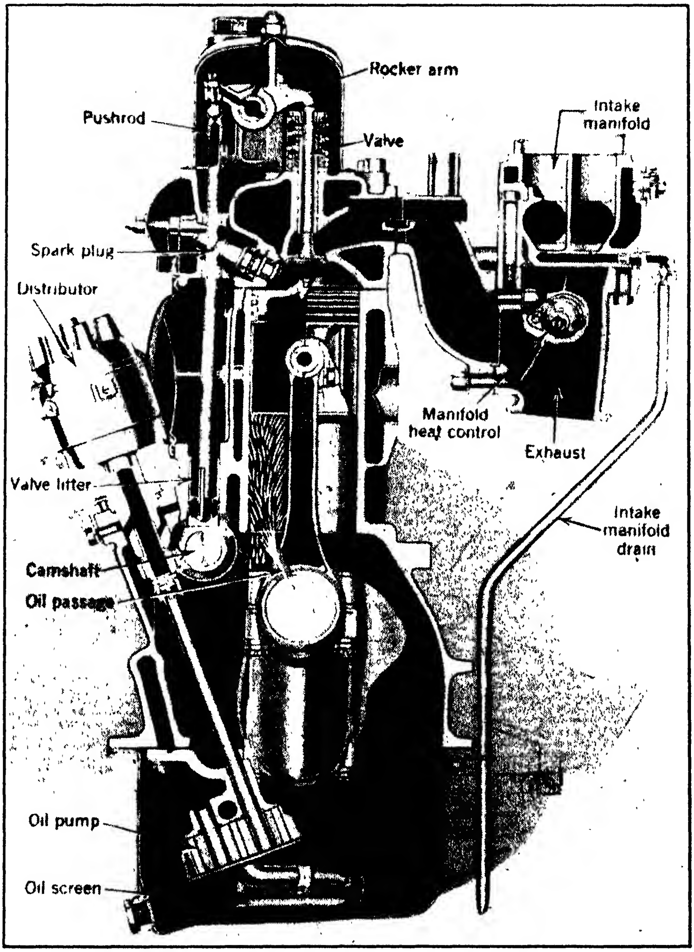


FIG. 43. End sectional view of Buick automobile engine.

carries a $\frac{1}{8}$-in. compression ring; a relief is machined on the upper inner diameter of this ring. A $\frac{3}{32}$-in. compression ring is used in the second groove. The two lower grooves are fitted with $\frac{3}{16}$-in. oil-control rings of the channeled high unit-pressure type. Each of these grooves is provided with ten $\frac{5}{32}$-in. return holes. The oil tends to work in around the back of the rings and thus travels up into the combustion chamber.

This is prevented by the oil drain holes which allow the oil to escape to the inside of the piston and return to the crankcase. The piston pin is clamped in the upper end of the connecting rod and floats in the piston bosses.

This engine employs a dual downdraft carburetor and a thermostatically controlled by-pass type of water-temperature control similar to that which has been previously explained in this article. The cylinder walls of the engine are completely surrounded by water and the water jackets extend below the head of the pistons when in their lowest position.

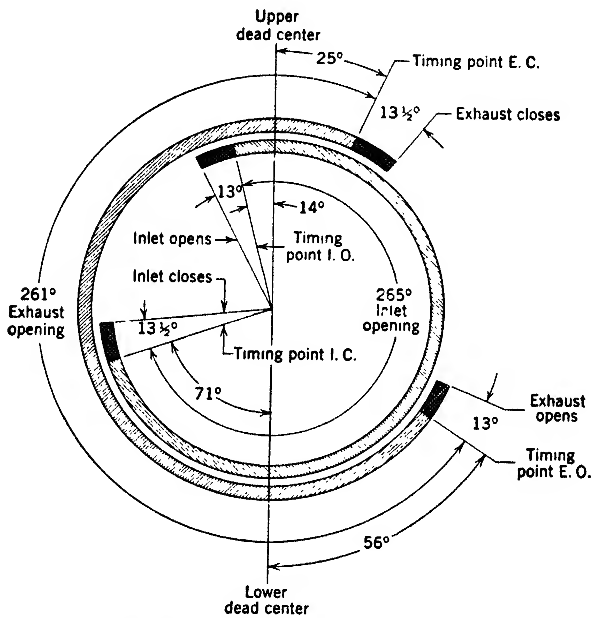


FIG. 44. Valve-timing diagram and cycle of operation for Buick automobile engine.

The valve stems and valve seats are water-cooled for their entire circumference. An oil temperature regulator is provided inside the rocker-arm shaft to control the temperature of the oil supplied to the valve mechanism maintaining the lash at a uniform clearance. This device consists of a water tube through the valve rocker-arm shaft which carries water from the rear of the cylinder head to the water pump. Oil is fed into the space between this tube and the valve rocker-arm shaft and because of the shape of the tube must travel from one-quarter to one-half the length of the tube before reaching the holes feeding the push rods. When the engine is started cold the regulator quickly brings the oil up to water temperature and closes the valve lash to normal; and when the engine is driven under conditions giving high oil temperature the regulator cools the oil.

The inlet valve of any given cylinder opens 14 degrees before the piston reaches top center on the suction stroke, see Fig. 44, and closes 71 degrees after the piston reaches bottom center. The exhaust valve opens 56 degrees before the piston reaches bottom dead center on the

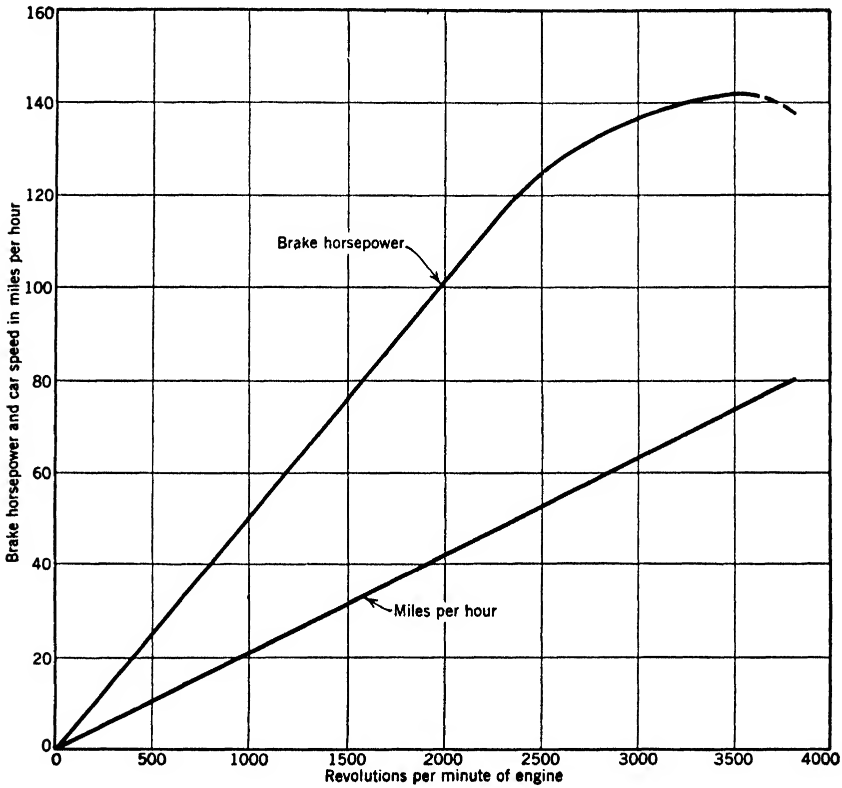


FIG. 45. Full-throttle power and speed curves for Buick automobile engine.

power stroke and closes 25 degrees after the piston reaches top dead center. The point when the valves are 0.004 in. off their seats, indicated on Fig. 44 as the timing point, is considered to be the beginning or end of their effective opening. The interval between the opening and closing points and the timing points represents that portion of the cam where the rate of lift is slow, permitting some small variation in valve clearance without causing noisy valve action.

The power developed by this engine on the test stand is shown by the brake horsepower curve in Fig. 45, when operating at full open throttle. The maximum horsepower is 141 at 3600 rpm; as a matter of interest the car speeds attained for the various engine speeds have also been plotted, a speed of 75 mph is attained when the maximum horsepower of 141 is developed; for higher car speeds the brake horsepower decreases very rapidly; hence lower efficiencies and increased fuel consumption are effected. The nominal brake horsepower rating by the S.A.E. equation,[1] $\frac{d^2n}{2.5}$, is 37.81 bhp at 1000 ft per min piston speed. At 3600 rpm the piston speed is $\frac{3600 \times 4\frac{5}{16} \times 2}{12} = 2588$ ft per min. If the brake horsepower is increased according to piston speed it will be $37.81 \times 2.588 = 97.85$ at 3600 rpm, which is considerably less than the actual developed 141 bhp. This discrepancy is due to improved design and operating features, notably, larger valves, larger bore, shorter stroke, higher compression ratio, improved fuels, better carburetion, lighter pistons, less power to auxiliaries, reduced weight throughout because of improved materials, better balancing, better lubrication and cooling, and other changes.

40. Airplane Engines. One of the marked features of aircraft development has been the effect it has had upon the refinement and perfection of the internal-combustion engine. Gasoline engines intended for aircraft are nearest perfection of any type yet evolved. Because of the peculiar demands imposed upon the aviation engine, it must possess all the features of reliability, economy, and efficiency now present in automobile, industrial, and marine engines. Because of the unstable nature of the air medium through which it is operated and the fact that heavier-than-air machines can maintain flight only as long as the power plant is functioning properly, an airplane engine must be more reliable than any internal-combustion engine used on either land or water. Whereas a few pounds of metal, more or less, make practically no difference in a marine engine and have very little effect upon the speed or hill-climbing ability of an automobile, an airplane engine must be as light as possible, that is, obtain maximum power with minimum weight. The weight of airplane engines usually varies from 1 to 2 lb per hp.

Airplane engines, as a rule, must operate continuously at high speeds in order to obtain a maximum power delivery with a minimum piston displacement. Automobile engines are not required to run constantly at maximum speed, but most airplane engines must function for extended periods at speeds as nearly the maximum as possible. The necessarily light framework of the airplane makes it difficult for an

[1] This equation is used for automobile taxation by some states. d is the cylinder diameter in inches and n is the number of cylinders.

engine to perform at maximum efficiency on account of the vibration of its foundation while the craft is in flight. Automobile engines, although not placed on foundations as firm as those provided for stationary engines, are installed on supports that are more stable than the light structure of an airplane. The aircraft engine, therefore, must be very well balanced and must be capable of being operated continuously under the most unfavorable conditions.

As numerous forms of airplane engines have been devised, it would require a volume of considerable size to describe even the most important developments of recent years. Only a relatively brief review of the features of several typical airplane engines can be given here, thus enabling the reader to obtain an understanding of the art, recognize the various types, and study the advantages and disadvantages of each type. Airplane engines may be divided into four main classes: rotary, radial, vertical, and "V" type. All of these engines operate on the four-stroke Otto cycle; the utilization of Diesel engines (weighing about 2 lb per hp) has also been proved to be practicable for aircraft purposes.

The World War began during a very early stage in the development of aircraft. France possessed large numbers of airplanes propelled by every conceivable type of engine, including air-cooled, liquid-cooled, fixed radial and rotating radial, four-, six-, eight-, and twelve-cylinder types, but with no definite policy as to which type to select for immediate military purposes. England and America had very few airplanes and still less experience. Germany, from the beginning, decided to restrict development to the six-cylinder in-line liquid-cooled engine on the assumption that this type, though heavy, would be reliable, economical of fuel, and permit of rapid production with limited manufacturing resources. This policy was probably correct, even as events turned out, and would certainly have been correct had the war, as Germany undoubtedly expected, proved to be of short duration. The progress of the war soon indicated that several types of aircraft were needed: (1) Pursuit planes for fast, small, and light fighting machines, capable of rapid maneuvering and climbing to high altitudes, but not required for long-sustained flight. Air-cooled rotating engines and liquid-cooled eight-cylinder V-type were used, but the present-day air-cooled fixed radial engine would have been far superior for this purpose. (2) Observation planes for general reconnaissance work capable of attaining high altitudes and long-sustained flight but not necessarily fast. The liquid-cooled eight- and twelve-cylinder V-types were used. (3) Bombing planes capable of carrying heavy loads and of flying great distances without refueling. The same engine types were used as for observation work.

Present-day airplane engines still reflect the researches and developments begun during the World War and resulting from the war's urgencies. The practicable arrangements of cylinders in airplane engines fall into two groups, radial and in-line. Each of these divides at present into a few simple classes as shown in Table XV. The twenty-

TABLE XV

CLASSIFICATION OF AIRPLANE ENGINES

Radial, Air-cooled	In-line, Liquid-cooled
5-cylinder, single-row	12-cylinder, V
7-cylinder, single-row	12-cylinder, opposed
9-cylinder, single-row	16-cylinder, X
12-cylinder, hexagonal	18-cylinder, W
14-cylinder, double-row	24-cylinder, X

and twenty-eight-cylinder types are also in some respects attractive, with 5 or 7 radial banks of four cylinders in-line with liquid cooling. The five- and seven-cylinder radials are useful only in smaller sizes, but the nine- and fourteen-cylinder engines appear capable of developing 1000 hp or more with air cooling. The in-line types appear at best advantage when liquid-cooled. The twelve-cylinder V and opposed types at present seem to offer the most for capacities in the 1000- to 1200-hp class.

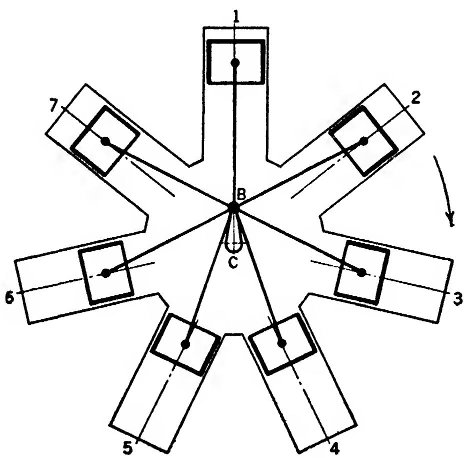


FIG. 46. Diagrammatic arrangement of rotary and radial airplane engines.

Rotary and radial engines, Fig. 46, have the cylinders arranged at equal angular intervals around a complete circle. The number of cylinders generally varies from 5 to 14. It is not possible to arrange more than 9 cylinders in a circle without increasing the size of the crankcase to dimensions which give an over-all diameter too large for use in an airplane. If more cylinders are desired they are arranged in two planes or rows, with the same number of cylinders in each row; with air cooling, the cylinders of the rear row are staggered with reference to those of the front row; with liquid cooling, the cylinders may be placed behind those of the front row.

In a rotary engine the crankshaft is stationary and the cylinders are mounted on a cylindrical crankcase which is free to revolve around the crankshaft. The reciprocation of all heavy parts is avoided, although there does exist a relative reciprocating motion between the pistons and cylinders. The pistons rotate about the crankpin as a center, *B* in Fig. 46, the cylinders about the crankshaft as a center, *C* in Fig. 46. Gasoline is fed under pressure through the hollow crankshaft to a spray nozzle in the crankcase of a rotary engine. Thence it is drawn into the cylinder during the suction stroke. The air for the explosive mixture enters the cylinder through the exhaust valve in the head of the cylinder during the first part of the suction stroke. Motion is imparted by the usual explosion in the heads of the cylinders when the gas has been compressed by the pistons. Since there is no movement of the crankshaft on which the rod holder (Fig. 51) turns, the force of explosion is exerted against the walls of the movable cylinders, causing the cylinders to revolve. This rotary motion is transmitted to the propeller, which is rigidly attached to the rotating parts of the engine and is free to turn on the crankshaft. The engine is very light in weight and its large rotational inertia exerts a steadying influence on the airplane. However, considerable energy is wasted in the resistance of the air to rotation, and the supply of fuel and lubricating oil is difficult and wasteful. Because of recent developments in the radial engine, it has superseded the more intricate rotary engine in airplanes.

A fixed-radial engine is one in which the center lines of the stationary cylinders are radii of the same circle, the connecting rod from each cylinder imparting motion to a revolving crankshaft. This type of engine has gained much prominence by reason of its use in successful transatlantic airplane flights. Smoothness of engine operation can be obtained only by using a considerable number of cylinders; practical limits are nine-cylinder, single-row, and fourteen-cylinder, double-row arrangement. The fundamental reason for adopting the radial type of engine is that a better distribution of stress is obtained by having all the pistons acting on the same crankpin so that the crank-throw and pin are continuously under maximum stress.

The " Wasp " engine, Figs. 47, 48, and 49, is a typical fixed-radial airplane engine. The Pratt and Whitney Company makes a number of sizes of radial engines, 320, 400, 550, 750 (all single-row), and 1150 hp (double-row) in order to meet the demands of military and commercial aviation. The medium and larger sizes have played a very definite part in the development of commercial aviation. Many of the commercial transport and mail planes are powered with engines of this type.

The features of this engine are its two-piece crankshaft, Fig. 50, solid master connecting rod, Fig. 51, forged aluminum crankcases, enclosed valve gear, automatic lubrication of valve gear, built-in supercharger, control mechanism for use of hydro-adjustable pitch propellers, radio shielding, dual ignition, air deflectors or baffles between cylinders, and the grouping of all accessories at the rear, Fig. 48. The nose or front section of the crankcase is fitted with a deep-row ball thrust bearing to carry the propeller thrust; the valve tappets are also

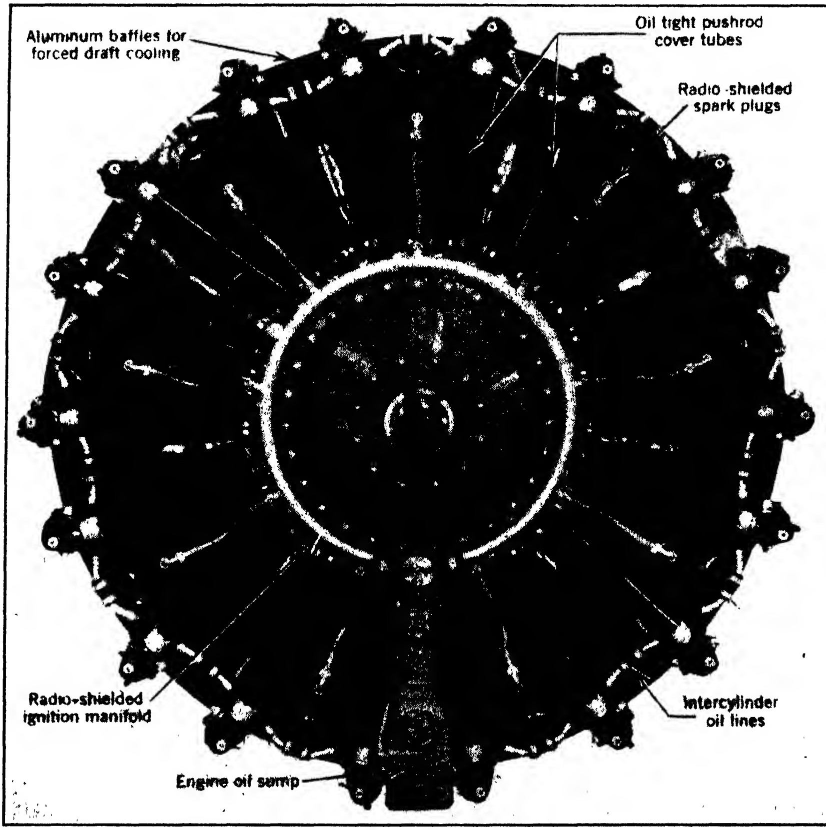


FIG. 47. Front view of Pratt and Whitney Wasp air-cooled radial airplane engine.

carried in this section, which encloses the cams and their operating mechanism. The main crankcase consists of two parts, divided on the center line of cylinders, and carries three large roller bearings which act as the main crankshaft supports. Immediately behind the crankcase is the blower or mounting section. External lugs and supports are provided for attaching the engine to the airplane. The supercharger is carried in this section, together with the gearing. The mixture of gasoline and air is drawn from the carburetor through the center of the section into the impeller of the supercharger, which discharges it through the diffusion vanes to an annular intake manifold from which tangential intake pipes, see Fig. 48, lead to the intake valve of each cylinder. The rear or accessory section of the crankcase, made of cast magnesium alloy, is attached to the blower section. It carries all the accessories, including two high-tension magnetos, an electric inertia starter, generator, fuel

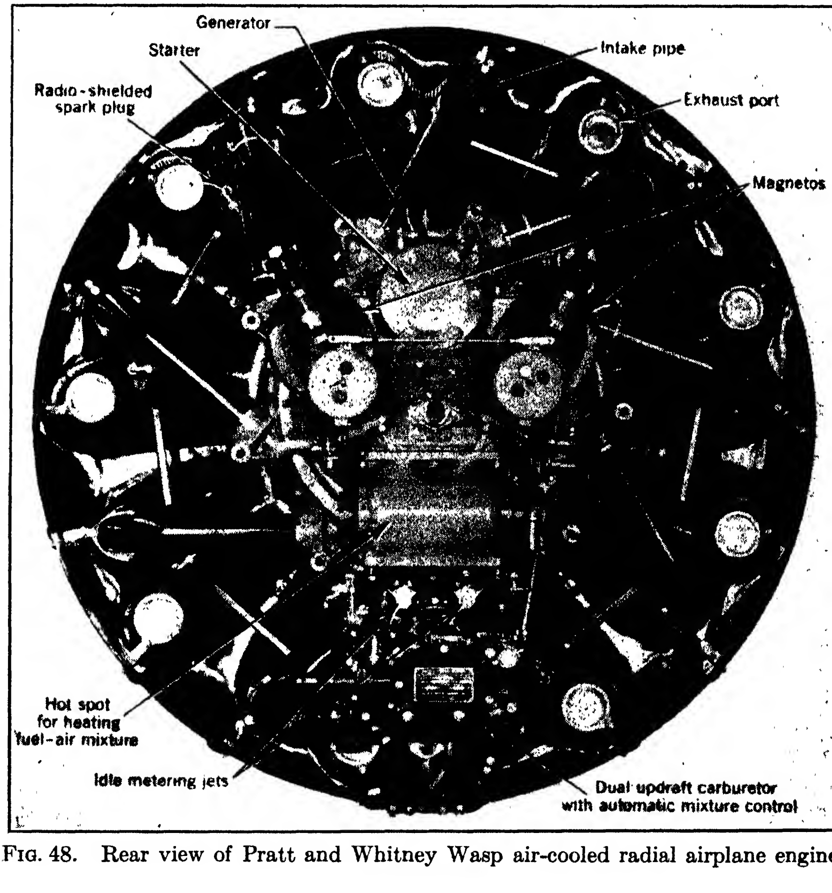


Fig. 48. Rear view of Pratt and Whitney Wasp air-cooled radial airplane engine.

pump, dual carburetor, two oil pumps, strainer, tachometer drive, and synchronizer drive.

The cylinders, Fig. 52, have integral cooling fins and are machined from steel forgings. Each barrel is screwed and shrunk into a cast aluminum cylinder head, and fastened to the crankcase by a hold-down flange at the bottom of the cylinder barrel. Each cylinder has one inlet and one exhaust valve, seating on inserts which are shrunk into the head casting. The exhaust valves are faced with "stellite" to prevent erosion and wear; they are also sodium-filled to assist cooling. All valve-operating parts are enclosed; see Figs. 47, 48, and 52. The eighteen valve tappets actuate the rocker arms through tubular duralumin pushrods, Figs. 47 and 49, which have hardened steel ball ends. The rocker arms are supported by ball bearings in the rocker housings which are part of the cylinder head. The odd number (nine) of cylinders makes it possible to secure a uniform and continuous

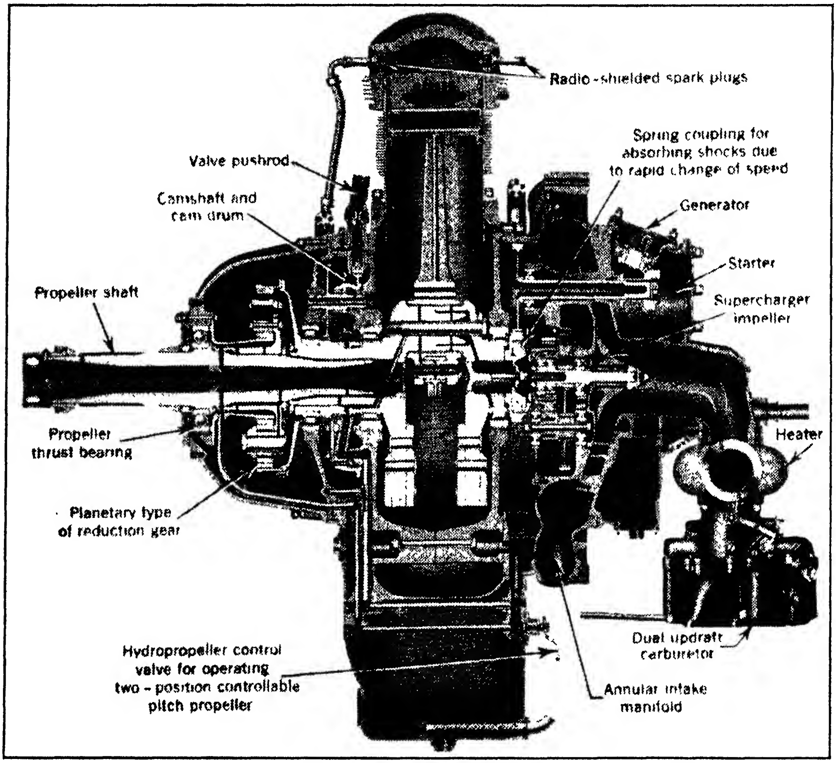


Fig. 49. Side sectional view of Wasp air-cooled radial airplane engine.

turning effort by having each alternate cylinder fire in sequence, thus, 1–3–5–7–9–2–4–6–8–1, etc.

Pistons (shown in Fig. 49) are machined from aluminum alloy forgings; the underside of the piston head is ribbed for additional strength and increased cooling area. Each piston is equipped with four compression rings and one oil scraper ring. The forged alloy-steel crankshaft, of the single-throw type two-piece construction, see Fig. 50, split in the center of the pin, is one of the features of Pratt and Whitney design. The rear section of the shaft telescopes into the front

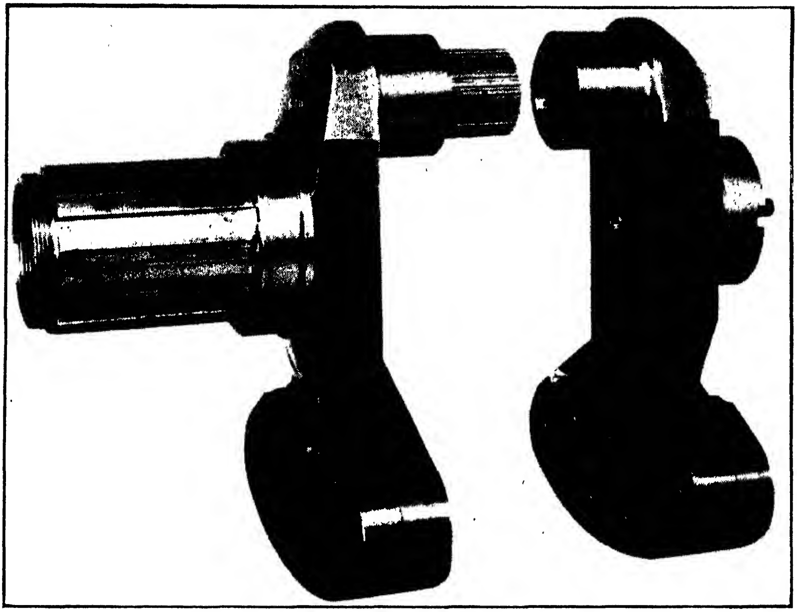


FIG. 50. Wasp single-throw two-piece crankshaft.

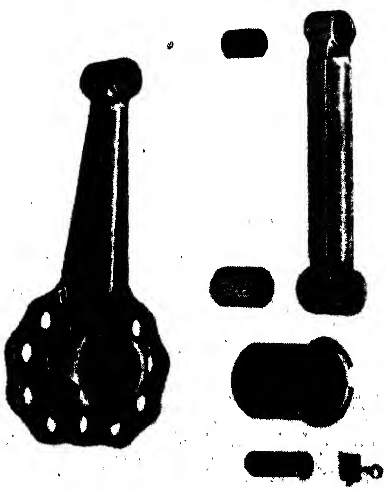


FIG. 51. Wasp master connecting rod, one link connecting rod, with accessory pins and bushings.

FIG. 52. Wasp air-cooled cylinder, inlet valve on right; note deep fins on head, radio-shielded spark plug, and thermocouple leads.

section and is held in angular position by 31 splines; the two sections are held together by a through bolt. The forward or propeller section of the shaft is integral with the crankpin, so that the power is transmitted through solid material. There is no tendency for the two parts of the pin to pull apart because the propeller thrust is taken up by the thrust bearing in the forward nose section. Because of the two-part crankshaft the crankpin end of the master connecting rod can be made solid, as shown in Fig. 51. Eight more link rods with knuckle pins complete the connecting-rod assembly.

Pressure lubrication is supplied by a gear-type pump to the crankpin, knuckle pins, cam drum, valve-gear mechanism, drum pinion shaft, accessory shaft, and supercharger gearing. A fine mesh screen is provided which cleans the incoming oil; this screen is readily removable from the underside of the accessory compartment. A pressure-relief valve maintains a uniform pressure on the system. All the oil is drained to a single sump located between the two lower cylinders, see Figs. 47 and 49, which is scavenged by another gear pump, the oil being returned to an external tank.

The intake heater consists of a stainless steel manifold cast in an aluminum housing through which the exhaust gases pass. This provides a heated surface which completely surrounds the intake mixture flow as well as a heated tube through the center of the mixture passage streamlined to obtain good vaporization of the air-gas mixture with a minimum of flow resistance. This heater facilitates all-weather operation and provides regulation of the heat of the intake mixture.

41. Liquid-cooled versus Air-cooled Aircraft Engines. The important problem of adequately cooling the combustion-chamber walls of an aircraft engine consists of providing sufficient surface in intimate contact with the coolant (air or liquid) of requisite volume and velocity to maintain the temperature of the interior of the walls within definite temperature limits. Also to provide the proper wall thickness to assure both adequate conduction of heat and physical strength at the temperature limits established, and to withstand successfully mechanical stresses imposed by gas pressures, expansion, and vibration.

One of the chief problems of the aircraft engine is the dissipation of the waste heat with a minimum loss of power. The possibilities, liquid cooling and air cooling, present two aspects of the same problem, one indirect cooling and the other direct cooling. By transferring the waste heat to a medium enveloping the cylinders which can then be passed to a cooler designed for cooling only and placed in the most advantageous position on the aircraft, the necessary cooling can be achieved for less power than the direct air-cooled case where the heat must be dissipated at its source. The liquid coolant has an extra reservoir of heat capacity over and above that of the engine itself and its latent heat. The former feature enables more heat to be absorbed under adverse conditions; or when power is low and cooling is high it forms a reservoir tending to maintain the engine surfaces at a uniform temperature, thus facilitating demands on the engine when the power is increased. The latent-heat effect is an addition to the sensible-heat effect and enables the cylinder head temperature to be kept low under exceptional conditions of heat dissipation.

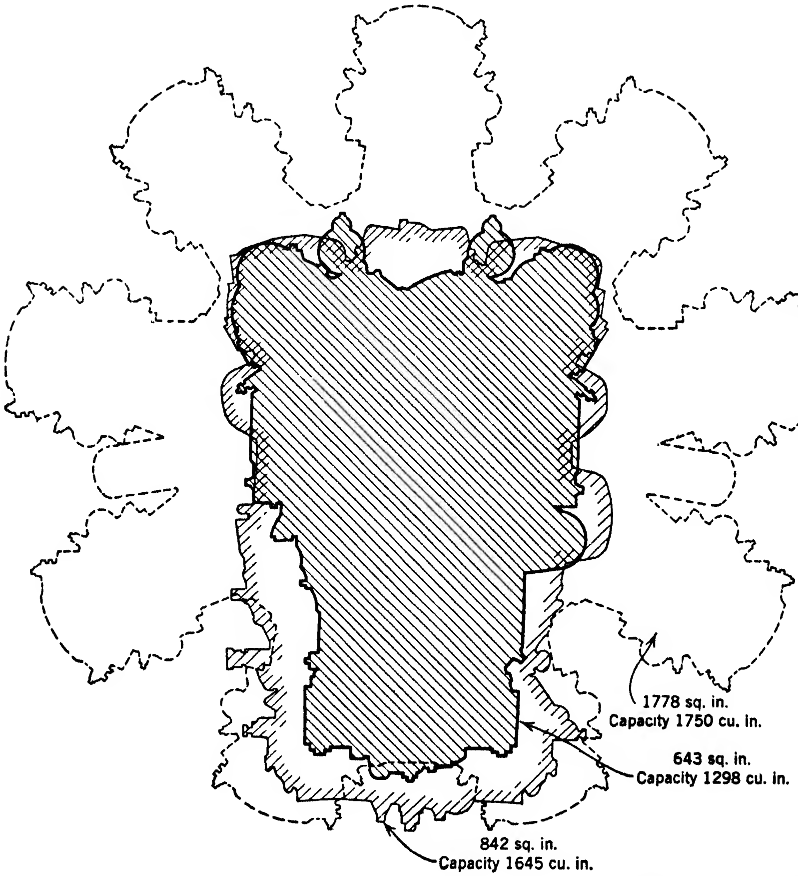
The liquid-cooled engine has an initial disadvantage of a radiator and its connections, involving considerable additional weight, and, what perhaps is even more serious for military purposes, much greater vulnerability; but against these defects must be offset a large advantage because of reliability, and the ability, owing to the lower cylinder temperature, to consume less oil, to employ a higher compression, and therefore to obtain a lower fuel consumption. The radial engine, by reason of the disposition of the cylinders and its relation to the slip stream from the propeller, offers the most ideal, and probably the only feasible, design for air cooling.

The use of fins on the exterior surface of the cylinder head and barrel has become universal practice for obtaining the necessary surface on the air-cooled cylinder unit to maintain the temperature of these parts within operating limits. From a secondary standpoint, the finned type of construction also provides a high order of resistance to deflection under explosion loading on both head and barrel. To be effective the cooling fins should be as deep, thin, and as closely spaced as manufacturing and considerations of air flow will permit; especially when applied to those portions of the cylinder head which receive the most heat, such as the exhaust port, the area adjacent to the exhaust-valve seat, and spark-plug bosses.

For any given air-cooled engine there is a limit to the direct cooling surface that can be placed on the cylinders. There is also a limit to the practical temperature at which the available materials can be operated and to the mean temperature difference available for cooling; therefore, there is a limit to the amount of waste heat which can be dissipated. There is no such limit to the cooling of the same engine when liquid cooled because the cooler, being independent of the engine, can be designed to suit the power and form of the engine.

There have been enormous strides made in the technique of air cooling and there is a great preponderance of air-cooled aircraft engines in the United States; whereas liquid-cooled aircraft engines are used more extensively abroad, because European engineers believe in the ability of this type to persist in the future when improved fuels will be available for use with higher compression pressures and when the problems of detonation and heat dissipation will be more intense than at present.

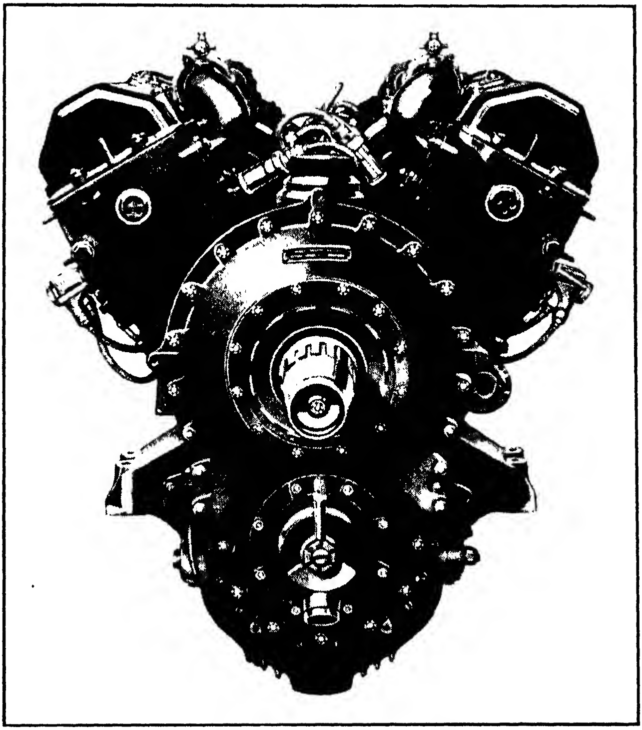
The reason for the popularity of the air-cooled engine in America is the simplicity attained in commercial service due to elimination of a liquid-cooling system, and the remarkable development work by several



Courtesy of *S.A.E. Journal.*

Fig. 53. Comparison of frontal areas of typical liquid-cooled and air-cooled aircraft engines.

of the manufacturers of radial engines. These engines have gained such progress in commercial service, as well as military service, that the liquid-cooled engine development has suffered and nearly resulted in stagnation in the matter of design. The unsatisfactory design features of American liquid-cooled engines have also worked against their popularity, whereas in Europe the prominent engine manufacturers have supported the liquid-cooled engine development during and since the World War. In the matter of cruising range it appears at present that there is little to choose between liquid- and air-cooled engines, provided the fuel distribution in the engine is sufficiently uniform that one or two cylinders are not required to operate excessively lean in order to bring down the average fuel consumption of the engine.

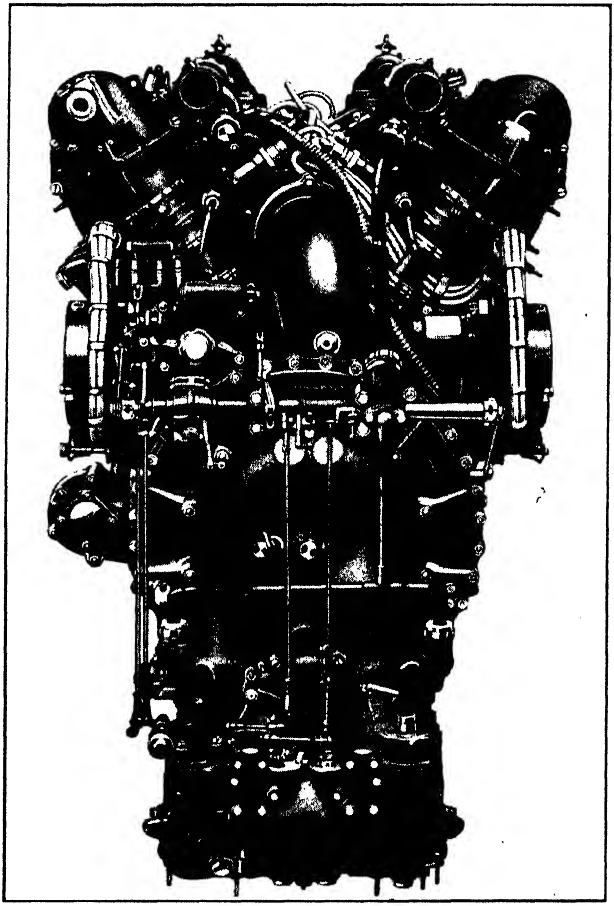


Courtesy of *S.A.E. Journal.*

FIG. 54. Front view of Rolls-Royce Kestrel liquid-cooled aircraft engine. Twelve-cylinder, 60 deg. V-type, 5 × $5\frac{1}{2}$ in., 585 hp (2500 rpm) at sea level.

The basic liquid-cooled aircraft engine is a twelve-cylinder, 60-degree, V-type, having a small frontal area, see Figs. 53, 54, and 55, as compared to a nine-cylinder radial air-cooled engine of similar capacity. When the weights of the engine mounting, cowling, exhaust system, propeller, coolant, etc., have been taken into account, it has been found that the difference in installed weights of liquid-cooled and air-cooled engines lies primarily in the difference between the net dry weights of the two engines, the balance being in favor of the air-cooled type. This finding refutes the generally accepted opinion that the weight of an installed liquid-cooled engine is much greater than that of an air-cooled engine of equal power.

The coolant used in liquid-cooled aircraft engines is generally ethylene glycol because of its high boiling (345 F) and low freezing (2 F)



Courtesy of *S.A.E. Journal.*

FIG. 55. Rear view of Rolls-Royce Kestrel liquid-cooled aircraft engine.

points, but it is inferior to water in other properties; there are possibilities of using sealed-water systems under pressure for cooling purposes. The substitution of ethylene glycol for water in a liquid-cooled engine will produce higher metal temperatures because: higher coolant temperatures are used, 230–280 F for glycol, 160–180 F for water; the heat-transfer coefficient of glycol is relatively poor, water having a heat-transfer coefficient about 4.7 times that of glycol; the evaporative effect, which with water cooling acts as a safety valve on hot spots, is reduced. The possibility of distortion and cracking may become more serious when a change-over is made from water to ethylene glycol and it becomes necessary to reconsider flows and velocities of the coolant in the same way as the air-cooled engine manufacturer arranges the cooling fins and directs the flow of the cooling air.

The weight and drag of cooling systems are of major importance when considered in relation to modern streamlined aircraft having a low weight per horsepower of the engine. With reference to the relative drag of the radial air-cooled and the in-line liquid-cooled engines, wind tunnel tests indicate that the total installation drag is considerably less for the latter than for current radial air-cooled installations designed for the same aircraft speed. Improved air-cooled engine cowling will no doubt reduce this difference.

It has been predicted that the limit of horsepower (1150 hp with 14 cylinders, double row) has been reached for air cooling and that it will be impracticable to handle more power in a single radial engine. However, there is no indication that the end has been reached in either of these matters. Large military engines, if successfully air-cooled, offer advantages in freedom from plumbing troubles and reduced vulnerability which should not be overlooked. Many of the large aircraft engines (above 1000 hp) now under development are liquid-cooled and of non-radial form. This is probably due to the fact that the streamlining of modern airplanes has been developed to the point where the drag of the air-cooled radial engines is a large portion of the total drag. Another logical step in improving aerodynamic features may be that of placing the engines entirely within the airplane structure and using "skin-type" radiators.

42. Potential Developments and Trends for Aircraft Engines. Air-cooled radial engines will probably develop by increasing the number of crankpins and by reducing the number of cylinders per crank, and the liquid-cooled in-line engine will employ more cylinders per crank. The selection of the liquid-cooled engine form appears to favor a crankshaft having six throws with multi-pistons on each throw. Increasing the number of cranks reduces the frontal area of the engine, but the length does not increase proportionally because the fewer banks of cylinders give space between for fitting accessories. Each type, therefore, will approach the other in regard to mechanical problems and general arrangement, and the differences in frontal area (see Fig. 53) will diminish. This predicted development will modify mechanical efficiencies, the tendency being to decrease on the radials and to increase on the in-line engines; specific weights also will change more favorably on the in-line engine.

An important feature of increasing the number of cylinders is the better detonation characteristics obtained because of the decrease of cylinder size, increased speed of rotation, and better cooling. The decrease in cylinder size improves the detonation characteristics by shortening the flame travel. The area of combustion surface for absorbing heat is larger per unit volume and, at the same time, the heat-flow paths are shorter, both effects tending to improve the cooling and reduce local hot spots. These features are particularly valuable in their effects on the piston and poppet valves, which have become very critical engine parts. Valve-seat distortion and cylinder-head cracking also should be reduced by the consequent increase in head stiffness; other advantages of using more cylinders are increased reliability and smoother running of the engine. The disadvantages of the many-cylinder (more than twelve) engine are the expense of development, greater initial cost, increased cost of maintenance, larger frontal area, more intricate crankshaft assembly, higher bearing loads, and more difficulty in the design of a satisfactory valve gear.

The tendency to streamline airplanes may result in the placing of the engine or engines entirely within the structure; the liquid-cooled in-line engine using skin-type radiators would be well adapted to this development. Many airplanes are now metal-covered; hence the skin-type radiator should not involve any great weight increase or other serious technical difficulties. It may be necessary to abandon present-day methods of cylinder construction that rely on the art of the pattern-maker and the foundryman, in order that cooling passages may be better controlled in sectional area and more uniform metal sections may be obtained to reduce heat stresses and weight.

The aircraft engine of the immediate future will have a rating of 1000 to 1200 hp, with an emergency rating about 10 per cent greater. The dry weight will be 1.2 lb per bhp or less, including reduction gear. The brake mep will be 185 to 200 lb per sq in., obtained with a geared built-in centrifugal compressor. Ignition will be by means of high-tension magnetos. The number of cylinders will be 12 for the V or opposed types, with liquid cooling, and 9 or 14 for the radial air-cooled types. Crankshaft rotational speeds will be 2500 to 3200 rpm. The fuel consumption at rated power will not exceed 0.50 lb per bhp per hr in the liquid-cooled types, and at cruising will be 0.45 lb per bhp per hr, or less. Fuel injection will probably be used, so that mixture distribution to all the cylinders will be uniform; improved fuels may also permit raising the compression ratio above 7 to 1 without destructive detonation.

The direction of rotation of future aircraft engines will be optional, so that the torque in a two-engine installation will be balanced. For single-engine installations, dual propellers rotating in opposite directions will be required because of torque considerations. A single propeller utilizing 1200 hp at 1200 rpm produces a torque of 5000 lb-ft. To resist this torque by wing rigging entails sacrifice in the matter of drag. Propellers will, of course, be fully controllable in the interest of take-off load, speed, and fuel economy. Sea-level performance will be maintained at altitude by means of the exhaust turbo-compressor type of supercharger. However, the theory that supercharging blowers for internal-combustion engines can be driven by turbines utilizing the exhaust gases from the engines themselves is in itself not sufficient to give the advanced results claimed for this method of supercharging. The advantages which at first appeared to be so obvious have not been easy to realize. The exhaust-gas turbo-charging of engines had to pass through successive phases of development for some years before the theory could be adopted in practice and satisfactory results obtained.

Developments in the Diesel engine field may alter the foregoing specifications, but at present this is a matter of conjecture. Experimental work is going on and in view of the high cost of gasoline in foreign countries, it seems logical to expect that the Diesel engine should become a serious competitor of the gasoline engine for aircraft use on account of its lower fuel consumption per unit weight, lower fuel cost, better ability to retain power at high altitudes, and the elimination of the fire hazard. Transatlantic flights, with intermediate stops at the Azores, of German airplanes powered with Junkers opposed-piston two-cycle six-cylinder Diesel engines have indicated the reliability of these engines for long-range flights. Another interesting engine, of which more should be heard, is the Beardmore (British) twelve-cylinder opposed-piston, liquid-cooled Diesel engine. It is a four-cycle engine, with two banks of six cylinders arranged at 180 degrees, which in its unsupercharged form develops 500 hp at 1750 rpm for a weight of 1450 lb or 2.90 lb per hp. The design is particularly attractive, as its shape allows it to be installed in the wing of an airplane completely out of sight, and with no head resistance except that of the propeller hub. In supercharged form and with the mechanical improvements made possible by several years of experience, the Beardmore engine should compare favorably with Otto-cycle types of aircraft engines.

Several American manufacturers have developed radial air-cooled Diesel engines for aircraft purposes. Reluctance on the part of commercial and military authorities to use Diesel-powered planes is due to the greater weight per horsepower, less speed, likelihood of smoky exhaust, and higher maintenance cost. The tell-tale plume of smoke from the Diesel-type engine must be eliminated before serious competition with the Otto-cycle type can occur in the military aviation field. The weight handicap of the Diesel is not serious provided the fuel consumption can be kept 10 to 20 per cent lower than that of the Otto type. Recent improvements in high-octane fuels available for Otto-cycle aircraft engines indicate fuel-consumption possibilities in the cruising range thus far acclaimed only by proponents of the Diesel-type engine.

The Otto-cycle engine burns the fuel explosively or at constant volume; this, with the low compression pressures attained, gives the largest practicable ratio of mean effective to maximum pressure and results in a favorable weight-power ratio. The Diesel engine may be operated with substantially constant-volume combustion, but the peak pressures may become extremely high and produce a fuel knock. The ratio of mean effective to maximum pressure is not favorable and the result is a high weight-power ratio. At low speeds nearly constant-pressure combustion will result; but at the high speeds required of aircraft engines, the constant-volume condition must be approached because of the extremely short combustion period. Supercharging the Diesel engine will require a lowering of the expansion ratio and preheating of the air, thus reducing the thermal and volumetric efficiency of the cycle. Based upon present knowledge the Otto-cycle engine can develop more power per cubic inch and per pound of weight than can the Diesel, and with cleaner combustion when the fuel-air ratio is kept high as demanded by high specific output.

In the future, indications are that 1500- to 2000-hp per engine will be demanded. Weight per horsepower will be still lower than in the 1000-hp field, varying possibly from 1 lb to 0.8 lb per bhp. Liquid cooling will be used, with low-drag (skin-type) or blower-cooled radiators, or, in the case of air cooling, by means of blowers and pressure-type cowling. Cylinders will not exceed 6 in. in diameter, and 18 to 24 or more cylinders will be required. High output per cubic inch and some form of compounding may result in lowering of the fuel consumption. Improved fuels may also permit raising the compression ratio without destructive detonation. Dual propellers with opposite rotation will be more essential than in the 1000-hp class. The matter of larger propellers or more small ones will require more study and the whole topic of propulsion may require new analysis, because of the higher airplane speeds which will be used.

CHAPTER VII

VAPORIZING OIL ENGINES

An oil engine may be defined as an internal-combustion engine that uses oil for fuel, the oil being sprayed into the cylinder or combustion chamber, during or at the end of the compression stroke. Oil engines are of three distinct types—the low-pressure, moderate-pressure, and the Diesel (high-pressure). Low-pressure oil engines operate on the Otto cycle and moderate-pressure oil engines operate on cycles approximating the Otto or Diesel cycle, or a combination of the two cycles.

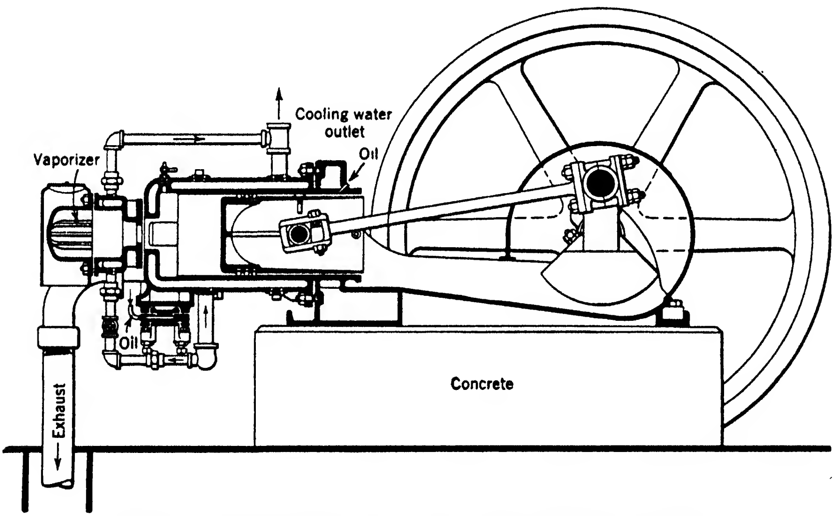


FIG. 56. De La Vergne type "HA" low-pressure oil engine.

43. Low-pressure Oil Engines. The low-pressure type of oil engine is so named because the compression pressure is usually less than 150 lb per sq in. Fuel is generally injected against hot plates, into a hot bulb, or into an uncooled vaporizer chamber during the compression stroke. When the atomized fuel comes into contact with the hot surface, vaporization occurs, and as the oil is usually kept out of the cylinder, only the expanding gases come in contact with the piston and cylinder walls. Initial vaporization is often produced by heating the auxiliary chamber externally; and after the engine has been started the surfaces retain enough heat from explosion to explosion to be continuously effective as a vaporizer. These engines operate on the Otto cycle and the explosion pressures are not high, seldom exceeding 250 lb per sq in.

Figures 56 and 57 show a longitudinal and a transverse section, respectively, of a typical low-pressure oil engine. This engine is of the horizontal four-stroke type, operating on the Otto-cycle principle with a compression pressure of about 60 lb per sq in. The exhaust and inlet valves are of the poppet type and are located in a valve box at the side of the cylinder, Fig. 57. The valves open upward and are operated, by levers passing under the cylinder, by means of cams on the camshaft. The exhaust cam is so designed that, on starting, the compression can be relieved by shifting the cam on the shaft. The oil is taken from the supply tank by a pump operated from the camshaft and forced into the vaporizer chamber through a spray nozzle, as shown in Fig. 58. The governor is of the fly-ball type and controls the speed by dividing the constant quantity of oil furnished by the pump into two parts—one of which (in proportion to the load) enters the spray nozzle, the other flows back to the fuel tank.

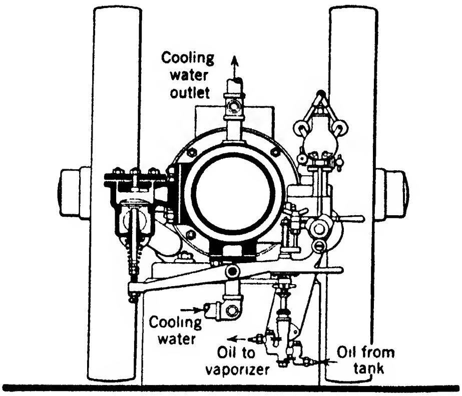


Fig. 57. Transverse section through cylinder of type "HA" oil engine.

The vaporizer chamber, Figs. 56 and 58, is heavily ribbed on the inside to increase its surface and is protected against radiation on the outside by a hood. To start the engine, the vaporizer is first heated by a lamp or torch. After the vaporizer has been heated to a dull red heat, a few quick strokes of the pump by a hand lever, Fig. 57, while the engine is being turned over in the normal direction by hand or by compressed air, usually suffices to start it. After the engine has been started, the heat of combustion is sufficient to keep the vaporizer hot enough to explode the charges regularly, so that no special igniter is required.

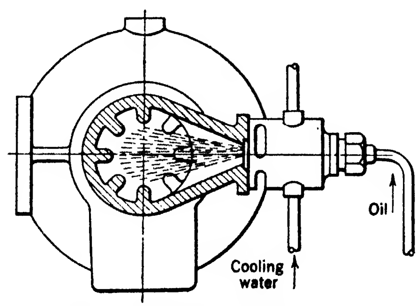


Fig. 58. Transverse section of type "HA" vaporizer.

The method of operation of this engine may be explained as follows: On the suction stroke of the piston, air is drawn into the cylinder through the inlet valve, left side in Fig. 57, and the pump injects oil into the vaporizer. The oil is almost instantly vaporized, but as yet the mixture is not explosive because the vapor is mixed with burned gases which remain from the previous explosion. On the return stroke, the piston compresses the air and forces a part of it into the vaporizer. It is possible that sometime during the compression stroke the vapor may begin to burn in the vaporizer; but the flame does not extend to the cylinder because the velocity of air flowing through the narrow opening into the vaporizer is greater than the velocity of flame propagation in the vaporizer. Near the end of the compression stroke the reverse takes place—combustion occurs with explosive force and drives the piston outward on the expansion stroke. This is followed by the exhaust stroke, after which the operation (cycle) is repeated.

Type " HA " engines have been built in sizes from $2\frac{1}{2}$ to 125 hp, those above 50 hp being of the twin-cylinder design. The combustion in an engine of this kind cannot be as complete as in the type where a thorough mixture of the fuel and air is accomplished, as with a carburetor. Some of the air admitted to the cylinder of a low-pressure oil engine will remain inactive, as it does not get near the oil. Combustion is not as complete as in the higher-pressure oil engines, hence the fuel consumption is higher and the efficiency is correspondingly lower.

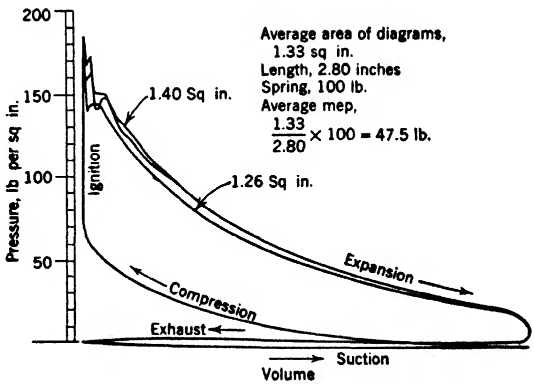


FIG. 59. Indicator diagram from low-pressure oil engine.

A typical indicator card for a low-pressure oil engine is shown in Fig. 59, which shows that the engine operates on the Otto-cycle principle. Because this type of engine can successfully use certain grades of Diesel fuel oils, it has been incorrectly called a " semi-Diesel " engine. Although many low-pressure oil engines are still being used for a variety of purposes, only a few are being manufactured. In recent years these engines have been displaced by the more economical, efficient, and reliable moderate-pressure oil engines and high-pressure (Diesel) engines.

44. Moderate-pressure Oil Engines. Moderate-pressure oil engines employ compression pressures of 200 to 350 lb per sq in. These engines may be designed to operate on cycles very closely approximating the Otto or Diesel, or a combination of the two cycles. Ignition in this type of engine may be effected by methods similar to those employed in low-compression oil engines, as explained in Art. 43. Injection of the fuel cannot occur during the compression stroke as in the low-pressure type, on account of probable premature ignition. The temperature corresponding to a pressure of 250 lb per sq in. is sufficient to ignite most fuel oils, but this temperature is not high enough to insure perfect ignition at every stroke. In order to avoid occasional pre-ignition, the

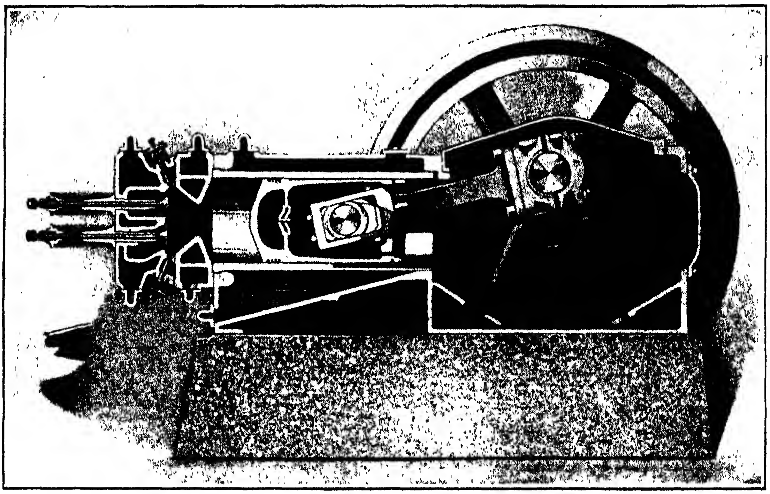


FIG. 60. Ingersoll-Rand type "PO" moderate-pressure oil engine.

fuel oil in moderate-pressure oil engines is injected at the end of the compression stroke.

Figure 60 illustrates a typical moderate-pressure oil engine, which operates on the four-stroke cycle having direct injection of the fuel oil. On the suction (outward) stroke of the piston the intake valve is open and air is drawn into the cylinder. Several degrees past the outer dead center the intake valve is closed, and during the remainder of the inward stroke the piston compresses the air into the combustion chamber (vaporizer) to a pressure of approximately 300 lb per sq in., as shown on the indicator card, Fig. 61. Before head-end dead center is reached the fuel-pump plunger is actuated by a cam, and sprays of oil are injected into the combustion chamber by the two fuel-oil spray nozzles which are mounted directly opposite each other in the cylinder head. These sprays come together near the center of the combustion space, and the force of their impact materially aids atomization, vaporization, and distribution of the fuel. Ignition takes place automatically from the heat of compression of the charge of air. The fuel injection is so timed that combustion at constant volume increases the cylinder pressure to about 450 lb per sq in., which is maintained until about 20 degrees after admission.

Expansion then begins, forcing the piston toward the right; this is the working stroke from which the engine develops power. At the end

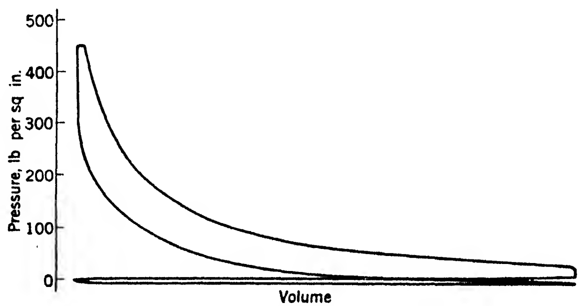


Fig. 61. Indicator diagram from moderate-pressure oil engine.

of the expansion stroke the exhaust valve opens, the exhaust stroke follows, and the burned gases are forced out of the cylinder. The air inlet valve opens a few degrees before the exhaust valve closes and a charge of fresh air is thus drawn into the vaporizer by the out-rushing exhaust gases, so that when the next suction stroke begins, the vaporizer is filled with air. The suction stroke of the next cycle follows, and the events are repeated. The flywheels, considerably heavier than for steam engines, absorb sufficient energy during the working stroke to maintain a nearly uniform rotary speed of the crankshaft during the suction, compression, and exhaust strokes.

These engines are built in single-cylinder units from 40 to 200 brake horsepower, also in twin- and four-cylinder units with corresponding increase in horsepower. They are of massive construction as compared with low-pressure units; a 110-hp moderate-pressure oil engine with a cylinder size of 17 × 19 in. operates at 260 rpm and weighs 30,000 lb.

45. Low-compression Spark-ignition Oil Engines. An interesting recent development is the Hesselman low-compression spark-ignition oil engine illustrated in section by Fig. 62. This engine is not offered as a substitute for the compression-ignition Diesel engine, but rather as supplementary to it, being more suitable in many places where a Diesel would not be economical. The Waukesha-Hesselman engine is a four-cycle low-compression engine which burns Diesel fuel oils. Injection of the fuel is similar to the Diesel method, but ignition is accomplished by means of a spark plug instead of by the heat of compression.

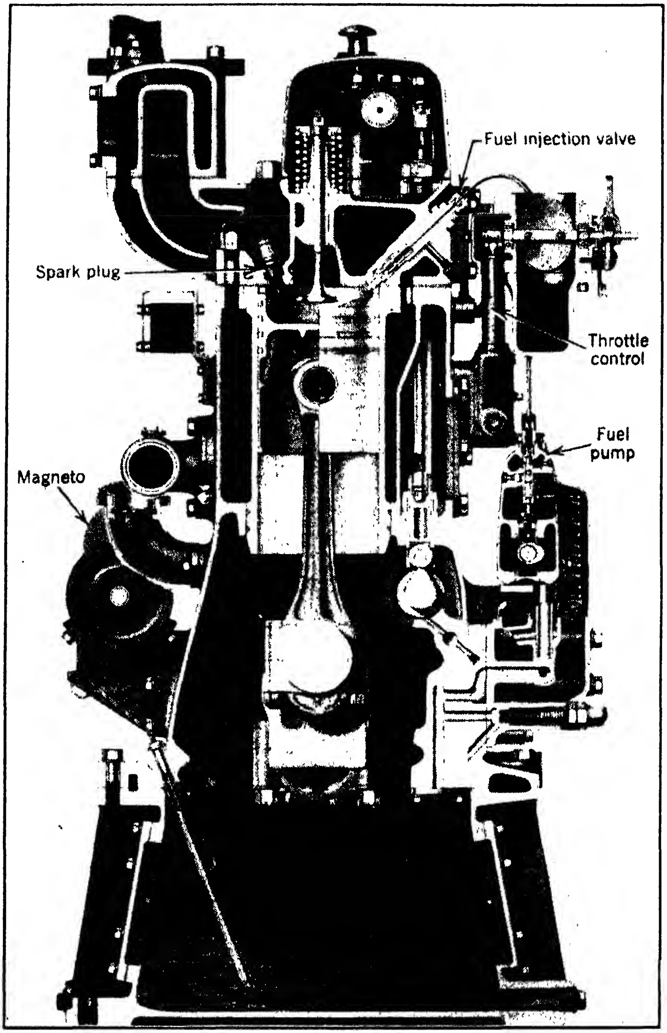


FIG. 62. Waukesha-Hesselman low-compression spark-ignition industrial oil engine.

With the intake valve open, the piston moves downward and a charge of fresh air is drawn into the cylinder. No fuel is admitted until compression is nearly completed. The air admission is controlled by a special throttle valve on the intake manifold interlinked with the fuel-injection pump control; see Fig. 62. The inlet valve closes; the piston moves upward and compresses the air in the cylinder to about 125 lb per sq in. Just before the end of the compression stroke, 50 degrees before top center, a measured quantity of fuel oil is injected into the combustion chamber through a Hesselman fuel valve by a Bosch fuel-injection pump, shown in Figs. 112 and 113. Injection of fuel ends about 25 degrees before top dead center.

The form of the combustion chamber, Fig. 62, and the arrangement of the inlet passages gives the air a tangential flow as it enters, and compresses it in a definite whirlwind pattern. This whirling air picks up the finely atomized fuel from the injector, thoroughly mixes it with the air, and sweeps it as a highly combustible mixture across the gap of the spark plug which ignites the fuel-air mixture about 15 degrees before the end of the compression stroke. The patented features of this engine make it possible to keep the mixture ratio in the zone surrounding the spark plug within the limits necessary for prompt and thorough combustion, independent of the engine load or speed. Combustion of the mixture delivers the power on the downward stroke of the piston. A peak combustion pressure of about 450 lb per sq in. is reached by a typical Hesselman engine of 5.8 to 1 compression ratio; the gases then expand to about 45 lb per sq in., the exhaust valve opens and, with the piston moving upward, the burned gases are forced out. The expansion line approximately follows the equation $PV^{1.3} = C$.

The cup-shaped crown in the piston forms the combustion chamber; the piston rings are located in the walls of the cup, thus providing cooling means for this part of the cylinder and permitting higher compression ratios with resulting high output. The Hesselman engine uses a magneto to supply the high-tension current to the spark plug. The magneto is of the inductor type having a stationary coil or winding with cobalt-chrome permanent magnets which rotate between laminated pole pieces. Details of the Hesselman fuel valve are shown in Fig. 107, and explained on the accompanying page.

Of particular interest are the three types of Hesselman cylinder heads that have been designed to cause the air to rotate around the axis of the cylinder, which movement continues during the compression stroke. These types are: head accommodating a wing valve, head with a Venturi port, and the ramp head. The head with the wing valve directs the air flow with a semi-circular wing or collar just below the valve seat. This wing prevents the air from flowing into the combustion chamber in any direction other than that not blocked by the wing below the valve seat. The cylinder head with the Venturi port controls the direction of air flow into the combustion chamber by the design of the air intake passage between the manifold and the valve seat. The ramp head is characterized by the unusual shape of the head, which takes the form of a ramp around the inlet valve. Here the head is so shaped that the air must follow the ramp and enter the combustion chamber in a spiral form.

The engine is started on gasoline supplied to the manifold by a few strokes of a conventional primer pump. It is practical to hand-start these engines because of the low-compression pressure. As the fuel is in the cylinder only a few degrees of crank angle before ignition occurs, the engine is less sensitive to the detonation of fuels than carburetor engines. It would not normally be expected that the spark would ignite the spray of liquid fuel and air in a Hesselman engine; however, it does ignite only because the residual heat in the combustion chamber is sufficient to vaporize a portion of the spray before it reaches the spark plug. Combustion is complete at 18 to 20 degrees after top dead-center position of the piston.

The Waukesha Company builds ten different models of Hesselman engines from $3\frac{3}{4} \times 4\frac{3}{4}$ in. to $8\frac{1}{2} \times 8\frac{1}{2}$ in., ranging in power from 30 to 300 hp with full-load speeds of 850 to 2500 rpm; however, some of these engines have operated intermittently at 3000 rpm. The brake mean effective pressure is high; a production industrial engine has shown 104 lb per sq in., whereas laboratory models have shown even higher values. Published performance curves show economies of 0.51 to 0.56 lb of fuel per bhp-hr at full load, see Fig. 154, with better economies at three-fourths load of 0.48 to 0.52 lb. This is slightly more than the fuel consumption of compression-ignition Diesel engines, and considerably better than the fuel consumption of most carburetor engines.

The Hesselman engine has provided a means for handling fuel oils at low-compression pressures and the Diesel engine has indicated the practical limits of high-compression pressures. Recent developments and present trends indicate that the future internal-combustion engine will probably be an engineering compromise, utilizing as many of the benefits of high compression as possible without sacrificing the advantages of the low-compression engine.

CHAPTER VIII

HEAVY-DUTY DIESEL ENGINES

The Diesel engine has been developed, both as a four-stroke cycle and as a two-stroke cycle engine, almost exclusively as an oil engine, and operates on a cycle, Fig. 17, which approximates the ideal Diesel cycle as described in Arts. 31 and 34. Of all types of commercial internal-combustion engines, Diesel engines generally give the best performance and are used for stationary, marine, and locomotive purposes, especially in the larger sizes. The charge of air is compressed in the engine cylinder to a pressure of about 500 lb per sq in. with a corresponding temperature of approximately 1000 F. At the end of compression, the temperature is sufficiently high to ignite the fuel when injected into the cylinder in the form of a finely divided spray. Combustion of this fuel results at nearly constant pressure; there is no explosion.

In general, Diesel engines, with the exception of fuel-oil pumps, fuel atomizers, and integral air compressors, are similar to gas engines in details of construction; but they are usually built heavier to withstand the higher pressures used with the Diesel cycle.

46. Recent Developments. Although the development of the Diesel engine dates back to the year 1890, it is only in the past fifteen years that this type of prime mover has really come into its own, particularly in the United States. Not only has it been brought to practical technical perfection in recent years, but also its field of application has been extended enormously. This is quite evident when the total horsepower of Diesel engine units made by American manufacturers is considered. Between the years 1920 and 1930, the total capacity of Diesel engines installed in this country increased from 1,000,000 to 4,000,000 horsepower; by the end of 1937 there was a total of about 12,000,000 horsepower installed.

In 1920, the average full-load fuel consumption of Diesel engines was about 0.50 to 0.55 lb per bhp-hr. Today, typical values per brake horsepower are as follows: full load, 0.40 lb; three-fourths load, 0.42 lb; and half load, 0.48 lb. The best performance of modern engines will go as low as 0.35 lb; in fact, one foreign 5500-hp engine uses only 0.329 lb per hp-hr with 18,000 Btu fuel, a thermal efficiency of 41 per cent. The improvement in fuel consumption since 1920 represents an increase in thermal efficiency of more than 20 per cent. Since the Diesel engine of today has an efficiency which approaches quite closely the theoretical efficiency possible with this type of prime mover, there is not much reason to believe that present-day engines will become obsolete before the normal length of their life has been reached. Engines have undergone great development in mechanical design, however, and from this standpoint alone it may be advisable to replace an old engine with a new one.

During the past ten years, the greatest advance in Diesel engine design has been the development and perfection of the solid-injection engines. Air injection has been almost entirely eliminated, except where previous engines in a plant are of this type or where a particularly poor fuel is to be burned. Solid-injection engines show a marked variation in design and, in so far as behavior is concerned, they may actually be divided into three classes on the basis of their combustion equipment. As a class, however, the performance of all solid-injection engines may be considered on an equal basis, as the difference in design has more to do with the matter of operation and maintenance than with thermal efficiency.

The Diesel engine has been subject to progressive refinement and to continual increases in size. Burmeister and Wain of Copenhagen, Denmark, have built the largest single Diesel engine in the world. This unit, installed in the Copenhagen electric light plant in 1933, is of the solid-injection two-cycle double-acting type. Each of the eight cylinders has a 33-in. bore and a 59-in. stroke; and at 115 rpm the engine delivers 22,500 bhp. One European manufacturer has offered to build a 40,000-hp engine; another engine of 80,000 hp has been proposed for installation on a ship; America's largest units are the five Diesels in Vernon, California. They are of 7000 bhp each, built by Hooven, Owens, Rentschler Company, Hamilton, Ohio; see first item of Table XVI, page 187, for data on these engines.

Above 1000- and below 4000-hp capacity, two-cycle engines are favored almost two to one; and above 4000 hp, two-cycle types are used exclusively. At present, there are no four-cycle double-acting engines, but of the large two-cycle units about half are double-acting. Considering only capacities within the range of both types, it is clearly a choice between the simplicity and accessibility of the single-acting engine, and the smaller space requirements and smaller number of cylinders for a given power of the double-acting type. All large Diesel engines have vertical cylinders. Between 300 and 1000 hp, the use of a two-cycle or four-cycle engine is a matter of choice; many builders make both types. The two-cycle avoids valves, has smaller cylinders, takes less floor space for a given power, has lower first cost, is lighter, and requires less maintenance. The four-cycle engine is generally considered the standard design, is easier to cool and lubricate, requires no scavenging equipment, and has lower mep.

Marked advance has been made in standardizing not only the general design of the American Diesel engine, which has been toward simplification of lines and centralization of control, but also the design of parts. Rotative speeds range from 2000 rpm for small high-speed engines to 120 to 150 rpm for large heavy-duty Diesel engines, and piston speeds of 1500 fpm in small engines to 1000 fpm in large engines. Considerable progress has been made in the development and application

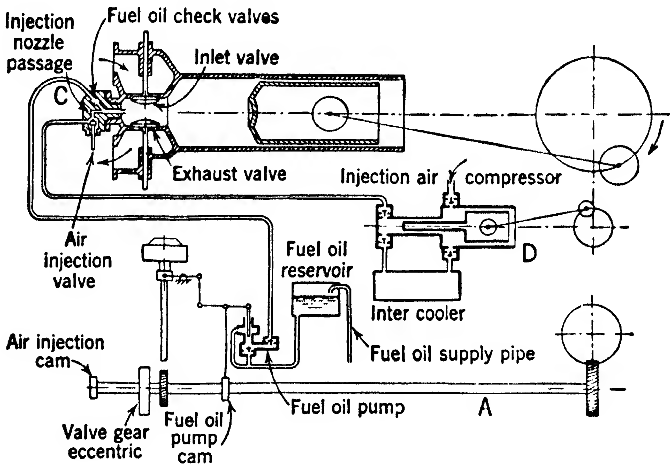


FIG. 63. Diagrammatic arrangement of a four-stroke cycle air-injection Diesel engine.

of alloy materials, welding, and in precision manufacturing methods. Controlled pressure lubrication, more careful consideration of fuel-injection methods, combustion, and scavenging have also contributed much to the perfection of the modern engine. Diesel-engine installations in communities must be silenced and isolated to prevent noise and vibration. Adequate intake and exhaust silencers are available, as well as bases utilizing cork, helical springs, and the like.

47. Four-stroke Diesel Cycle. Air Injection of Fuel. The elements of a Diesel engine power plant using air injection of the fuel are shown in Fig. 63. The engine illustrated is of the horizontal type operating on the four-stroke cycle. To the crankshaft is geared a camshaft A which revolves at one-half the engine speed. On this camshaft are mounted the cams used to actuate the inlet, exhaust, air- and fuel-injection valves. The fuel pump is also driven by camshaft A, the amount of fuel charge being controlled by the governor. The fuel is deposited in the fuel valve *C*, out of which it is forced by the air charge at the proper instant. The air blast is supplied by the two-stage air compressor *D*, which is driven by a crank on the end of the engine crankshaft. In this arrangement the air discharge line is not provided with an air receiver, and the air is delivered directly from the compressor to the fuel valve *C*. In addition to the valves shown, one or more cylinders of an engine are provided with air-starting valves which are used to supply compressed air to these cylinders for starting the engine.

When air injection of the fuel is employed, the pressure of the air for the blast must be considerably higher than the highest cylinder compression pressure. The fuel is injected with a blast of air at a high velocity, and is thus atomized and distributed through the main mass of compressed air in the cylinder. The air-injection pressure varies from 700 lb per sq in. for light loads to 1200 lb for maximum loads. The auxiliary air is generally compressed in a two- or three-stage compressor driven directly by the engine; the compressor uses 5 to 15 per cent of the power developed.

48. Two-stroke Diesel Cycle. Solid Injection of Fuel. The power used by the injection-air compressor may be eliminated, the mechanism of the engine simplified, and the first cost decreased by using solid injection of the fuel, also called airless injection or mechanical injection. Oil is forced into the cylinder at the proper time, through specially designed nozzles that produce the necessary fineness of spray, by one or more pumps at pressures of 2000 to 8000 lb per sq in., depending upon the load.

To reduce the size, weight, and cost for a given power, Diesel engines are built in increasing numbers to operate on the two-stroke cycle. Many recent installations of this kind have shown fuel economies, pound of oil used per brake horsepower-hour, equal to those obtained with four-cycle engines; see Table XVI. Figure 64 shows the action of a two-cycle solid-injection Diesel engine in which the completion of one cycle of operation is accomplished in two strokes of the piston. At *A* the scavenging air ports have been uncovered by the piston and air under a pressure of about 2 lb per sq in. is admitted to the cylinder from the crankcase. On entering the cylinder this air is deflected upward by the contour of the piston head and the shape of the intake ports, thus clearing the cylinder of exhaust gases as shown. On the upward stroke of the piston, first the air ports and then the exhaust ports are covered at about 50 degrees past bottom dead center.

The upward motion of the piston compresses the air charge to within a few degrees of top dead center when injection of fuel oil begins and combustion occurs. After a momentary pressure rise above the compression pressure, during which the piston passes its top dead center position, the piston continues its accelerated downward movement, *B* in Fig. 64, on the working stroke owing to the combustion of the fuel and expansion of the gases. As the piston moves farther down, the exhaust and scavenging air ports are uncovered and the cycle is repeated. The crankcase is used as an air pump for supplying the scavenging air; on the upstroke of the piston, *A* in Fig. 64, air is drawn through an automatic air valve into the crankcase and is there compressed on the downstroke of the piston. Many two-cycle engines use a reciprocating pump or rotary blower attached to or driven by the engine to supply the necessary scavenging air.

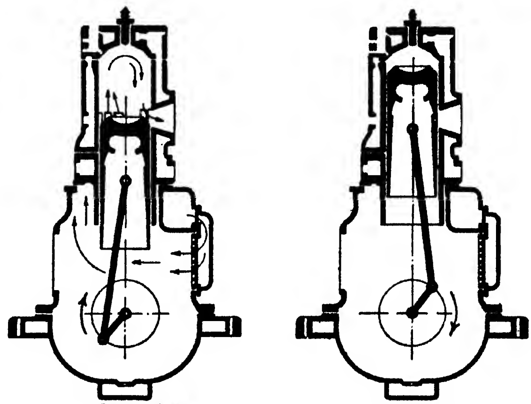


FIG. 64. Cylinder events of a two-stroke cycle solid-injection Diesel engine.

49. Four-cycle Air-injection Diesel Engine. The Fulton Type-BG, Fig. 65, is typical of this class of Diesel engines. An eight-cylinder engine, $17\frac{1}{2}$ in. $\times$ $24\frac{1}{2}$ in., of this type operating at 225 rpm is rated at 1000 bhp. This design provides a large cooling water space which extends the full length of the cylinder liner. The cooling water entering the cylinder head is directed to the center around the fuel-injection valve and between the intake and exhaust valves, thus insuring adequate cooling of the area where the greatest amount of heat is produced. The rocker arms are operated directly from the camshaft; pushrods are eliminated and a close setting of the valves is made possible. The removable cylinder liner is provided with a slip joint at the lower end to permit unrestricted expansion.

The lower casting or base plate contains all the crankshaft main bearings. The main bearings consist of babbitt-lined cast-iron shells which can be removed by raising the shaft slightly. The vertical engine frame is bolted to the base plate and is so constructed as to form the cylinder block which is bored for the reception of the cylinder liner. This type of framing transmits all stresses of vibration, compression, and expansion from the cylinder head to the bed plate in direct straight lines in contrast to the water-box type which is only half as deep as the distance between cylinder head and camshaft.

The piston is of the trunk type and made extra long to reduce wear, since no crosshead is used; pistons above 20 in. in diameter are water-cooled. The connecting rod is forged in round section from 0.45 per cent carbon steel, the upper end is provided with an integral eye bushed with babbitt for the wrist pin bearing. The piston end of the rod is

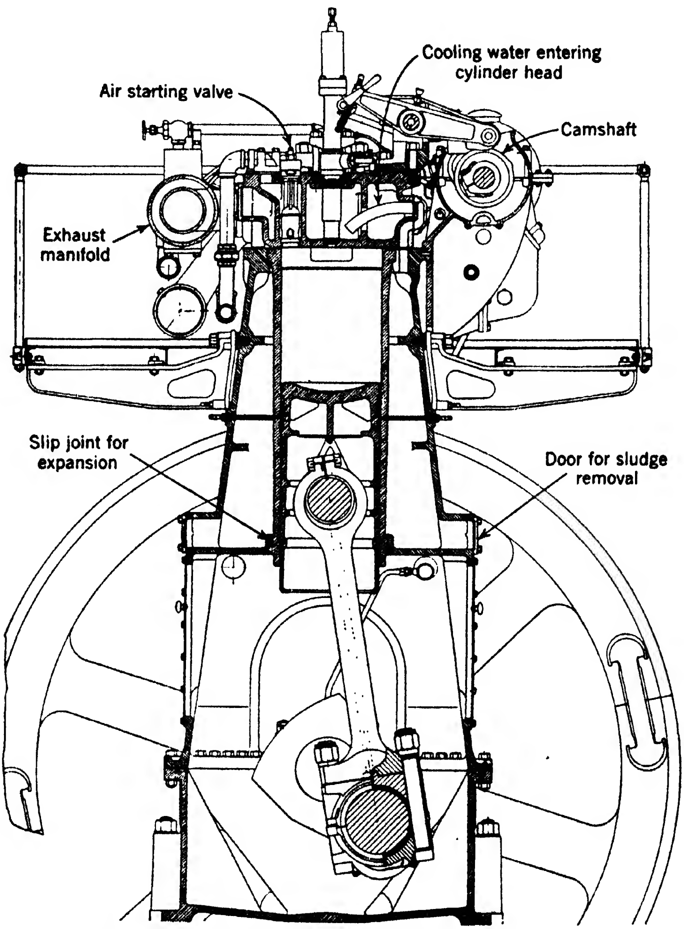


Fig. 65. Section through cylinder of four-stroke cycle air-injection Fulton Diesel engine.

provided with a patented lock plate to facilitate fitting the piston pin bearing; the lower end is fitted to a cast-steel crankpin box provided for making adjustments by the use of shims. The connecting rod is drilled to allow the flow of lubricating oil from the crankpin to the piston pin.

A fuel pump of constant-stroke type with separate plunger for each cylinder is employed; it is mounted conveniently at the level of the camshaft. The entire pump is submerged in fuel oil as the only positive means of preventing air leaks; air would disrupt the accurate measurement of fuel during the suction stroke of the pump plunger. The cylinder head contains the air-inlet valve, exhaust valve, air-starting valve, and fuel-injection valve. Inlet, exhaust, and fuel-injection valves are housed in removable cages; inlet and exhaust valves are interchangeable. The fuel oil is blown into the cylinder by means of high pressure air furnished by a three-stage air compressor.

50. Four-cycle Solid-injection Diesel Engines. One type of heavy-duty four-cycle solid-injection Diesel engine is shown in Fig. 66, which is a vertical section through one cylinder. These engines are built with three to eight cylinders, varying in capacities from 125 to 200 bhp per cylinder. The inlet and exhaust valves are located in the cylinder head, which is completely water-cooled and designed for the De La Vergne two-nozzle fuel-injection system. The combustion chamber is shaped into the cylinder head and consists essentially of two cone-shaped sections. At the apex of each cone, and located at opposite sides of the cylinder head, is a fuel nozzle. These point along the cone axes at angles of about 15 degrees from the horizontal.

Each cylinder has an individual fuel pump which delivers fuel oil to the two nozzles. Each fuel nozzle has one orifice and discharges the oil toward the center of the combustion chamber, which, because of its shape, prevents the spray from impinging on the cooled walls and promotes intimate mixing of the oil and compressed air. The piston crown is conical and fits closely into the lower cone-shaped part of the cylinder head. As the piston approaches the top of the compression stroke it forces the air through the throat from the cylinder proper to the combustion chamber, thus creating considerable turbulence in the conical combustion chamber at the instant when the fuel is injected. This throat is completely water-cooled and forms a head shield between the combustion chamber and the piston. Hence, since no special arrangement for cooling the piston by oil or water with the attendant complications is necessary, this design can satisfactorily use a large uncooled piston.

McIntosh and Seymour four-cycle Diesel engines are built both as air-injection and solid-injection types. The four-cycle solid-injection engines are built in units from 225 to 1090 horsepower. The 1090-bhp engine employs eight cylinders, $17\frac{1}{2} \times 25$ in., and operates at 240 rpm. The enclosed design of this engine, Fig. 67, makes possible forced feed lubrication for all moving parts. It also shields bearing surfaces from dust and grit—particularly essential in flour mills, cement mills, etc. The cylinder heads are individual castings and contain removable

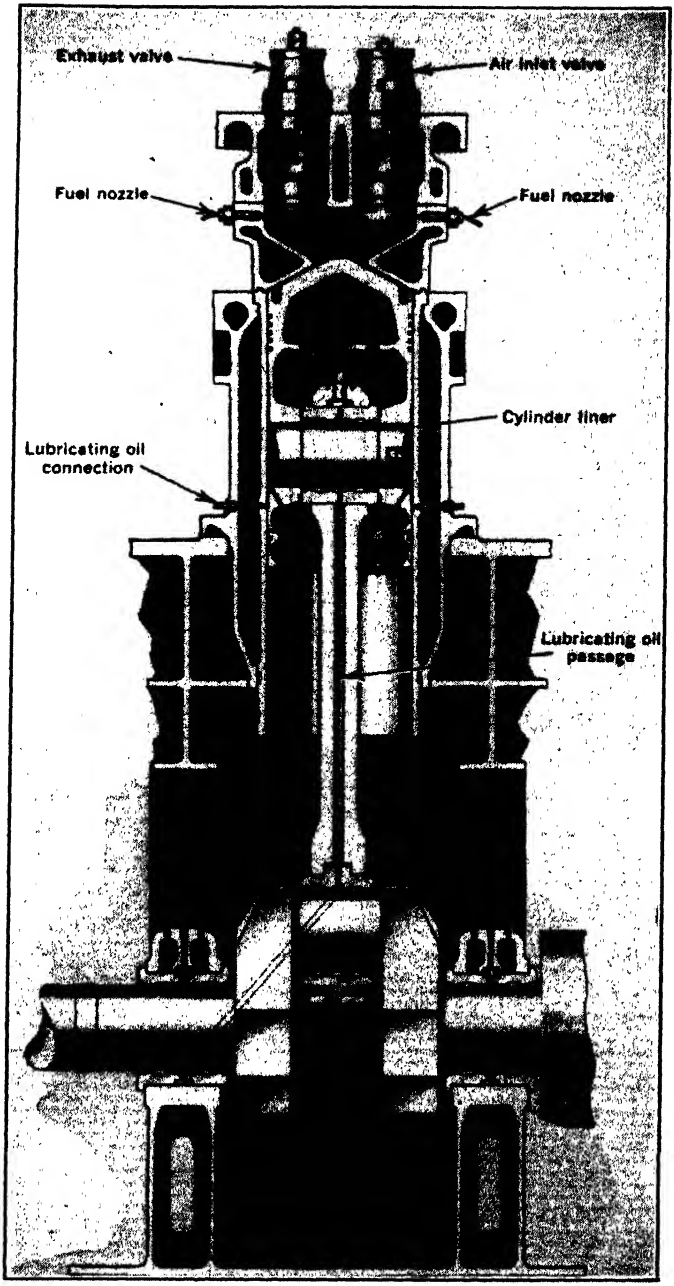


Fig. 66. Section of cylinder, De La Vergne four-stroke cycle solid-injection Diesel engine.

starting, inlet, and exhaust valve cages. Removable cylinder liners are used (see cylinder 2, Fig. 67), and are slip-fitted into the cylinder. The concave surface of the piston head is designed to aid the distribution and atomization of the oil spray as it is injected into the cylinder. Pistons exceeding $12\frac{1}{2}$ in. in diameter are oil-cooled; in the very large sizes, water cooling is used.

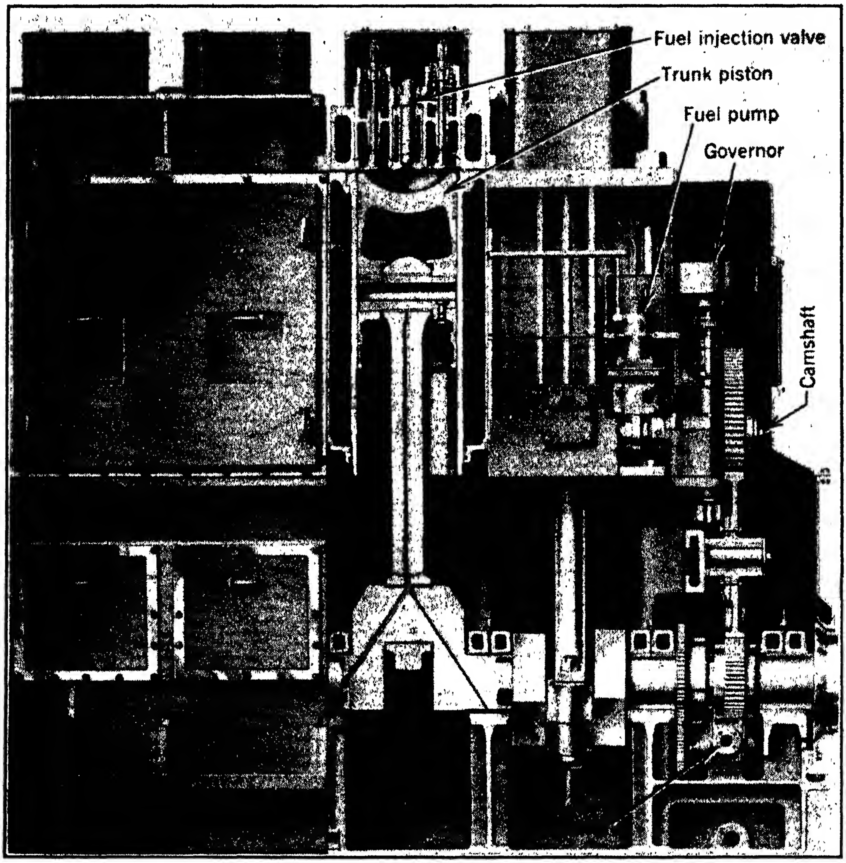


FIG. 67. Elevation and section, showing four of six cylinders of a McIntosh and Seymour four-stroke cycle solid-injection Diesel engine.

A small pump, geared to the crankshaft, forces lubricating oil to crankshaft bearings, connecting-rod boxes, and through the hollow-drilled connecting rods to piston pins where the oil drips down the inside of the piston to the bottom of the engine base. Here it is drained into a sump tank, from which it is pumped and passed through lubricating-oil coolers before repeating its circuit. Another circuit leads to the camshaft bearings and rollers, and through the pushrods to cylinder valves and rocker arm assemblies. There is an individual fuel-injection pump for each engine cylinder. The pumps are of the plunger type and actuated from a single camshaft. The fuel-pump stroke (charge) is controlled through a link connection, see Fig. 67, between the governor and fuel pumps.

The cylinder head contains the main inlet and exhaust valves, the fuel-injection valve, and the air-starting valve (not shown in Fig. 67). These valves are placed in cages, any one of which may be removed without disturbing the others. The pistons in Figs. 66 and 67 show effectively the massive construction necessary in an engine of the Diesel type, where maximum pressures are from four to six times the pressures obtained in Otto-cycle (gas and gasoline) engines.

Several makes of solid-injection engines, usually less than 100 hp per cylinder, employ the *precombustion* (also called two-stage or

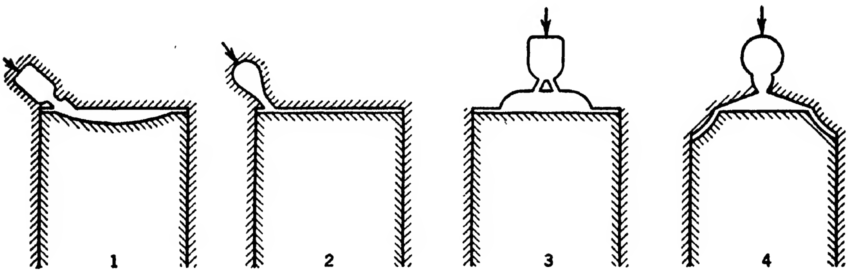


FIG. 68. Precombustion types of cylinder head arrangement for solid-injection Diesel engines.

divided combustion) principle of fuel injection and atomization. In this design the combustion of the atomized fuel is accomplished by means of the heat of compression, exactly as in the true Diesel engine. The sectional views of Fig. 68 show the cylinder-head arrangements of four precombustion types of engines. The operation of the auxiliary chamber is based on the fact that every fuel oil, no matter how heavy it may be, contains some light hydrocarbons that will vaporize at a fairly low temperature. The explosion of these lighter parts of the fuel in the auxiliary chamber provides the means whereby the remainder of the fuel is injected into the cylinder in a finely divided atomized condition. A further matter of common knowledge is that the temperature of ignition of an oil is somewhat dependent upon the degree of atomization.

The fuel pump forces oil through the fuel-supply line to the fuel nozzles on the cylinder head, Fig. 68. The oil is usually injected into the precombustion chamber near the end of the compression stroke;

as the piston moves upward, with all valves closed, the pressure of the air in the cylinder and precombustion cup will increase to nearly 500 lb per sq in. at the end of the stroke. As the pressure rises in the compression space of the cylinder, it also rises inside the cup because of the restricted connection; but the pressure does not rise as rapidly in the cup as in the cylinder. At 400 lb pressure or less the temperature is sufficient to start ignition of the oil within the cup. The resultant high pressure (about 700 lb per sq in.) forces the rest of the oil out through the restricted passage into the combustion space and cylinder.

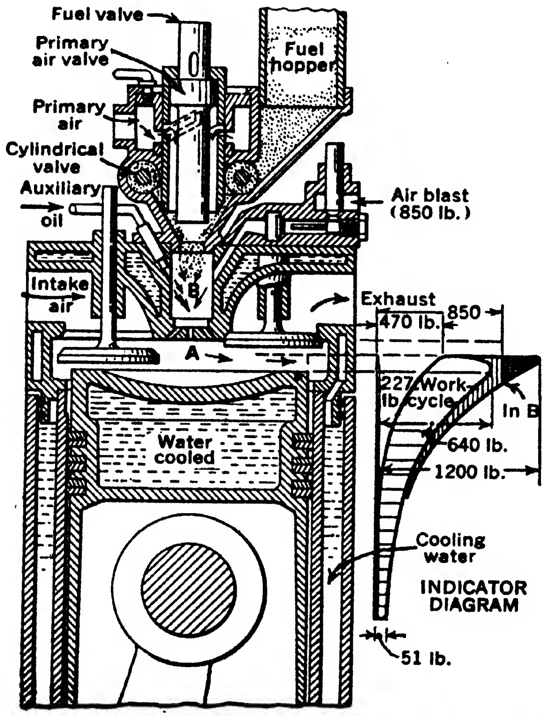
The movement of the fuel from the first stage to the second stage is determined and controlled mainly by the downward movement of the piston, which causes a differential pressure between the small chamber and the top of the cylinder. However, the spraying of the fuel into the cylinder is started and assisted by the precombustion that takes place during injection. It is evident that the fuel cup performs two important functions: first, delays and therefore times the ignition; second, atomizes the fuel as it is forced through the small passages and thereby aids combustion. The amount of fuel consumed in the auxiliary chamber is small because there is only a limited amount of air present in the cup to support combustion. As the fuel in an atomized and vaporous state comes into contact with the heated air in the combustion space, very rapid combustion takes place as the piston starts downward on the expansion stroke. The fuel oil burns with an increase in temperature at nearly constant pressure, and the gases expand and do work on the piston.

Combustion in this type of engine, Fig. 68, is fixed by the size and shape of the injection (precombustion) chamber, the fuel used, the outlet passage arrangement, and the fuel spray. Two-stage combustion has the advantage of limiting the rate of burning of the oil during injection, so that accurate timing and careful injection control is not necessary. In the timing of direct mechanical injection, wherein the fuel meets the whole air charge at once, the combustion control required is represented by only a few degrees of crank angle.

51. Four-stroke Cycle Diesel Engine Using Solid Fuels. An interesting application of the precombustion chamber is the "Rupamotor," Fig. 69, a Diesel engine developed by Rudolph Pawlikowski, a German engineer, to operate on any solid or liquid fuel that will burn.[1] These fuels include pulverized coal, lignite, peat, coke, potato peelings, sawdust, leaves, wood residues, pine needles, hay, corn and cereal stalks, soybeans, fruit peelings, cotton hulls, rice hulls, palm seeds, residues of peanuts, olives, palm, and cocoanuts (after oil extraction). Fuel

[1] "This Engine Runs On Anything That Will Burn," *Power*, page 258, May, 1936.

from the hopper is metered through the cylindrical valves into a concentric area around the lower end of the fuel valve which seals the opening into precombustion chamber B; the upper end of the fuel valve controls the primary air supply. The fuel valve opens shortly after the beginning of the suction stroke, the fuel falls into B, and a regulated amount of primary air passes between the fuel valve and the cylindrical valves; the air enters through two sets of spiral slots on the fuel-valve plunger. This primary air mixes with the fuel and also cleans the seat of the fuel valve as it closes. The primary air and the fuel are drawn in by the downward movement of the piston, making it possible to operate the engine without pressure on either air or fuel. The secondary air, which supports combustion in the cylinder, enters through the intake air passage as indicated in Fig. 69.



Courtesy of *Power*.

Fig. 69. The "Rupamotor" four-stroke cycle solid-injection precombustion-chamber Diesel engine.

There is an air-blast injection valve that can admit auxiliary air as required at about 850 lb pressure. This intermediate air valve is kept closed until the main fuel charge is blown by precombustion from chamber B into the cylinder, then the intermediate air blast (if used) blows the slow-burning fuel particles into the cylinder and also aids in scavenging the chamber. Generally, the chamber can be kept clean and free from ash without this air. Figure 69 also shows an auxiliary oil nozzle terminating in chamber B. This permits the engine to be started and operated on oil, or the nozzle may be used to admit oil as a supplementary fuel if the primary fuel is hard to ignite, as when it is too damp, too coarse, or contains too much ash.

Mr. Pawlikowski compares operation of his engine with that of a Bunsen burner, in that successful combustion requires addition of some air before the fuel is ignited. Primary air mixing in the Rupamotor is like the air addition at the base of the burner. Hydrocarbon molecules in the fuel undergo early splitting up and preoxidation, because they are heated while still enveloped in primary air. The more volatile parts evaporate first, the heavier particles crack, and their hydrogen goes into the formation of benzine, probably accompanied by dissociation of any moisture present. Then burning takes place in the secondary air. Just as in a Bunsen burner, adjustment of the primary air affects the intensity and duration of the partial combustion in chamber *B*, therefore also in the cylinder *A*. The indicator diagram drawn to the right of the cylinder in Fig. 69 shows the pressure variations in chamber *B* and in the cylinder during the cycle. For a maximum pressure of 640 lb in the cylinder, the explosive pressure in chamber *B* will reach 1200 lb.

Very little difficulty has been encountered from the ash, as one engine converted from a Diesel has shown only $\frac{1}{4}$-in. of cast-iron cylinder wear in 20 years of testing with various fuels while supplying power for the company's machine shop. During this time the piston rings were changed occasionally to compensate for the cylinder wear. The use of improved materials for cylinder liners and pistons should materially reduce this wear. Thermal efficiencies for the Rupamotor range from 32 per cent for pulverized anthracite coal to 21 per cent for finely ground forest and field residues.

52. Two-cycle Air-injection Diesel Engines. The Nordberg Manufacturing Company of Milwaukee, Wisconsin, builds both four-cycle and two-cycle Diesel engines. This company builds the four-cycle type in sizes from 150 to 1000 hp, the two-cycle single-acting type in sizes from 750 to 5800 hp, and the two-cycle double-acting types in sizes from 7500 to 22,500 hp. The two-cycle types, built either as air-injection or solid-injection engines, use port scavenging with the elimination of intake and exhaust valves; this results in a unit of extreme simplicity with a minimum of moving parts. This type of engine is built in 17 × 25 in., 21 × 29 in., and 29 × 44 in. cylinder sizes and can be furnished in any number of cylinders per unit, capacities of the three sizes being 250, 375, and 725 bhp per cylinder, respectively. In the design of the engine can be incorporated either the crosshead- or trunk-type pistons, which can be arranged for either air or solid injection.

Figure 70 shows a sectional end elevation of a two-cycle air-injection Nordberg Diesel engine. This engine is of the single-acting, crosshead type, with oil-cooled piston. The scavenging and exhaust ports, clearly seen in Figs. 70 and 71, will be described more fully later. The simple cylinder head is a distinct feature, containing only the fuel-injection valve and the air-starting valve. Each cylinder is provided with a constant-stroke fuel pump. The governor, acting in conjunction with each fuel pump, by-passes that amount of fuel which is in excess of the quantity required for the engine load. The fuel-injection valve is so constructed as to distribute the fuel directly and atomize it thoroughly as it is forced into the cylinder. Air injection is capable, with proper

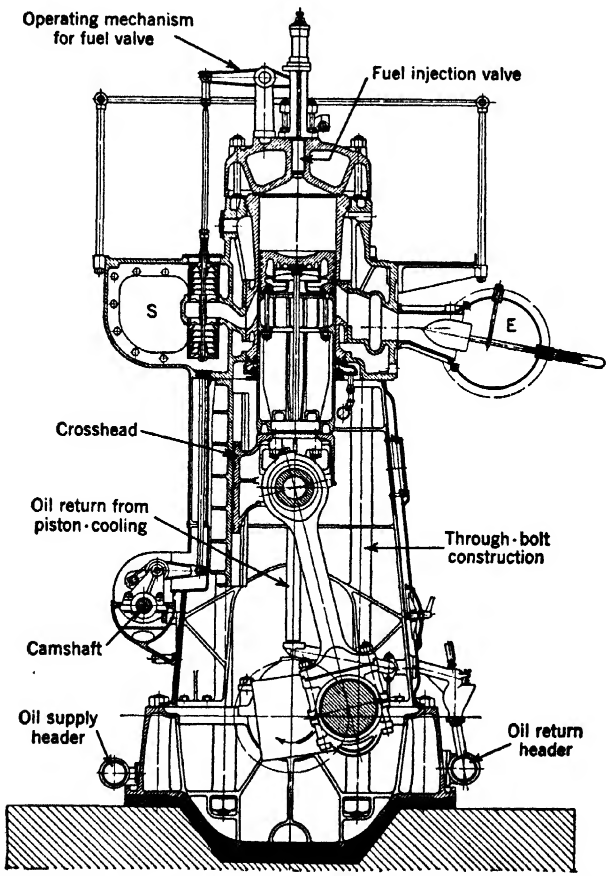


FIG. 70. Section through power cylinder of Nordberg two-stroke cycle air-injection Diesel engine of crosshead construction.

heating devices to render the fuel fluid, of continuously operating on fuels as low as 10 degrees Baumé.

Scavenging air for the power cylinders is provided by a double-acting pump located on the bed plate at the end opposite the flywheel. This pump is driven from a crank on the crankshaft, see Fig. 72, and delivers low-pressure air into the scavenging manifold *S* extending the full length of the engine and connecting with the power cylinders. The three-stage tandem air compressor is superimposed on the scavenging pump, as indicated on Fig. 72, and driven from an extension of the scavenging pump rod. An intercooler is placed between the second and third stages and an after-cooler after the third stage. The coolers, consisting of separate units mounted on the compressor frame, are made of copper coils, the air passing through the coils, the water being on the outside. The compressor furnishes air for starting as well as for fuel injection. The air-starting receiver pressure is not allowed to fall much below 500 lb per sq in.; pressures as low as 200 lb may be used, however.

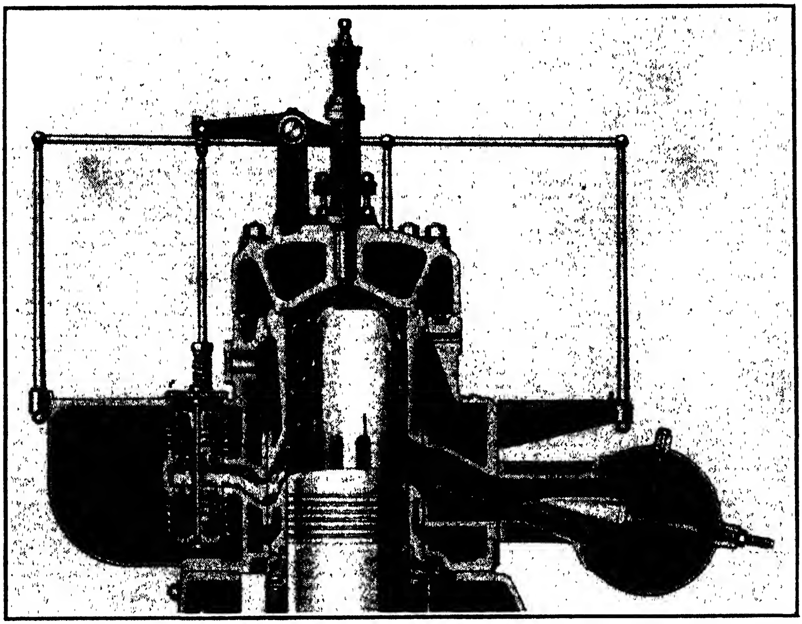


FIG. 71. Nordberg system of scavenging a two-stroke cycle Diesel engine cylinder.

The cylinders are of block-type design, Figs. 70 and 71, provided with cast-iron liners. On one side of the lower end of the liners are the scavenging ports, and on the opposite side are the exhaust ports which register respectively with corresponding openings into the scavenging air manifold S and exhaust header E. Cast integral with the cylinder block is the scavenging header, the top of which is finished in floor-plate design and forms the overhead working platform. Inside the scavenging air manifold S are placed two nests of automatic check valves, Fig. 71, which control the passage of air from the manifold to the power cylinders and prevent any blow-back should the exhaust pressure be higher than that of the scavenging air when the ports are uncovered.

As the piston approaches the bottom of the stroke the exhaust ports and the scavenging ports are uncovered. The exhaust ports are uncov-

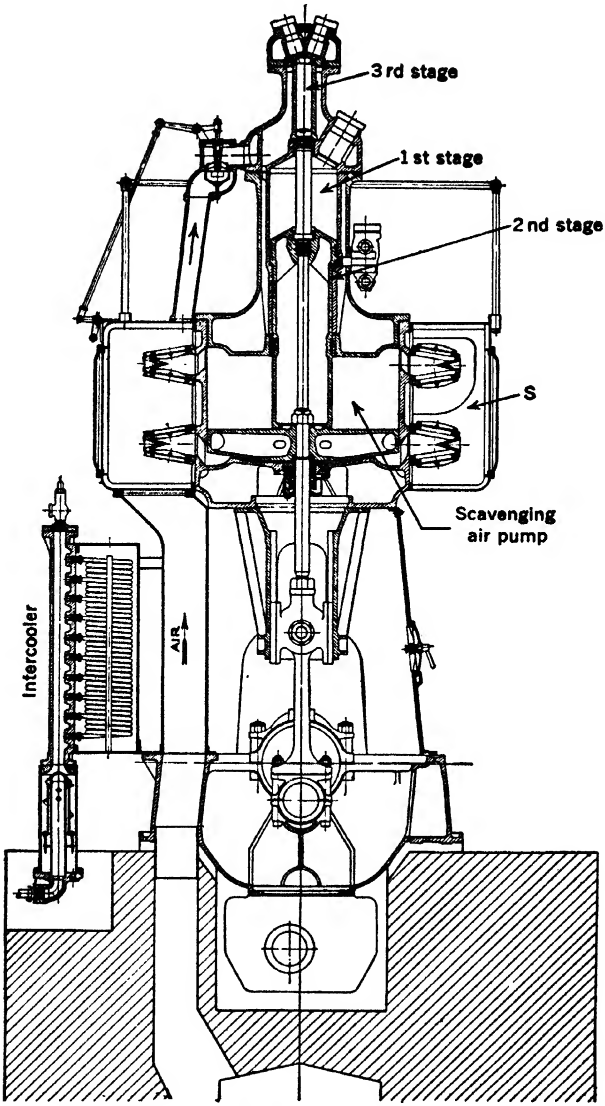


FIG. 72. Section showing scavenging pump and three-stage air compressor for air-injection Diesel engine.

ered first and consequently close last. The pressure in the cylinder falls rapidly as soon as the exhaust port is uncovered and soon thereafter the air scavenging port is uncovered, admitting air at about 4 lb pressure which is deflected diagonally upward by the inclined port. This removes practically all the exhaust gases, besides filling the cylinder with fresh air for combustion during the next cycle.

The venturi-shaped exhaust passage aids in maintaining a constant back pressure at the cylinder port, thus minimizing the effect of the multi-cylinder pulsations in the exhaust header *E*. The size and shape of this opening are affected by the number of cylinders and conditions of exhaust existing in the plant in which the engine is installed. The size of the venturi exhaust passage of each cylinder may be varied by means of a cone-shaped bulb, installed as shown in Figs. 70 and 71.

The piston is of two-piece construction. The head contains the cooling arrangement, to which is bolted the skirt or trunk; this trunk, in turn, is bolted to the crosshead and transmits the piston load directly to the crosshead pin. The piston is oil-cooled, see Fig. 73, the cooling medium being the same oil that is circulated through the engine for lubrication. The circulating pressure pump delivers a large excess over that required for bearing lubrication, and this excess is passed through the crankshaft, connecting-rod, and crosshead-pin bearings. From the crosshead-pin bearing, Fig. 73, a connection is made to convey the oil to the piston head and maintain a uniform temperature. Upon entering the piston head the oil is made to circulate (return) at high velocity through a specially running telescopic pipe connected to a header (see Fig. 70) which drains back to the sump for recirculation.

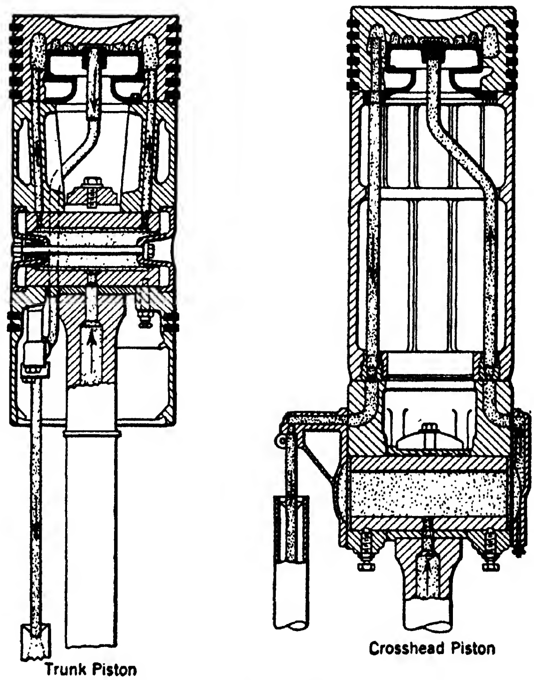


Fig. 73. Piston cooling arrangement for trunk and crosshead design.

53. Two-cycle Solid-injection Diesel Engines. The Model 33-D Fairbanks-Morse is a typical two-cycle solid-injection Diesel engine. Figure 74 shows a section through one-power cylinder and important connecting parts. The engine embodies oil-cooled trunk pistons, open-head combustion, differential fuel-injection nozzles, back-flow scavenging, and special oil-wiper rings. These engines are built with three to

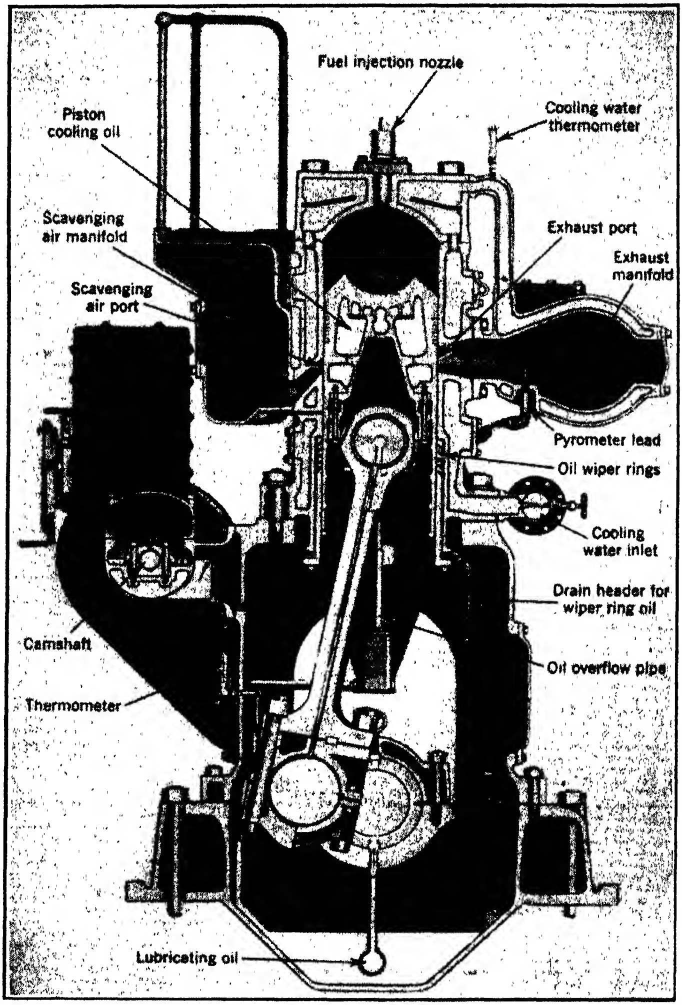


FIG. 74. Section through cylinder of Fairbanks-Morse two-stroke cycle solid-injection Diesel Engine.

eight cylinders in capacities varying from 250 to 1400 bhp; the 1400-bhp unit employs eight 16 × 20 in. cylinders and operates at 300 rpm.

The cylinder, Fig. 74, is completely water-cooled with large jacket spaces extending more than its working length. The liner is cast integral with the cylinder and has a liberal wall thickness. This construction results in a good distribution of the forces from the cylinder to the frame. The lower part of each cylinder is bored to a larger diameter than the piston. Five lubricating-oil wiper rings are placed in this recessed portion of the cylinder and are held in place by a sleeve which is bolted to the bottom of the cylinder. These rings prevent contamination of the main oil supply in the lower base by collecting any oil which has been in contact with the products of combustion. A small pipe header conducts the oil collected by these rings to a separate sump, whence it is passed through the purification system and thence back into the oil circulating system.

The cylinder head is a simple one-piece casting of symmetrical design; the inner surface is spherical, thus permitting a minimum thickness of metal with consequent rapid flow of heat to the large volume of cooling water. Cooling water is admitted to the jacket of each cylinder at a point near the base from which passages lead to the head; see Fig. 74. The cooling-water outlet is at the top of the cylinder head. The exhaust manifold is cooled throughout its entire length by the water discharged from the cylinder heads of the engine. Each cylinder is arranged for the admission of starting air and the cylinder head is therefore fitted with a simple check valve assembled in a removable cage. This check valve automatically opens when the starting air is admitted by manipulation of the control wheel but is closed at all other times. The fuel-injection nozzle, of the differential spring-loaded, needle-valve type, is automatically operated by the fuel pressure, thus eliminating the cams, pushrods, rockers, etc., which are required for the mechanically operated type.

The cam-operated fuel pumps, one for each cylinder, are located on a deck (left side in Fig. 74) which forms the bottom of an auxiliary fuel-oil reservoir, kept filled by an auxiliary fuel pump. All the pumps are thus submerged in the fuel oil, an arrangement which eliminates piping to the pumps, establishes a perfect air seal, and enables one fuel suction line and one return line to serve any number of cylinders. The governor consists of centrifugal weights and suitable linkage for transmitting their action to the fuel pumps so that the individual fuel charges are just sufficient for maintaining the desired speed under the existing load. The length of the fuel-pump stroke is constant, the amount of fuel going to each cylinder being metered by by-passing the excess according to the position of the governor.

Air for scavenging is supplied by a special double-acting built-in scavenging pump at one end of the engine which is operated directly from the crankshaft. The capacity is so designed as to maintain a pressure of about 2 lb in the scavenging air manifold. When passages for conducting the scavenging air to the cylinder are uncovered by the downward movement of the piston, the air is admitted to the cylinder. On entering the cylinder this air is deflected upward by the contour of the ports and the piston head, thus driving the burnt gases out of the cylinder through the exhaust ports on the opposite side. On the next upstroke, first the air ports and then the exhaust ports are covered and compression begins.

Each connecting rod is made of forged steel; the piston-pin end is bored out and the lower end is fitted with a cast steel babbitt-lined crankpin box in two halves. The piston is of the trunk type, being long in proportion to its diameter. This design reduces the side wear and increases the life of both piston and cylinder. The piston is an unbroken cylindrical shell, a construction which is accomplished by bolting the piston-pin housing on the inside. Circular ribs are provided around the inside of the piston, Fig. 74, which permits light construction with great strength. Six piston rings are used on each piston. They form a gas-tight joint and provide a heat-transfer medium to the water-cooled cylinder wall. The steel casting, which forms the piston-pin housing and is bolted to a flange inside the piston, also forms the inner shell of the space through which oil is circulated for cooling the piston. This construction eliminates the piston-pin cross-boring generally found in other trunk-piston engines.

A built-in high-pressure pump circulates lubricating oil to a header in the lower part of the crankcase, Fig. 74, from where it is distributed by branch lines to all main bearings. A circulation groove cut in the inner circumference of each main bearing registers with a hole in the crankshaft and permits the major part of the oil to pass through the crankshaft to the crankpin bearing. In the same manner, a circulation groove in each crankpin bearing registers with a hole in the crankpin, allowing oil to flow continuously from the pin to the groove. This groove, in turn, connects with a hole in the upper half of the crankpin bearing and the drilled connecting rod. A hole through the piston-pin bearing permits the oil to enter the drilled piston pin. Because of the clearances between crankshaft journals and bearings, crankpins and bearings, piston pins and bearings, sufficient oil is directed from the main flow to lubricate the bearings.

Piston cooling is accomplished by continuous circulation of cooled oil through the passage in the piston crown, Fig. 74. Oil has a number of advantages over water for cooling pistons. No mechanical construction yet developed insures a continuous supply of cooling water without leaks, and when water mixes with lubricating oil in the crankcase, a dangerous condition results. In this method of piston cooling no stuffing boxes are required and the construction is simple. From the piston pin the oil passes through a spiral groove in the piston crown, into the jacketed portion of the piston and then overflows through a pipe to an oil outlet pocket situated in the upper part of the crankcase. From this outlet pocket the oil is discharged into the lower base, where it mixes with the oil from the bearings. It then goes through the oil cooler and is recirculated.

The Busch-Sulzer Type-DH engines, Fig. 75, also belong to this class of Diesel engines. An eight-cylinder engine, $19\frac{1}{2}$ in. $\times$ 27 in., of this type operating at 240 rpm is rated at 2400 bhp. The distinctive features of this engine are the design of the cylinder wall and three-piece trunk piston. A separate piston-pin housing provides full-length bearing for the piston pin, reducing the bearing pressure per square inch about one-third, practically eliminating heating and, with forced lubrication, making the wear negligible. The pin turns in the housing and is bolted to the connecting rod. The port-scavenged Busch-Sulzer type of two-cycle engine, with pressure always on top of the piston, lends itself well to this construction, the pressure being constantly downward on top of the pin. This separate housing permits an unpierced piston skirt, avoiding distortion due to assembly of a tight-fitting pin with two press fits; or later heat distortion, due to warming up of the pin during operation. This three-piece design permits a piston head of forged steel, an excellent material for large Diesel pistons.

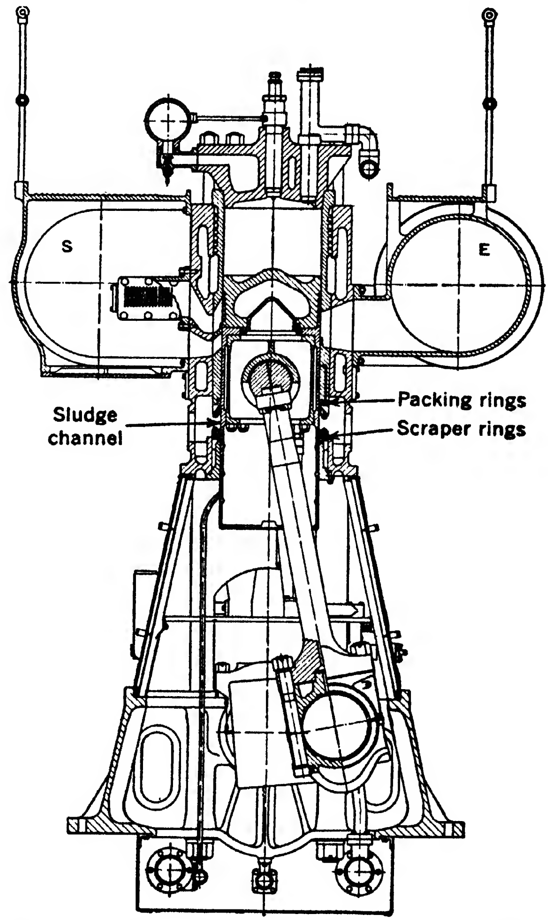


Fig. 75. Section through cylinder of Busch-Sulzer two-stroke cycle solid-injection Diesel engine.

In the lower part of the cylinder jacket is provided an annular open chamber through which the piston is at all times under the observation of the operator. Any lack of proper lubrication, or undue heating of the piston because of insufficient cooling medium, is at once noted and corrected while the engine is in operation and before damage has occurred. A hot piston turns brown; in contrast, a cool properly lubricated piston glistens; and one brown piston among several bright, shiny pistons immediately attracts attention and invites investigation by the operator. If increase in lubrication or cooling medium does not restore normal safe operating conditions, the particular piston may be relieved of load by cutting off its fuel.

Above and below this inspection chamber are cast-iron packing rings, held in place by bolted glands, the upper rings scraping from the piston into sludge channels lubricating oil from the combustion chamber portion of the cylinder, and the lower rings removing and draining back into the crankcase clean lubricating oil carried by the upward moving piston. Contamination of the crankcase lubricating oil by sludge or by combustion gases blowing past the piston is thus prevented. The piston becomes longer with the addition of the sludge chamber to the lower end of the cylinder; and this lower portion, with its two sets of scraper rings, acts with the lengthened piston to form a lower guide, eliminating vibration, the tendency of the piston to slap, and attendant noises.

The increased area of the long piston and cylinder reduces the wall pressure substantially, with corresponding reduction in cylinder wear. The longer piston also requires a longer connecting rod of 5.25 times the crank-throw, compared with 4.5 times for conventional type of trunk piston engine; and the angularity forces are about one-third less than such forces of the shorter connecting rod of the double-acting crosshead type, the length of which is usually only four times the crank-throw. The elimination of the crosshead reduces the height and weight of the engine about one-third; and as the width of the base is determined by the height, the bed plate is narrower, further reducing the weight of the engine, occupying less floor space, and requiring smaller, less expensive foundations.

54. Two-cycle Double-acting Air-injection Diesel Engines. The power possible per cylinder in a double-acting two-cycle engine is about three times that of a four-cycle single-acting engine, the engines being equal in other respects. This statement is based on the assumption that the ports in the two-cycle engine cut off about 20 per cent of the stroke, leaving 80 per cent effective, and that the piston-rod area in the double-acting engine cuts off 10 per cent of the active area and possible power development on the lower side (crank end) of the cylinder; see

Fig. 76. The thermal efficiency of the double-acting engine is about the same as for the single-acting type. In the two-cycle double-acting engine the auxiliary mechanism, including the scavenging pump, is so arranged as to afford in general as simple, as effective, and as satisfactory a design as for a single-acting engine.

Bearing conditions and lubrication of the double-acting two-cycle engine are the same in character as in the standard steam engine which is double acting, every stroke being a working stroke. In the double-acting engine combustion pressure on one side is working against compression on the other; this gives smoother running conditions and better working control than in a single-acting cylinder, and fewer double-acting cylinders are required for satisfactory operation, independent of power per cylinder.

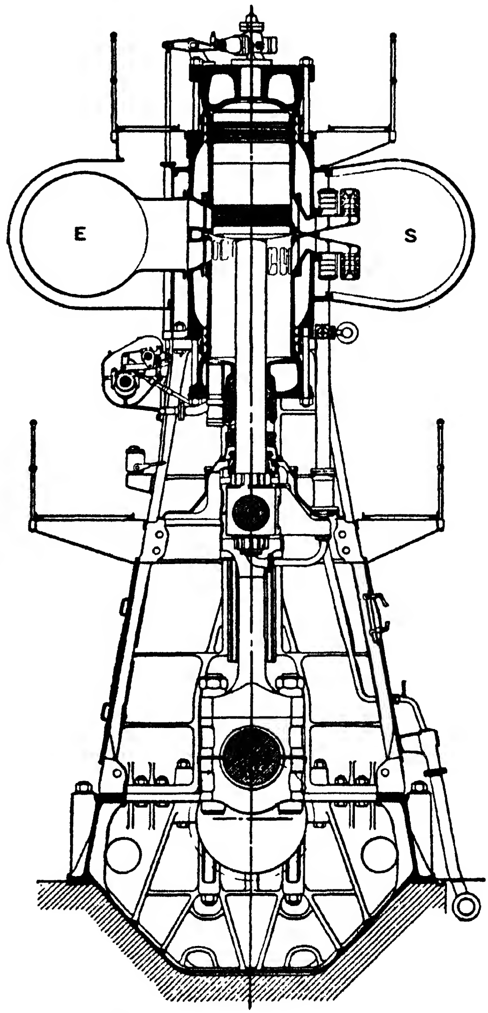


Fig. 76. Sectional elevation through a cylinder of the Sulzer two-stroke cycle double-acting air-injection Diesel engine.

In a steam engine and turbine, the problem has always been to keep the cylinder hot enough to prevent condensation of the steam. With the internal-combustion engine the problem has been the exact opposite: to keep the cylinder cool enough so the metal is not injured and its expansion kept within practicable limits while receiving that part of the heat which must inevitably pass to the walls. For larger cylinders, the quantity of heat per square foot becomes greater and the walls thicker; thus the metal tends to become hotter for these two reasons. Two-cycle cylinders of a given size generate more heat than four-cycle types; so their protection against injury in large sizes, where double-acting pistons are used, is doubly difficult.

The largest Diesel-electric power station in the world is the 51,200-hp plant at the French Concession in Shanghai, China. Sulzer Brothers of Switzerland installed in this plant the largest two-cycle double-acting air-injection Diesel engine in 1934. It is an eight-cylinder, 30 × 47 in., 13,700-bhp engine operating at 136 rpm, and direct-connected to a three-phase alternator of 8000 kw, 5200 volts at 50 cycles.

The Sulzer double-acting two-cycle cylinder, Fig. 76, may be described as consisting of two single-acting cylinders placed opposite each other with their respective pistons flanged to the same piston rod. Actually, the division between the two cylinders is small; just sufficient to provide exhaust and scavenging ports and a frame connection. The comparative sizes of the piston and the cylinder are shown in Fig. 76 with all the related parts in position.

The top and bottom parts of the cylinder are alike except for the piston-rod stuffing box, which in the bottom cylinder occupies the corresponding point to the top fuel-supply valve, and the two bottom fuel valves, which are placed symmetrically opposite each other, entering the cylinder vertically. A safety valve and an air-starting valve are also located in each cylinder head. All valves have vertical axes to avoid uneven wear and are easily accessible. Air-injection engines were selected for the Shanghai installation because the fuel has high asphaltic content. Separate fuel pumps serve the upper and lower chambers. The pumps are arranged in two sets of eight, for upper and lower chambers respectively, and each has its own eccentric drive and regulating mechanism. Fuel is delivered to the injection-spray valve slightly before the spray valve opens; the spray valve is cam-operated and is constantly exposed to injection-air pressure. Injection air is supplied by two three-stage compressors, scavenging air by a double-acting tandem pump, all integral with the engine.

Scavenging air enters the cylinder from *S* (Fig. 76), through two receivers in series, baffles being incorporated for silence. Downward sloping ports in the lower combustion chamber cause air to descend in a stream on the admission side, pass the rod near the cylinder head, then rise to force the products of combustion into the exhaust manifold *E*. The piston, Fig. 77, consists of three parts, the top and bottom parts being bolted together with the cast-iron sleeve to the piston-rod flange. The piston rod is fastened to the crosshead with special opposed nuts which permit adjustment of compression and an even distribution of stresses. An exterior sleeve protects the rod from wear or contact with hot gases, and is fastened to the rod at its upper end only, the lower end being free for expansion. Piston cooling water enters through this annular space and leaves by an axial passage in the rod. No contact with the rod occurs, however, because the rod is surrounded and lined with protective tubing; see Fig. 77. Cooling of the outer sleeve with water keeps the stuffing box cool, therefore lubricated and gas tight. Fresh water only is used for cooling; 1320 gpm for the cylinders and air compressors, 720 gpm for the pistons, are required for full-load operating conditions.

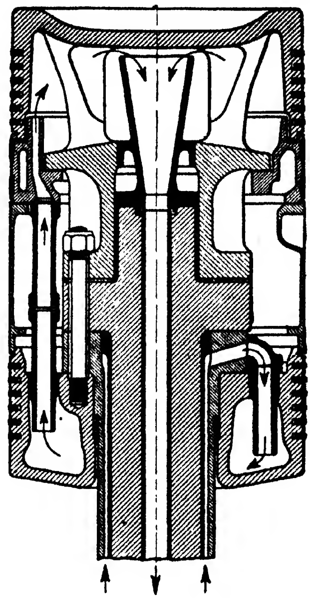


Fig. 77. Section through Sulzer double-acting piston showing construction and cooling water flow.

In double-acting engines cylinder and crankcase lubrication are generally kept separate for maximum economy and best operating conditions. From an elevated tank the cylinder oil flows to engine lubricators; these supply oil in timed injections to cylinder and piston rod surfaces. The various crankcase bearings are lubricated from an entirely separate system. The crankcase shaft assembly is isolated from the cylinder by the stuffing box of a double-acting cylinder, hence the lubricating oil in the crankcase cannot be contaminated by carbon from exhaust gases or diluted with unburned oil. This non-contamination and non-dilution of the crankcase lubricating oil is an outstanding advantage of the double-acting Diesel engine.

55. Two-cycle Double-acting Solid-injection Diesel Engines. The five Hamilton-M.A.N. 7000-bhp units installed in the Vernon, California, municipal plant [2] are worthy of close study. Since June, 1933, these engines have been used to drive the electric generators of this major power station, the largest Diesel station in America and the second largest in the world. A general view of the Vernon plant is shown in Fig. 78. Of particular significance is the simple cylinder-head design and the absence of protruding appurtenances on the sides of the engines.

Each of these units is a 24 × 36 in., 7000-bhp, double-acting two-cycle solid-injection, Diesel engine of eight cylinders operating at 167 rpm, and direct-connected to a three-phase alternator of 7500 kva, 7200 volts at 50 cycles. The mean effective pressure is 66 lb per sq in., and

[2] For complete description refer to *Diesel Power*, pages 614 to 646, inclusive, October, 1933; also "35,000-Hp Diesel Plant a Complete Success," A. B. Newell, *Diesel Power*, page 370, June, 1936.

the piston speed of 1002 ft per min is conservative in view of modern materials and design. Figure 79 is a sectional elevation of one of the power cylinders; much of the following description will refer to Fig. 79.

The cylinder block is separate from the crankcase housing; cylinders for each engine are in two groups of four-in-line, with fuel pumps and other auxiliaries for each cylinder correspondingly segregated. Pistons are made up in four parts—the upper end a steel forging and the lower end a steel casting. These two sections are bolted to a shoulder on the piston rod, and a cast-iron guide ring with a skirt fills the space between the ring and the lower piston end. The pistons carry five angle-jointed

FIG. 78. Five 7000-hp Hamilton–M.A.N. Diesel engines in the Vernon, California, municipal power plant.

standard rings on each end, two of which are of the oil-control type. Pistons are internally water-cooled, the water entering through a telescopic feed line which terminates at the bottom of the piston rod. Water outlets from the pistons are similarly arranged; air chambers are located on both inlet and outlet lines to eliminate water hammer.

In the design of the cylinder heads a two-piece construction is employed. The inner section, which is exposed to the heat of combustion, is made of cast steel with suitable cavities for the cooling water. A cover of cast iron is clamped to the steel castings by bolts; the joint is ground without any form of gasketing.

The steel casting of the upper cylinder head has a central thimble, which provides the opening for the centrally placed fuel spray valve.

Fuel injection into the lower side of the cylinder is necessarily complicated by the presence of the piston rod. The lower cylinder head has two fuel-spray valves placed parallel to the piston rod, as shown in Fig. 79. To prevent flame impingement upon the rod, the lower cylinder head is provided with a separate sleeve encircling the rod and extending into the cylinder cavity a distance equal to the clearance plus a 3-in. recess in the lower end of the piston. The lower head of each cylinder has an opening for an air-starting valve; both the lower and the upper heads have relief valves.

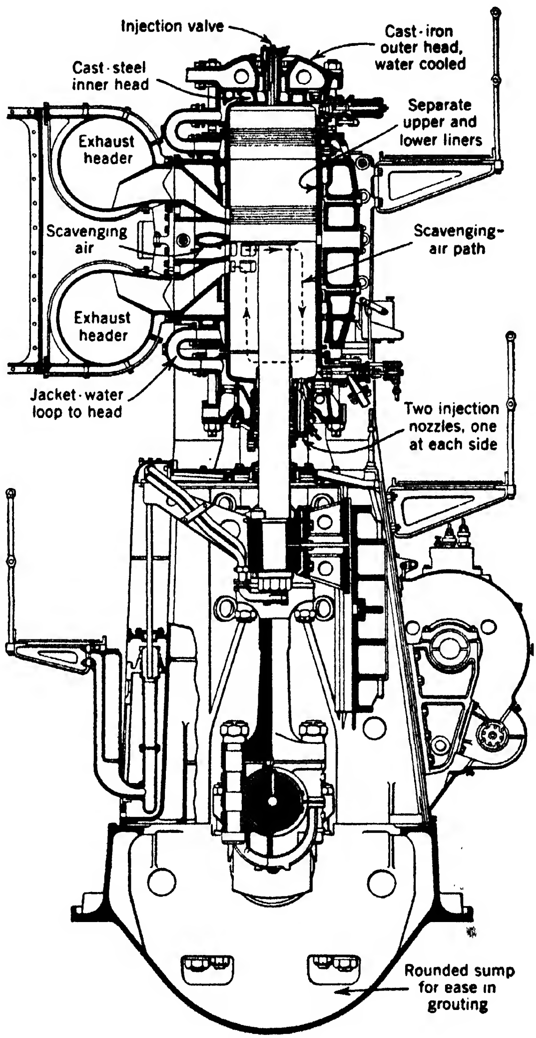


Fig. 79. Section through cylinder of double-acting two-stroke cycle solid-injection Hamilton–M.A.N. Diesel engine.

Into each of the individual cylinder blocks are inserted two liners, one for each cylinder end. As the engine is two-cycle, double-acting, double rows of scavenging air and exhaust ports are provided. These are located in the upper liner, with a sinuous line of contact between the two liner sections. This joint is sealed from the lower row of exhaust ports by two rubber rings, and from the water space by a series of rings. A pipe connects to the lantern thus formed, to give visible signs of any joint leakage.

This engine has the row of exhaust ports placed between the cylinder head and the scavenging air ports. Air entering through the scavenging air ports sweeps across the cylinder, then upward toward the cylinder head, and finally flows downward to the exhaust ports, making a complete loop and forcing the burnt gases out into the exhaust piping. Because of the shape of the scavenging ports, the air is given a circular motion which persists during compression, to give a thorough mixing of air and fuel.

The fuel oil being used is of 26.6 API gravity, 178 F flash, 65 sec viscosity at 100 F, and 1.64 per cent carbon residue. Fuel from the day tank is delivered to the engine injection pump by gravity. The injection pump is of the by-pass control constant-stroke type. The amount of fuel forced through the discharge valve depends upon the timing of the by-pass valve, which is actuated by a control shaft and oil servometer. The latter is under control of the engine governor which is driven from the crankshaft. The governor, drive, and servometer are placed centrally along the front of the engine, and are driven, as is the fuel-pump camshaft, by a train of gears from the crankshaft.

With sixteen power impulses per revolution, the heavy generator rotor supplies all the needed flywheel effect. No separate flywheel is used.

The fuel-spray valves are of the multi-orifice spring-loaded type, set in steel holders. The spring is placed at the approximate center of the assembly, which reduces the length of the needle and thereby eliminates deflection. To avoid carbon trumpets at the orifices, the inner end of the valve cage is water-cooled. All fuel lines are of heavy steel tubing with ball-and-socket connections.

The maximum compression is 430 lb per sq in. for both upper and lower ends of the cylinders, whereas the scavenging air pressure is about 2 lb per sq in. at all loads.

The heat resulting from fuel consumption in the engine is allocated, at full load, as follows:

	Btu per Hour	Percentage
Engine shaft (net hp)....	16,500,000	36.3
Cooling water..........	11,800,000	26.0
Exhaust gases..........	15,400,000	33.7
Friction, radiation, etc...	1,800,000	4.0
	45,500,000	100.0

The net efficiency of 36.3 per cent indicates the nearly perfect combustion and scavenging of these double-acting two-cycle Diesels. The exhaust gas analysis reveals that at full load the air charge is more than 250 per cent of the theoretical amount needed for combustion, and the volume of air handled by the blower, 27,000 cu ft per min, is but 7 per cent more than the cylinder volumes, indicating a low loss due to air short-circuiting from the scavenging ports to the exhaust ports.

Two 100-lb pressure waste-heat boilers per engine recover 62 per cent of the exhaust heat. Gilled rings over 2-in. tubes increase the heat-absorbing area six times. Boilers can be operated as mufflers only (without water) for exhaust temperatures below 750 F. Total heating surface is 1872 sq ft per boiler. Water volume is $36\frac{1}{2}$ cu ft per boiler, generating 2240 lb of steam per hour at 50 lb absolute pressure and $87\frac{1}{2}$ per cent engine rating. Gas temperature leaving boiler is 300 F. Steam heats fuel and lubricating oil before centrifuging, and supplies evaporators. Evaporators are of the horizontal bent-tube submerged type with bowed finned tubes. Only distilled water is used in the double-circuit enclosed type of cooling system. Cooling towers dissipate heat from the engine cooling water, generator cooler, oil cooler, and condenser water.

The 22,500-bhp Burmeister and Wain engine installed in the Copenhagen (Denmark) Power Station also operates on the double-acting two-cycle principle. In this engine, the scavenging ports are at the center, while exhaust gases are discharged through piston-plug valves at the top and annular piston valves around the piston rod at the bottom of the cylinder. Fuel is injected under a pressure of about 6000 lb through atomizing valves, of which there are three at the top and three at the bottom. The eight cylinders have a diameter of 33 in. and piston stroke of 59 in.; speed is 115 rpm. The brake mean effective pressure is 87 lb per sq in. The cylinder diameter of 33 in. is the largest thus far used in Diesel engine construction; and even in their proposals for engines of double the above output, the builders do not propose increasing the cylinder bore. For a 40,000-bhp unit they would increase the number of cylinders to twelve and raise the speed to 187 rpm.

56. Opposed-piston Diesel Engine. An unusual application of the Diesel principle has been made in the opposed-piston engine, shown diagrammatically by Fig. 80. At position 1, the two pistons are shown in their inner dead center positions and the combustion space is filled with compressed air. At this position fuel is injected into the space between the piston heads, when the resulting combustion (and expansion) forces the top piston upward and the lower piston downward. At 2, the pistons are at the end of the expansion stroke with the exhaust ports open (top), just before the opening of the scavenging ports (bottom). At 3, the scavenging ports have been uncovered, thus permitting the entry of the scavenging air which takes a straight course upward and forces out the exhaust gases. At 4, the fresh air charge is being compressed for another combustion cycle.

The engine has three crank-throws per cylinder; two connecting rods from the outside cranks are required to operate the upper piston. The lower or main crank is 180 degrees ahead (or behind) the two cranks for the upper piston. With this arrangement, the explosion pressure is taken up by the connecting rods and results in balanced turning effort, as no pressure is transferred to the main bearings.

The cylinder is a plain barrel, having no cylinder heads and made to withstand only radial stresses from the gases compressed therein. Thus there are no valves in the cylinder heads to complicate the operation and require mechanical arrangements for their functioning. The only

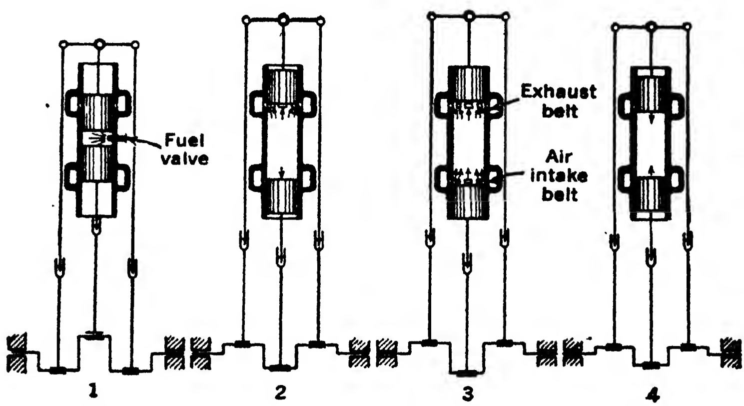


FIG. 80. Schematic diagram of opposed-piston two-stroke cycle solid-injection Diesel engine.

mechanically operated valves are the one or two fuel valves in the side of each cylinder. The absence of cylinder heads and the long strokes adopted reduce the surface exposed to the hot gases to a minimum; hence a high thermal efficiency is obtainable. The uniflow scavenging process is very efficient, inasmuch as the air is admitted to the coolest part of the cylinder, insuring that a greater weight of fresh air will be trapped on the compression stroke.

The Sun Shipbuilding Company offers Doxford opposed-piston Diesel engines up to 10,000 hp per unit for marine and stationary service. A 3000-bhp unit at 85 rpm consists of 4 cylinders (8 pistons) with $23\frac{1}{2}$-in. diameter, total stroke 91 in. (crank-throw $22\frac{3}{4}$-in.), piston clearance at dead center 9.44 in., and firing order 1–3–2–4.

The opposed-piston design has also been applied to high-speed aircraft Diesel engines, as in the Junkers (German) "Jumo 4," which delivers 725 hp at 1716 rpm with a four-cylinder engine weighing 2.45 lb per hp and a fuel consumption of 0.37 lb per hp-hr. In this compact design, the pistons transfer the gas pressures through connecting rods to two crankshafts, one along the top and one along the bottom of the engine; a train of five spur gears connects these crankshafts with each other and to the airplane propeller shaft.

57. Two-stroke Cycle versus Four-stroke Cycle Diesel Engines. The number of power impulses given to the crankshaft per revolution is equal to the number of cylinders in a two-cycle engine, thus giving a more even turning effort than would a four-cycle engine. In the four-cycle engine the number of power impulses given to the crankshaft per revolution is equal to one-half the number of cylinders. This even torque of the two-stroke-cycle type makes the engine very steady in operation, as is illustrated by a six-cylinder engine (without flywheel) in regular operation producing alternating current in parallel with other prime movers. This regularity of turning effort has particular advantages in a marine installation, in that it permits a reduction in the size of the line, thrust, and propeller shafts.

A four-cycle engine requires the provision of an inlet valve and an exhaust valve for each cylinder, cam-opened and spring-closed, with a camshaft to carry the cams and gears to drive the camshaft from the crankshaft. It is evident that, with the four-cycle method of charging, there is required a certain amount of mechanism which needs considerable care and attention, involving gears, cams, rollers, and rockers; these parts are multiplied by the number of cylinders, the engines being usually multi-cylinder. The lower manufacturing costs of the two-cycle engine are offset by the higher charging efficiency of the four-cycle unit. By higher efficiency is meant that the cylinder can be fully charged or more nearly fully charged with air, with less expenditure of power in so doing, than with the two-cycle method.

The two-cycle Diesel engine requires a large air pump to furnish the scavenging and cylinder-charging air. This pump requires considerable power to operate it and reduces the mechanical efficiency of the unit. The larger amount of fuel consumed in the two-cycle cylinder —about twice that used in a four-cycle cylinder of equal bore and stroke —causes the heat stresses to be high. With correct cylinder head and liner designs, as shown in Fig. 74, this trouble has been minimized.

For large units the two-cycle engine, by reason of possible lower costs and less floor space, will probably be in greater demand. In a smaller type engine (less than 500-horsepower), it is considered doubtful whether the two-cycle engine can be made more cheaply than a four-cycle engine of equal capacity. Small cylinders for the two-cycle engine present difficult air-charging and scavenging problems. There is no doubt but that for the same cost of building, the two-cycle engine can be equipped with better material and with parts of better design than can the four-cycle type. For this reason the two-cycle principle will probably be adopted more generally in smaller units than at present, but in every case the engine must be better built than a four-cycle engine of like capacity.

58. Diesel Power-plant Trends. Many of the Diesel engines now produced in the United States are being built under foreign patents, although American ideas of simplicity and accessibility are constantly modifying the designs. The greater use of large Diesel engines is one of the trends that is noticeable in this field; but America, because of conditions more favorable to the use of coal and natural gas, lags behind European practice in its adoption of the larger Diesel engine sizes. The larger sizes are being used principally in marine service, where Diesel engines have almost completely displaced steam equipment.

The leading characteristics of representative Diesel engines recently manufactured in the United States are listed in Table XVI. Only one engine listed employs double-acting cylinders, and this is the largest size in the table. Larger capacity per unit has been sacrificed by the other builders in favor of the greater simplicity of construction of single-acting engines, due to freedom from stuffing boxes and crossheads, and to greater simplicity of piston cooling and of valve-operating mechanism. Choice between four-cycle and two-cycle engines appears to favor the two-cycle type for capacities exceeding 1000 bhp; from 450 to 1000 bhp recent four-cycle and two-cycle engines are about equal in number; and below 450 bhp most engines use the four-cycle principle.

According to Table XVI, more than 75 per cent of the engines listed employ solid (or airless) fuel injection; this condition is due in part to the greater simplicity of this form of fuel injection. It will be observed that the air-injection engines are generally of the four-cycle type; whereas the two-cycle types usually employ solid injection of fuel. An air-injection engine will burn a heavier and dirtier fuel—such as oils with a high asphaltic content—than one of the solid-injection type, and will also negotiate overloads better; but it has the disadvantage of having an additional compressor, which consumes some of the power developed by the engine. Fuel injection with high-pressure natural gas instead of air injection is being successfully used.

Fuel-injection pressures are increasing, and manufacturers employ pressures up to 10,000 lb per sq in. as the result of a desire to perfect the combustible mixture in the combustion space by means of turbulence. These higher pressures also effect cleaner spray nozzle action and reduce nozzle troubles. Piston speeds of the engines listed average nearly 1000 ft per min, regardless of size, type, or rotative speed. Higher piston speeds influence the reduction in weight per horsepower; reduction in weight is desirable and the trend toward lighter weights of Diesel engines

TABLE XVI

DOMINANT CHARACTERISTICS OF RECENT HEAVY-DUTY DIESEL ENGINES *

Location	Service	Brake Horse-power	Rpm	Cylinder		Cycle	Type of Fuel Injection	Pound of Fuel per bhp-hr	Bhp per Cylinder	Piston Speed, ft per min
				No.	Size					
Vernon, Calif.	Generator	7000	167	8	24 × 36†	2	Solid	0.385	875	1002
Tucson, Ariz.	Generator	4100	124	6	30 × 42	2	Solid	0.416	683	868
Rockville Center, N. Y.	Alternator	2865	120	8	29 × 48	4	Air	0.41	358	960
Grand Haven, Mich.	Generator	2250	225	6	21 × 29	2	Solid	0.40	375	1090
Berlin, Md.	Generator	1666	200	8	17 × 27	2	Air	0.43	208	900
Great Bend, Kan.	Generator	1400	300	8	16 × 20	2	Solid	0.39	175	1000
Bramley, Calif.	Generator	1125	240	3	21½ × 27½	2	Solid	0.39	375	1100
Bryan, Texas	Generator	1000	257	8	16¼ × 24½	4	Air	0.41	125	1050
LaCrosse, Wis.	Pump	1000	300	8	16¼ × 21	4	Solid	0.38	125	1050
Wahoo, Nebr.	Generator	750	257	6	16¼ × 24½	4	Solid	0.38	125	1050
Fernandina, Fla.	Generator	600	225	4	16¼ × 23	4	Air	0.42	150	863
Lake Mills, Wis.	Generator	600	400	6	12 × 15	2	Solid	0.40	100	1000
Volmeyer, Ill.	Generator	560	400	8	12½ × 15	4	Solid	0.40	70	1000
Farmer City, Ill.	Generator	450	300	6	14 × 17	2	Solid	0.38	75	850
Walters, Okla.	Generator	400	225	4	16¼ × 23	4	Air	0.42	100	863
Houston, Texas	Pump	340	225	3	17 × 24	2	Air	0.41	113	900
Girard, Pa.	Generator	300	300	4	14 × 17	2	Solid	0.38	75	850
St. Anne, Ill.	Generator	300	400	8	9½ × 14	4	Solid	0.40	38	933
Waimea, Hawaii	Alternator	275	300	6	11½ × 15	4	Solid	0.42	46	750
Elizabeth City, N. C.	Compressor	240	277	4	13 × 16	4	Solid	0.42	60	739
Holton, Kan.	Generator	225	300	3	13¾ × 18	4	Solid	0.40	75	900
Rochester, N. Y.	Generator	187½	400	5	9½ × 14	4	Solid	0.40	38	933
Connersville, Ind.	Generator	165	300	4	11½ × 14	4	Solid	0.42	41	700
Arp, Texas	Cotton gin	150	250	2	13 × 18	2	Solid	0.40	75	750
Carthage, N. C.	Generator	150	360	6	8½ × 12	4	Solid	0.45	25	720

* Abstracted from *Power*: pages 322 and 323, June, 1936; pages 333 and 334, June, 1937.

† Double-acting.

is evident. Weights per horsepower range from 25 to 300 lb, with an average of 100 lb.

Design trends are apparently toward higher speeds, weight reduction, greater unit output, supercharging, greater use of double-acting cylinders, more compactness, greater use of small engines, convertible Diesel-gas units, incorporation of accessories in the engine enclosure, frames and other parts welded, more complete enclosure, simplified control, higher mean effective pressures, increased use of normally wasted heat, and combination of Diesel engines with existing steam plants to improve heat balance—all resulting from various engineering improvements. Better design and workmanship and better auxiliaries are reducing labor and maintenance costs, increasing reliability. Some authorities doom the four-cycle engine, because the two-cycle has no valves, has smaller cylinders, requires less floor space, has lower first cost, weighs less, and needs less maintenance. Others think it is not doomed, claiming two-cycle units present greater cooling problems, are harder to lubricate, have lower mean effective pressures, require excellent scavenging equipment, and frequently require supercharging.

There seems to be a trend toward solid-injection Diesel engines in all sizes, as shown by Table XVI, but a number of well-known manufacturers still build air-injection engines and will probably continue to do so as long as Diesel engines are built. Solid-injection engines cost less, that is, the first cost, but whether they are cheaper to operate than air-injection engines depends entirely upon conditions. At certain places, where heavy low-grade fuels are very cheap, an air-injection engine would certainly be preferable. There are, no doubt, some locations where the comparative costs of fuel for air injection and for solid injection would be such that solid-injection engines would be a better choice; however, no definite selection should be made without consideration of all the factors involved.

The four-cycle Diesel engine still maintains a strong position in units above 1000 horsepower, and practically dominates the field below 1000 horsepower. It is still limited to less than 500 horsepower per cylinder, but the number of cylinders has risen to 8 in-line. The single-acting engine is practically limited to 800 horsepower per cylinder. The double-acting two-cycle engine dominates the field for everything above 4000 horsepower, and it may even eliminate the single-acting two-cycle types above 2000 horsepower. In the United States, owing chiefly to the greater activity of manufacturers, two-cycle engines hold a relatively more important position in the 1000-horsepower, and below, class than they do in Europe.

Many authorities see the greatest future of the Diesel engine in motive power such as long-haul truck and bus service, light-weight high-speed trains, and wherever its compactness and efficiency are effective. In marine work, the Diesel engine means low-cost operation and no fire hazard; in airplanes, dependability and reduced fuel cost, and longer cruising range. But the automobile presents a different problem. Car owners today pay for easy starting, quick acceleration, surging speed—not fuel economy. Diesel engines have not proved as satisfactory for these desired characteristics, but medium-compression spark-ignition oil engines approximating the Diesel principle may be used in the near future and make possible the wider adoption of Diesel engines for automobiles.

Technically, the Diesel engine for central-station service is equal to the steam plant in units up to 7000 horsepower, and is now proving itself in units up to 25,000 horsepower. Several plants of 15,000 to 50,000-horsepower are in successful operation, and plants up to 100,000 horsepower are now practicable. In any case, we shall need more power. It is logical to believe that Diesel engines will supply much of it, particularly as design and reliability are bettered, cost is reduced, and we as a nation better understand these engines.

CHAPTER IX

HIGH-SPEED DIESEL ENGINES

During the early development of the Diesel engine, the trend was toward increasingly larger units. The large Diesel having attained success, there is now apparent a definite trend toward commercialization of the smaller sizes. Much has been done along this line in recent years and manufacturers of both Diesel and gasoline engines are making efforts to produce Diesel engines that will compare favorably with gasoline engines in power, speed, size, weight, ease of starting, performance, and reliability.

59. Development and Application. In 1920, with a few marine exceptions, only stationary and heavy-duty internal-combustion engines were Diesels; and the applications requiring high speeds (exceeding 500 rpm) and light weight (50 lb or less) per horsepower were left entirely to gasoline engines. Since then commercially successful Diesel engines have been substituted for the usual heavy-duty gasoline engines. High-speed solid-injection Diesel engines are being used in ever-increasing numbers with such installations as ships and boats; aircraft; rail cars; locomotives; tractors; generator, compressor, pumping, and oil-drilling sets; excavators; hoists; draglines; shovels; dredges; irrigation; cotton gins; stone crushers; etc.

It was long believed that high speeds (1000 rpm and more) were not possible for Diesel engines. Even after the mechanics of high-speed designs became fairly well understood and improved materials became available, it was held that the combustion processes of the Diesel engine were too slow to warrant serious consideration. Subsequent research work has shown that these assumptions were false. Ricardo has demonstrated that combustion in an oil engine can be speeded up to a higher rate than that normally obtainable in a gasoline engine. Efficient combustion has been attained in engines operating at speeds in excess of 2000 rpm; hence, in recent years, commercially successful Diesel engines have appeared with characteristics which have made it possible to substitute them for heavy-duty gasoline engines. The field of the light-duty gasoline engines has not yet been reached commercially by Diesels, but they are moving toward lighter-duty applications.

Diesel engine manufacturers have steadily improved engine designs, materials, and manufacturing methods. The improvements in combustion systems have resulted in higher brake mean effective pressures (about 100 lb per sq in.), together with complete combustion and good efficiency. Better design of cylinder heads, use of improved materials, and welding of engine frames have strikingly increased the sturdiness of the high-speed engine as a unit. Improved crankshaft materials, stiffer shafts, and more effectual understanding of critical-speed phenomena have greatly reduced crankshaft breakages. There have been notable developments in the direction of higher and higher speeds, producing engines of smaller weight, space, and cost. These developments, in turn, have resulted in smaller foundations, lighter reciprocating parts, less vibration, easier handling of parts, and electric starting of the engine.

Modern developments in high-speed Diesel engines have tended to depart considerably from the slow-speed type and to follow more closely the gasoline engine, in both mechanical design and the thermodynamic cycle on which these engines operate. The constant-pressure combustion cycle, not strictly adhered to in the slow-speed engine, is seldom approximated in the high-speed engine, and in its place is substituted a cycle resembling more nearly the constant-volume cycle of the gasoline engine. The true constant-pressure cycle can be employed only in relatively slow-speed engines, and then only when the fuel is injected by means of high-pressure air, since this method alone combines the requisite degree of atomization, penetration, and turbulence. The air-injection constant-pressure cycle is not a desirable cycle for small high-speed engines, because its over-all efficiency is relatively low, and its principal advantage of a lower maximum pressure is of little benefit when applied to a small piston.

The low fuel cost, coupled with the high thermal efficiency obtainable regardless of load, renders a high-speed Diesel engine very attractive for use where fuel cost is a major item and where the mess and smell, frequently inseparable from heavy petroleum products, are not a serious nuisance. In certain applications, also, as in marine and aircraft, the great reduction in fire hazard resulting from the use of a fuel of high flash point is a very important consideration in favor of the Diesel engine.

60. Four-cycle High-speed Diesel Engine. Not many American manufacturers are today building four-cycle direct-injection full-Diesel engines of light weight and high speed; most four-cycle high-speed Diesel engines are being built with auxiliary combustion chambers. Winton four-cycle engines, Fig. 81, are built in two, three, four, six, and eight cylinders per unit in-line and up to sixteen cylinders in V-type, ranging from 40 to 1200 hp. The construction varies according to the size of the engine and the materials used. With semi-steel casting construction the bed plate is in one piece and carries the main bearings. In the small sizes the crankcase and cylinder block are in one piece; for the larger engines they are separate castings. All the larger sizes have long through bolts, see Fig. 81, for taking the cylinder loads and for holding the entire structure together. To save weight in the smaller sizes, the bed plate and crankcase are made of aluminum alloy. However, the cylinder block is always a close-grained iron.

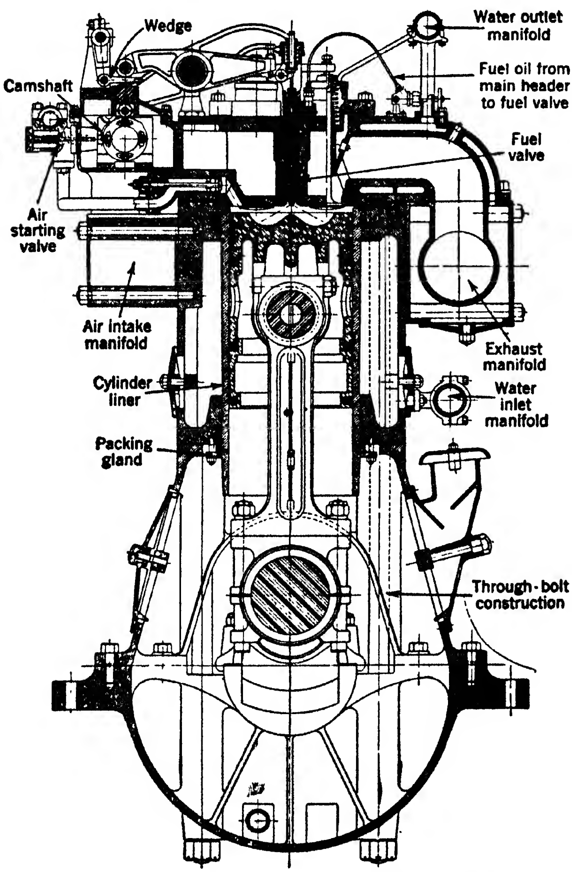


Fig. 81. Winton four-stroke cycle high-speed Diesel engine.

The V-type engine using lightweight construction has an iron cylinder block and an aluminum-alloy crankcase; the latter is in one piece, cast integral with the bed plate. Welded-steel construction is used in some cases; here the bed plate, crankcase, and cylinder block are all in one piece. Through bolts extend the height of the engine and take the cylinder loads. Separate cylinder liners are used for all engines. Removable cylinder heads, with central fuel-injection nozzle, are used exclusively; these heads are cast individually in large sizes and in groups for the smaller cylinders. The main and crankshaft bearings are either bronze- or steel-backed and babbitt-lined. The chrome-nickel wrist pins are clamped in the connecting rods and the bearings are in the piston bosses. The pistons are either cast iron or aluminum alloy, depending on the conditions of service.

When cylinders of $8\frac{1}{2}$-in. diameter and smaller are used, the valves seat directly in the chrome-nickel iron cylinder heads; valve cages are used in the larger sizes. Cylinders of 11-in. diameter and larger employ two inlet and two exhaust valves. When the piston stroke is 14 in. or less the camshafts are in the crankcase and operate the valves through pushrods and rockers. All larger engines, also V-type and welded-steel-frame engines, have the camshafts at the cylinder-head level, see Fig. 81, and drive the valves directly by rockers. The camshaft is driven by a set of spiral gears from a vertical shaft, which in turn is driven by the crankshaft. All these engines have the valves and valve gear fully enclosed. The six- and eight-cylinder marine engines are made directly reversible by sliding the camshaft.

The constant-pressure or common-rail type of fuel injection, as explained in Art. 76, is used on all Winton four-cycle Diesel engines. The injection-valve wedge control mechanism is mounted on the camshaft box. This mechanism consists of a roller plug and wedge, see Fig. 81, at each cylinder for controlling the lift of the fuel valve and, consequently, the amount of fuel delivered into the cylinder at each firing stroke. The roller plug guide carries a hardened steel-roller plug and hardened steel roller and pin. The roller rides on the injection cam and the top of the roller plug is flat-pointed where it comes in contact with the hardened steel wedge which increases or decreases the duration and lift of the fuel-injection valve, by the governor-controlled rotating movements of the wedge lever shaft running the full length of the engine.

Pressure lubrication is used; for some engines the bed plate is used as a sump and the pump delivers the oil directly to the bearings; for other engines the double pump and reservoir system is used. Suitable strainers or filters and oil coolers are included, with all the models. The smaller sizes employ electric motors for starting and charging generators. The large sizes have a compressed-air starting system, which consists of a throttle valve, air-distributor valves for each cylinder, and the air-starter check valves mounted on the side of the cylinder heads; see Fig. 81. The air-distributor valves, mounted on the camshaft housing, are operated directly by a cam on the camshaft. Starting air is admitted to these valves from the air-starting bottles by the hand-throttle valve.

61. Four-cycle High-speed Diesel Engines with Auxiliary Combustion Chambers. The Comet type of Diesel engine, as built by the Waukesha Company under Ricardo patents, is of the four-cycle mechanical-injection type with a special form of combustion chamber; see Fig. 82. It is called the Comet because the spherical chamber resembles a comet's head and the tangential outlet the tail. Practically the entire charge of air is compressed into this small chamber, which communicates with the cylinder proper by a venturi-shaped passage. The lower half of the combustion chamber is a single-piece (separate) casting which is inserted in a recess cast in the cylinder head. A clearance space of

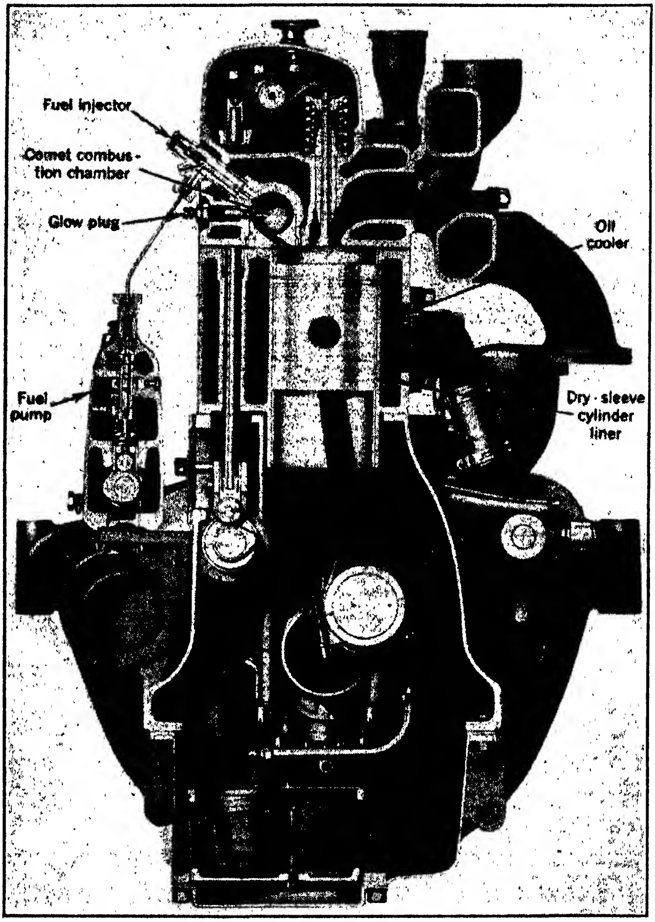


Fig. 82. Waukesha-Comet high-speed Diesel engine; section through cylinder, valve mechanism, spherical combustion chamber, Bosch fuel pump and injector, and glow plug for starting.

0.025 to 0.035 in. is provided around its circumference to prevent dissipation of the heat in the combustion chamber. This insulation of the combustion chamber, together with its shape, results in a reduction of the ignition delay period so that these engines operate smoothly, start readily, and require no advance or retarding of the injection timing.

The fuel is burned by forcing it into the spherical combustion chamber at an injection pressure of about 2000 lb per sq in., as in the regulation high-pressure Diesel engine. By its special design, a large portion of the heat of combustion is held in the walls of the spherical combustion chamber, to be imparted to each fresh charge of air, and added to the normal heat of compression. This effectively reduces the ignition lag by starting the combustion immediately after the beginning of fuel injection, and burning the charge smoothly and progressively. A pintle-type Bosch injection nozzle, Fig. 105, is used. It is free from the usual difficulties experienced with nozzles having several small orifices, and is practically self-cleaning. For cold-weather starting, a glow plug operating on two volts is connected with the starting switch so that, whenever the electric starter is in operation, the glow plug will switch on for a moment until the engine starts.

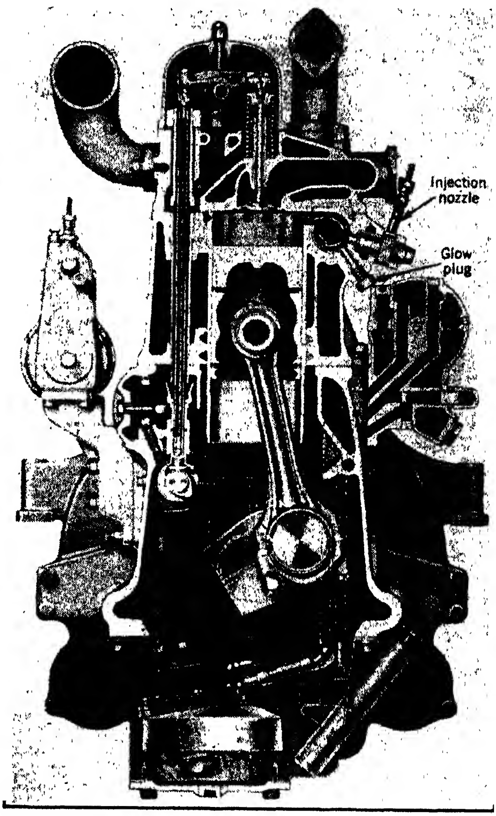


FIG. 83. Section through Hercules automotive type Diesel engine.

The fuel pump is a single, self-contained Bosch unit, Fig. 112, with individual cylinder and plunger for each engine cylinder, and a metering mechanism to control the quantity of fuel injected at each stroke. Engine speeds are varied by the centrifugal governor and throttle connection to the fuel pump. These engines are built in six-cylinder units of 100-, 125-, and 140-horsepower capacities at 2200 rpm and are especially adaptable to heavy-duty automotive service, as for tractors, trucks, or buses. Waukesha-Comet engines have seven-bearing crankshafts, dry-sleeve and non-rocking truncated cylinders, five-ring aluminum alloy pistons, and full-floating piston pins. All standard accessories commonly used in automotive work can be applied.

Figure 83 is a sectional view of the Hercules high-speed Diesel engine which has a spherical combustion chamber located to the side of the top of the cylinder. The piston, at the top of its stroke, comes very close to the cylinder head, with the result that the compressed air is projected with much turbulence into the combustion chamber, along the under side of the cylinder head, as indicated by the arrows in Fig. 83. The injection nozzle sprays the fuel across the main path of this turbulent air stream. On the upstroke of the piston, the velocity of the air into the combustion chamber decreases from minus 60 degrees (crank angle) until the piston begins to cover the passageway, at which time there still remains, in the space over the piston, by far the greater percentage of the combustion air. As the piston closes the passageway into the combustion chamber, thus reducing its area, the velocity of air increases rapidly to a maximum at the time fuel injection occurs. The inertia of the air, passing from the cylinder to the combustion chamber, probably empties the cylinder quite completely into the combustion chamber, where combustion occurs at a rate controlled by the speed of the engine. A 130-hp six-cylinder 5 × 6 in. Hercules Diesel engine, with a compression ratio of 15 to 1, operating at 1300 rpm gave a fuel consumption of 0.40 lb per bhp per hr and 112 lb per sq in. brake mep.

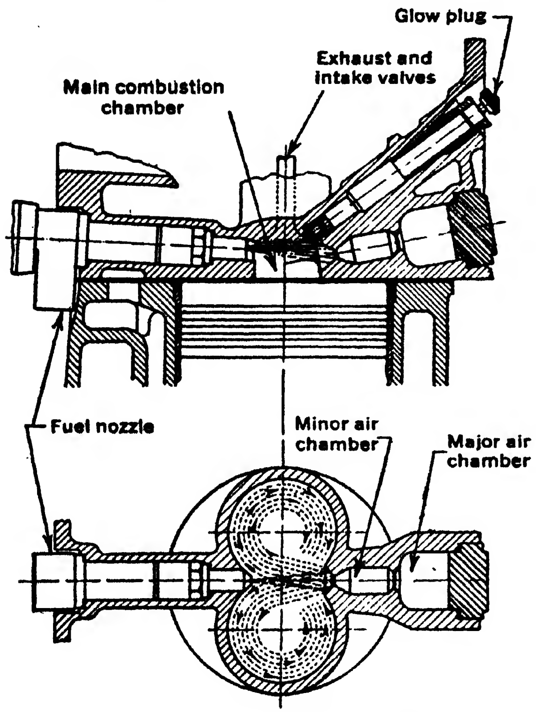


FIG. 84. Buda-Lanova high-speed Diesel engine; above, part sectional elevation of combustion head; below, sectional plan showing the double annular combustion space.

In the Buda-Lanova engine, the combustion chamber is in the shape of a double annular space, each half being symmetrical, Fig. 84. The Bosch fuel-injection nozzle is on the left, whereas on the right a double air chamber is arranged; this consists of a small chamber communicating with a larger one as shown. When the piston is almost at the top of its compression stroke the fuel stream is injected across the space between the two cylinder-head recesses, towards the entrance to the minor air chamber. By the time the spray reaches this entrance it has become heated so that ignition begins before the fuel enters the air chamber. As the piston descends, the pressure in the combustion chamber will be lower than in the air chamber, with the result that the air rushes out in the opposite direction to that of the fuel spray, thus atomizing the spray and carrying it around in the two annular combustion spaces, and forming two turbulent mixture streams, as indicated in Fig. 84.

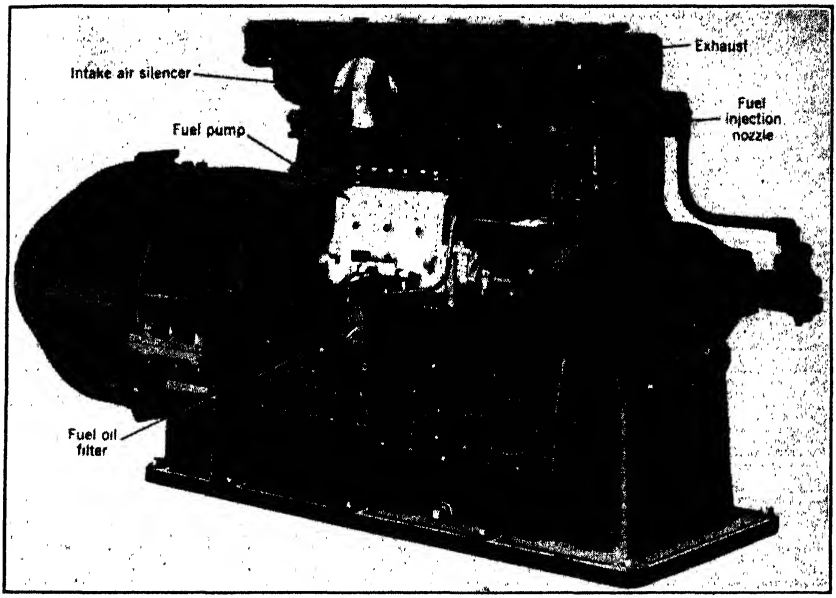


FIG. 85. Buda-Lanova high-speed Diesel engine direct connected to an electric generator.

This self-induced turbulence gives accelerated burning and progressive rise in pressure by accelerating the end of the combustion cycle instead of the beginning; thus it is possible more nearly to approach the character of gasoline combustion. The rate of pressure rise at the beginning of ignition is relatively small, since only a portion of the air is available for combustion; indicator diagrams show that the maximum cylinder pressures attained are 550 to 625 lb per sq in. The maximum compression pressure employed is about 400 lb, the highest mean effective pressures (based on brake horsepower) are 115 to 125 lb, and the injection pressure is about 1200 lb per sq. in. Glow plugs are provided, as shown in Fig. 84, for easy starting in cold weather.

As shown in Fig. 85, modern high-speed Diesel engines have a remarkably clean-cut exterior and are free from complications. Their general lines of construction adhere closely to well-proved gasoline-engine design.

An unusual combination of fuel valve and combustion chamber, used in Cummins high-speed Diesel engines, is shown in Figs. 86 and 87. The fuel is delivered by a low-pressure (about 100 lb per sq in.) fuel pump through ball check valve *V*, during the engine suction stroke,

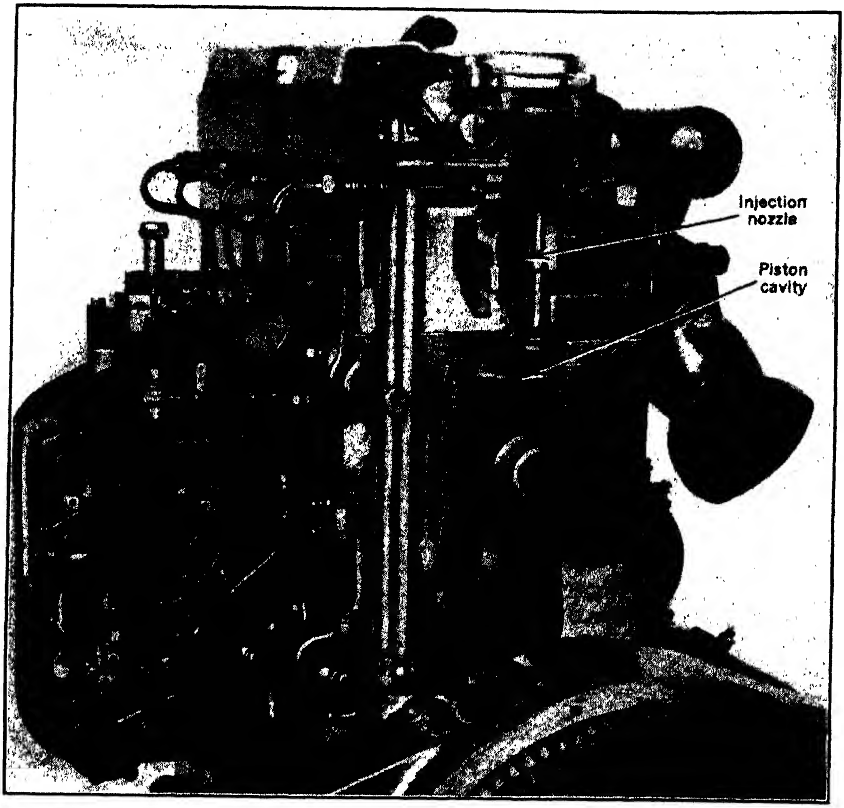


Fig. 86. Cutaway view of Cummins automotive type Diesel engine.

into annular space *S*. During the latter part of the suction stroke and during the compression stroke, plunger *P* moves upward, creating a vacuum which prevents the fuel from entering the cylinder. At the instant when injection is desired, plunger *P* is forced down rapidly as shown in Fig. 87, and injects the fuel, which by this time has been preheated and partially vaporized. The cavity *C* in the top of the piston is filled with compressed air when the piston is at the end of the compression stoke; as the piston moves downward and the cylinder pressure decreases, the air flows from the piston cavity *C* into the cylinder, creates turbulence which assists in obtaining complete combustion, breaks up the rich fuel zone around the end of the injection nozzle, and prevents the formation of carbon cones on the nozzle tip. Cummins engines are of the compression-ignition type, in which a single-plunger pump is used for delivering fuel to all the cylinders. The Cummins fuel-injection system is shown in Fig. 102 and explained on accompanying pages.

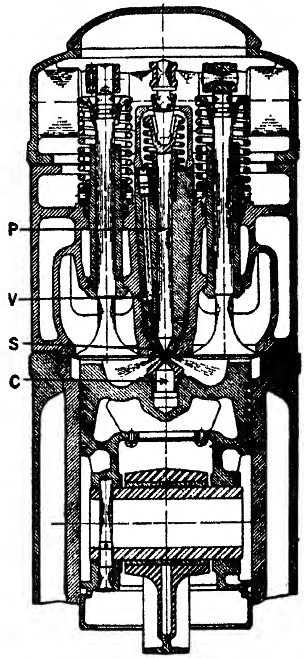


FIG. 87. Section through Cummins combustion chamber, fuel injection nozzle, and valves.

62. Two-cycle High-speed Diesel Engines. The two-cycle Diesel engine has achieved considerable success in the large heavy-duty slow-speed engines. Its inherent advantage of simplicity and greater power output per unit weight make it very desirable for the smaller high-speed engine designs. The two-cycle engine will give a much better torque than the four-cycle type under similar conditions, and in certain cylinder arrangements a smaller number of cylinders can be used to give as good or better results than those obtained with the equivalent four-cycle multi-cylinder engine. The application of the two-cycle principle to small high-speed engines involves many difficulties, which are by no means insurmountable. These difficulties include high-speed fuel injection at twice the speed of the four-cycle type, the proper elimination of the exhaust gases, the prevention of loss of volumetric efficiency due to the charge of air escaping through the exhaust port, the provision of a suitable blower, and the prevention of excessive fuel consumption.

With the piston-controlled cylinder-port type of engine, where the inlet and exhaust ports are uncovered at the same time, there is always the tendency for part of the air charge to sweep into the exhaust passage, thus causing a loss in volumetric efficiency. Although a larger air charge may be arranged for intentionally in order to obtain complete scavenging, it must not be overlooked that this beneficial effect must be offset to some extent by the increased power required by the air-pumping system. One of the important problems of two-cycle engines is introducing the air charge with the minimum decrease of volumetric efficiency. In this case the cylinder must be filled in a fraction of the time of a four-cycle engine running at the same speed; also, there is no exhaust stroke to clear the cylinder of most of its burnt gases as in the four-cycle engine.

The simplest method of introducing the charge of air into the cylinder of a two-cycle engine is to use crankcase compression, as shown in Fig. 29. This method is inefficient because the pumping action is not correctly phased with the inlet port opening; the inlet and exhaust ports reduce the effective part of the cylinder charging stroke; the crankcase (pump chamber) has a large unswept area; and the lubricating oil vapor mixes with the air charge in the crankcase. The amount of air charge transferred by the crankcase compression method is only about 40 per cent of the cylinder volume; thus the power output of the engine is limited. For these reasons crankcase compression is not employed in modern high-speed Diesel engines; an integrally built blower is generally used for air scavenging and charging the engine cylinder.

At the end of the expansion stroke of a two-cycle engine, the cylinder is filled with burnt gases. The problem confronting the designer is to clear these gases away in the very short time left between the end of the expansion stroke and the beginning of the air-charge injection. The most satisfactory method is air scavenging, a surplus of air being required for this purpose; in effect, this approximates the use of a form of supercharging device. Various types of rotary, centrifugal, and gear or exhaust-turbine driven compressors have been used for supercharging in an effort to obtain a light, efficient, and compact form of air blower or compressor that will provide a large volume of air at a low pressure. A 100-hp two-cycle Diesel engine requires a scavenging air pressure of about 5 lb per sq in. gage for engine speeds of 1200 to 1500 rpm in order to be relatively efficient and economical. The capacity of the air blower is generally expressed in terms of swept cylinder volume and is dependent upon the port design, speed, type of blower, and its location. The usual practice is to arrange for the blower to deliver twice the cylinder volume at each charging stroke. The power required to supply air to the cylinders under these conditions, allowing for a certain amount of wastage through the exhaust ports and the usual blower losses, is about 12 to 18 per cent of the power developed by the engine.

In order to insure efficient scavenging, it is necessary to direct the flow of the air as it enters the cylinder. It is not merely sufficient to place the inlet and exhaust ports on opposite sides of the cylinder; the use of inclined ports (or piston deflectors), as shown in Figs. 71, 74, 75 and 90, tends to give air-flow paths and if these paths can be obtained the top part of the cylinder can be scavenged and cooled. A certain amount of experimental work, involving the measurement or indication of the actual air flow, is necessary to obtain the desired flow direction and effectiveness when using different sizes, locations, and inclinations of inlet and exhaust ports.

Public interest has been aroused by the advent of high-speed lightweight trains, and the fact that Diesel engines were installed as the motive power has focused attention on the ultimate possibilities of this particular type of engine in the transportation field. These trains weigh less than half the ordinary steam locomotive and tender. This reduction in weight was made possible by the use of streamlining, Fig. 88, light alloy steels, electric welding, and the Diesel-electric drive. The forward

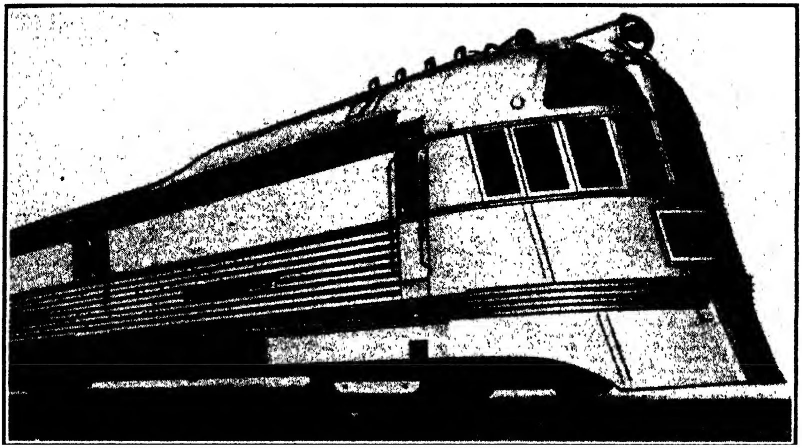


FIG. 88. Burlington Diesel-powered high-speed locomotive.

car of these trains includes the engine room (Fig. 89) for the Diesel-generator unit and its accessory equipment; air for cooling the jacket water enters through the grid openings at the front (top) of the car. The compactness of the two-cycle engine has made this type very desirable for railroad use, as more power can be developed for a given space and the engine weighs less per horsepower than an equivalent four-cycle engine.

Winton two-cycle Diesel engines with 8-in. × 10-in. cylinders are built in 8, 12, and 16 cylinders per unit, the two latter units being of the V-type; ratings range from about 600 to 1200 hp at 750 rpm. The uniflow principle of air scavenging is accomplished by air entering the lower part of the cylinder through piston-controlled ports and leaving through the four exhaust valves in the cylinder head. The inlet scavenging ports fill the entire periphery of the cylinder, thereby insuring ample air flow at 3 lb per sq in. pressure from a built-in rotary blower; left end in Fig. 89.

Five valve openings are provided in the cylinder head, where the four exhaust valves are placed symmetrically around the central fuel-injection nozzle. The fuel pump and injection nozzle are in a single assembly, bolted to the center of the cylinder head, and called the unit fuel injector, details of which are shown in Fig. 108. Cams on the overhead camshaft operate the two rockers for the four exhaust valves, whereas rotation of the fuel-pump plunger is effected by a rack connected to the governor lay rod. The fuel-pump plunger is directly above the injection valve and operates on the downward or discharge

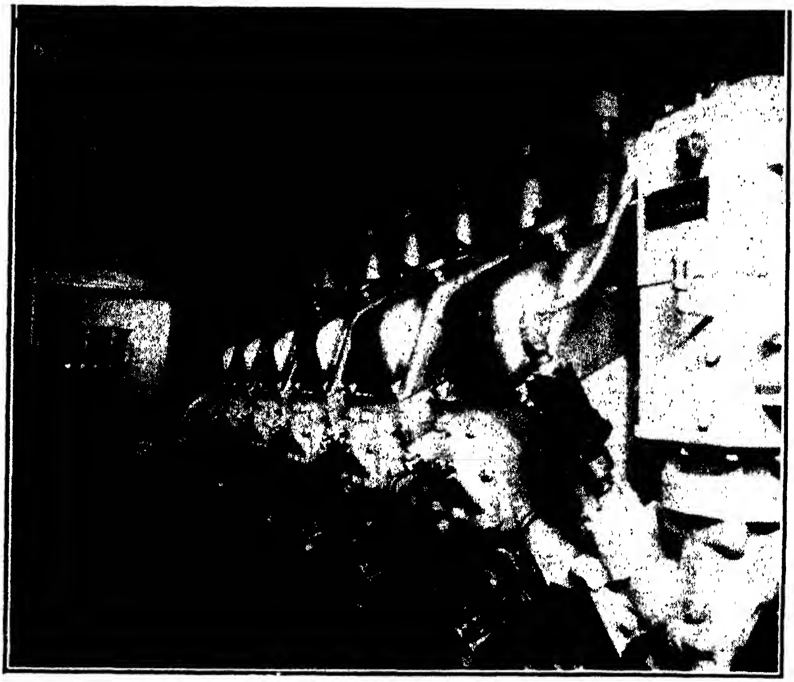


FIG. 89. Winton two-stroke cycle high-speed Diesel engine and radiator assembly installed in locomotive engine room.

stroke, building up pressure at the bottom of the pump which contains the spring-loaded needle valve. Fuel is supplied to the pumps at about 20 lb pressure and enters the plunger spaces through ports in the plunger bushing. Control of the amount of fuel injected is accomplished by rotating the fuel-pump plunger by the governor. The fuel is distributed in the combustion chamber by multi-hole nozzles.

Pistons are of aluminum alloy; and they, together with the wrist pin, connecting rod, and crankpin bearing, compose an assembly unit weighing less than 87 lb. The 19-in. rod has a drilled passage through which oil flows to the wrist-pin bearing and out through an opening in the top of the rod, to impinge upon the webbed lower surface of the piston crown; this gives piston cooling without complicated piping.

The working stresses are not carried by the frame, for eighteen tension rods connect the cylinder-head plate with the cross members supporting the lower bearing shells. In effect, the cylinders, heads, pistons, rods, and crankshaft compose a power machine about which the welded frame is placed on a non-stress jacket. Direct-connected to the Winton Diesel engine through a steel disk coupling is a single-bearing differential-wound generator. The front truck of the leading car carries two series-wound traction motors, which supply the motive power for the entire train.

The Busch-Sulzer two-cycle V-type engine of medium high speed, Fig. 90, has been developed for heavy switching, transfer, and main-line Diesel-electric locomotives. These engines, using 14-in. × 16-in. cylinders, are built with 8 to 16 cylinders and rated at 1600 to 3600 bhp at 550 to 600 rpm. The angle of the vee between the cylinder banks is 45 degrees. The engine can be built with a weight of 23 lb per bhp, but if a high starting tractive effort is required an engine weight of 36 lb per bhp is recommended. Scavenging air is supplied by a gear-driven Roots-type rotary positive-displacement blower mounted across the top of the vee between the two banks of cylinders. The blower housing acts as a cover and the enclosed vee is thus utilized as the receiver for the scavenging air.

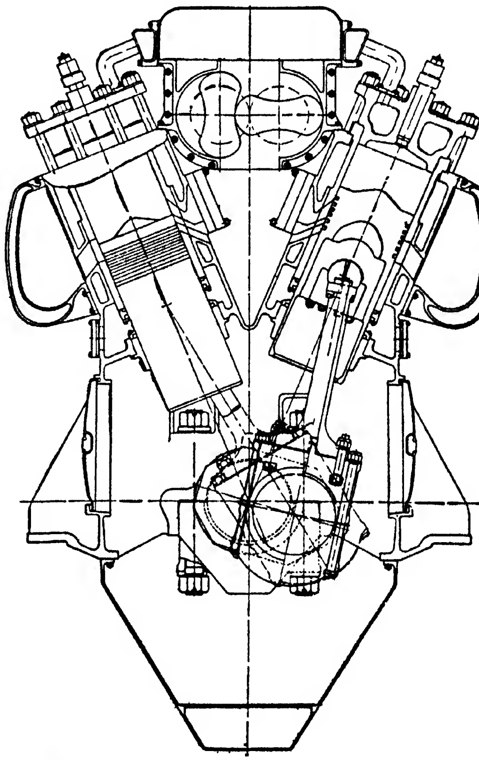


FIG. 90. Section of Busch-Sulzer two-stroke cycle V-type medium high-speed Diesel engine.

The scavenging and charging of the cylinders are effected with two tiers of ports around half the cylinder circumference adjacent to the scavenging air receiver. The upper tier, which is uncovered by the pistons before the exhaust ports are uncovered, is controlled by automatic valves, opening toward the interior of the cylinders. These valves, located in the scavenging air receiver, act as check valves and prevent a blow-back of exhaust gases into the receiver. The scavenging ports in the lower tier form a direct communication from the receiver into the cylinder and are uncovered by the piston after the exhaust ports have been uncovered, and when the pressure has fallen well below that of the scavenging air. On the upward stroke of the piston the scavenging ports in the upper tier remain open after the exhaust ports have been covered by the piston and the flow of air for charging continues until these ports have been covered. Therefore, the cylinders are filled with fresh air at a pressure above that of the back pressure of the exhaust, and a moderate degree of supercharging is effected. The exhaust ports are located around half the circumference of the cylinder, opposite the scavenging ports, and communicate directly from the interior of the cylinder into a water-cooled exhaust header which extends along the front of each bank of cylinders.

The engine main frame and cylinder block, of one-piece construction, Fig. 90, is cast from aluminum alloy or iron, depending upon weight limitations. The removable cylinder liners are made of alloyed cast iron. Each cylinder is provided with a water-cooled cylinder head; there are only two openings in each head, one for the fuel-injection nozzle and the other for a compression relief valve. The cylinder block and liners are so arranged that each cylinder has a sludge chamber near its lower end, also shown in Fig. 75 and explained in Art. 53. Pistons are made of cast aluminum. The wrist-pin bearing is provided in a separate housing which is inserted into the piston from below. The piston skirt is, therefore, not pierced by the wrist pin, resulting in a construction that permits full freedom for expansion. The piston becomes longer with the addition of the sludge chamber and accommodates a longer connecting rod of 5.25 times the crank-throw, as compared with 4.5 times for the conventional type of trunk-piston engines. The angularity forces are considerably reduced owing to this greater ratio; the increased area of the long piston and cylinder also reduces the wall pressures substantially with corresponding reduction in cylinder wear and tendency of the piston to "slap." The crankshaft is a single steel forging of heat-treated open-hearth steel. Each crankpin serves for two opposite cylinders, so that the number of cranks is half that of the cylinders. The engine uses the Hesselman system of direct fuel injection. Starting is effected by applying electric power from storage batteries to the generator attached to the engine crankshaft. During the starting period, the generator, therefore, acts as a starting motor.

63. Opposed-piston Two-cycle High-speed Diesel Engines. Eliminating crankshaft, connecting rods, cylinder heads, gaskets, valves, and camshafts, the Sterling crankless high-speed engine is a revolutionary trend in design. The compactness of the engine is shown by Fig. 91 and the principles of operation by Fig. 92. The engine has four horizontal cylinders, each containing two opposed reciprocating pistons. Thus the engine is equivalent to the usual two-cycle eight-cylinder Diesel engine of corresponding bore and stroke. The cylinders are arranged about a straight drive shaft which carries an inclined disk (also called wobble plate) at each end. The disks are virtually flywheels. The pistons act directly upon the circular disks set angularly upon the

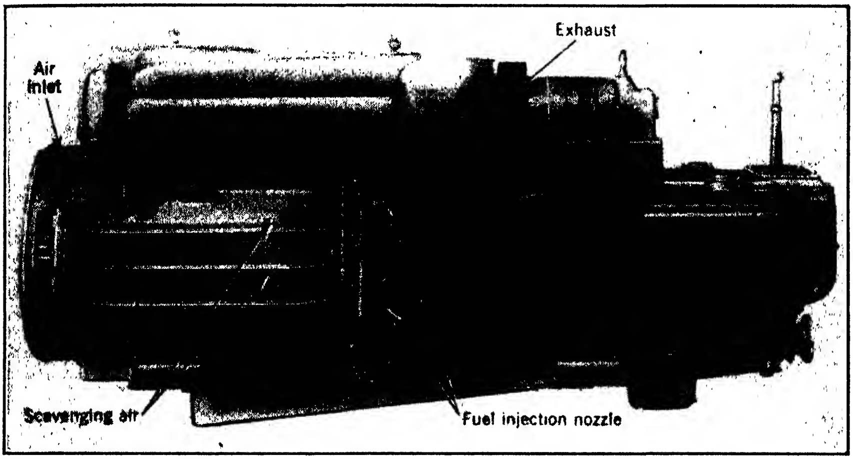


Fig. 91. Sterling crankless opposed-piston Diesel engine; two-stroke cycle, mechanical-injection; four-cylinder.

shaft, so that the gas pressure transmitted by the pistons forces the disks and shaft to rotate.

The mechanical bearing units which transfer the power from pistons to drive shafts utilize the same principle as the Michell and Kingsbury thrust bearings; that is, from the piston base, a U-shaped bridge passes over the rim of the disk and carries two bearing surfaces. The thrust surface is a slipper of rectangular shape with babbitt bearing surface, mounted on a half sphere; the other bearing (which serves only to position the piston) is a bronze-faced half sphere. This provides a universal mounting which permits the bridge to adapt itself to any motion of the inclined disk. The bridge is guided by short rods extending through guide holes, Fig. 92.

The engine follows conventional two-stroke cycle design in that circumferential ports replace valves. Scavenging air is provided by a pressure pump. Instead of the usual rotary or centrifugal blower, however, piston-type pumps are used, and the piston is mounted directly on the end of the piston bridge rod; left side in Fig. 92. Air is distributed by a simple rotary valve at the left end of the shaft.

There are separate rows of intake and exhaust ports at opposite ends of the cylinder, and extending completely around it. Thus one piston uncovers the exhaust ports, the other the intake ports. Exhaust ports are uncovered first. The charge of fresh air is given a spiral or swirling motion to clean out the combustion chamber. The air charge is compressed between the pistons, and the fuel oil is injected under a pressure of about 2500 lb per sq in. Slightly concave piston surfaces prevent spray impingment on top of the pistons. Air enters at one end of the cylinder and leaves at the other, providing uniflow scavenging and

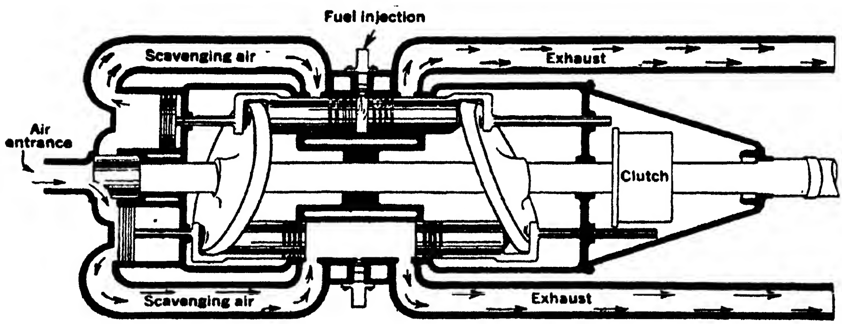


FIG. 92. Principles of operation, Sterling crankless opposed-piston Diesel engine.

charging. Out-of-phase pistons provide a longer combustion interval, at approximately constant pressure.

The high speed (1200 to 1800 rpm) of the engine and the two-stroke cycle combine to provide uniform average cylinder temperatures and greater combustion flexibility. The heat economy resulting from the proper combination of combustion space and port area, together with the absence of water-cooled cylinder heads, results in high cylinder capacity and excellent power output. Horizontal opposed movement of the pistons provides smoothness of operation, and the absence of a crankshaft avoids piston side thrust on the cylinder walls; this lessens piston and ring wear. Torsional wear is practically eliminated, as is direct structural loading of the frame. Electric starting is possible because of the light weight of the engine; it weighs but 13 to 20 lb per hp, depending upon size and service. Lubrication of inclined disks and piston shoes is accomplished by spraying oil through nozzles directly upon the disks. The pistons, being exposed for a considerable portion of their length to the oil thrown from the disks by centrifugal action, are satisfactorily lubricated and cooled.

The Hill Diesel engine embodies in its design a clever utilization of the advantages of the opposed-piston principle and elimination of certain disadvantages that have previously been considered inherent in this type of engine, the important features of which are shown in Fig. 93. The cylinder is cast integral with its water jacket, with scavenging chamber, exhaust passage, combustion chamber, and crankcase with bolted-on covers closing the bottom and ends.

An unusual departure from conventional practice is the placing of a rocker arm at each end of the crank, to the upper end of which is attached

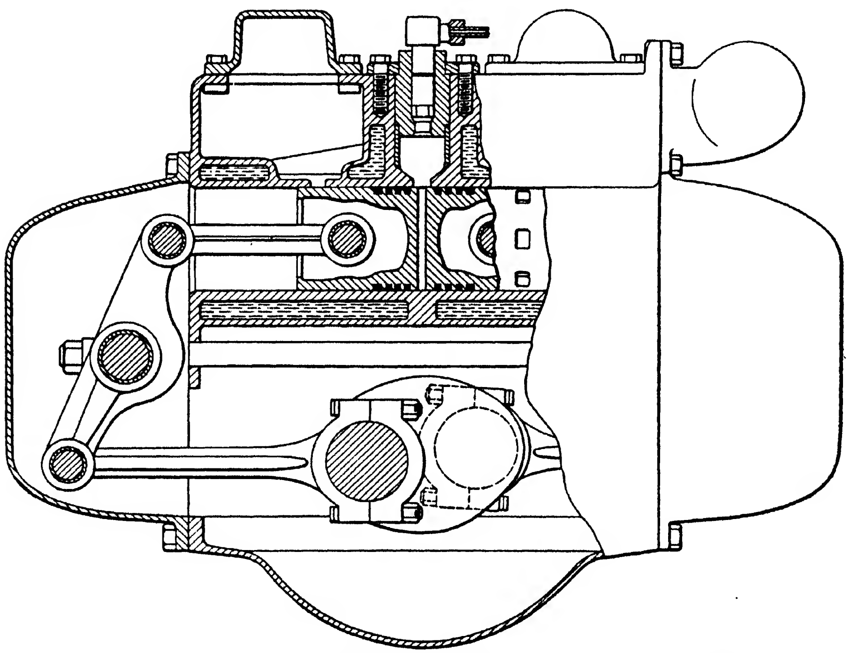


FIG. 93. Cutaway view of Hill opposed-piston two-stroke cycle Diesel engine.

the usual connecting rod, while at the lower end is attached another connecting rod which is joined to the crank. This rocker arm is offset to bring the upper end in line with the piston and the lower end in line with its crank. A steel tension rod, passing under the cylinder, ties together the two fulcrum bearings in which the rocker arms are mounted. This arrangement causes the pressure load from the two pistons to be exerted equally and in opposite directions on the two ends of the tie rod and concentrates all the load in the rod. By the disposition of many moving parts it may be seen that the two groups of masses move in opposite directions at the same rate of speed, resulting in good balance and minimum of noise.

The fuel-injection and combustion system consists of a single-orifice injection nozzle, mounted on top of the cylinder, discharging into a precombustion chamber, and a single Bosch pump mounted adjacent to the fuel nozzle. The extreme simplicity of the engine is an outstanding feature; it is stated by the manufacturer that about 250 fewer parts are used than in a conventional four-cycle engine of the same horsepower. No cylinder valves, camshaft, pushrods, or other valve mechanism are required, and there are no gaskets under pressure in the engine. Scavenging air is supplied by a blade-type eccentric-rotor blower, which is placed at one end of the engine frame and is driven by the crankshaft. The air is delivered into the passage above the left piston, see Fig. 93, and enters the cylinder when this piston uncovers the ports. Exhaust takes place through ports uncovered by the right piston, as shown in Fig. 93, and the gases flow out through a passage to the exhaust piping.

64. Convertible Diesel-gas Engines. The effect of the present economic trend upon oil and gas engine design is shown by the introduction of an increased number of engines capable of operating on either liquid or gaseous fuels. Simplicity, increased production of a minimum of parts, and interchangeability make such an engine desirable by increasing the range of application of the unit. With the growing use of natural gas and the extension of pipe lines from established or newly discovered gas fields, high-speed gasoline and Diesel engines, which found a good market in small plants, now have a more flexible competitor. Advantages of the multi-fuel engine are the elimination of fuel-oil storage and the use of gas as a reserve, much like purchased electricity. Also, the extension of refinery practice to gasoline reforming units and the desirability of conserving steam for process work have caused the utilization of by-product still gases as engine fuel.

Another reason for the convertible engine is the demand for it in the oil fields where, during certain stages of drilling, either gas or oil is not available. That the gas engine, converted to a Diesel, can be as efficient as a strictly individual design has been proved by experience and extensive tests. However, this is true only when the original design is developed with the various compromises necessary to insure correct combustion spaces for both types. The engine must be designed primarily as a Diesel, thus giving the frame and reciprocating parts of the gas and gasoline conversion sturdy construction. The combustion chamber and cylinder head for each engine type must provide the requisite turbulence, cooling, and valve-port construction.

A number of convertible engines are at present available in the high-speed type. The method of one builder for converting the gas engine to a Diesel requires changing cylinder heads, gas-mixing valve, and electrical ignition to a fuel-injection system. The valve area in the Diesel is the same as for the gas engine, permitting interchangeability of valves and valve springs. The average time required for making a complete change to operation on fuel oil is four hours.

Two types of convertible engines are shown in Fig. 94. The upper gas engine has one spark plug at the center of the combustion chamber. The cylinder head is flush and the piston is short. The Diesel cylinder

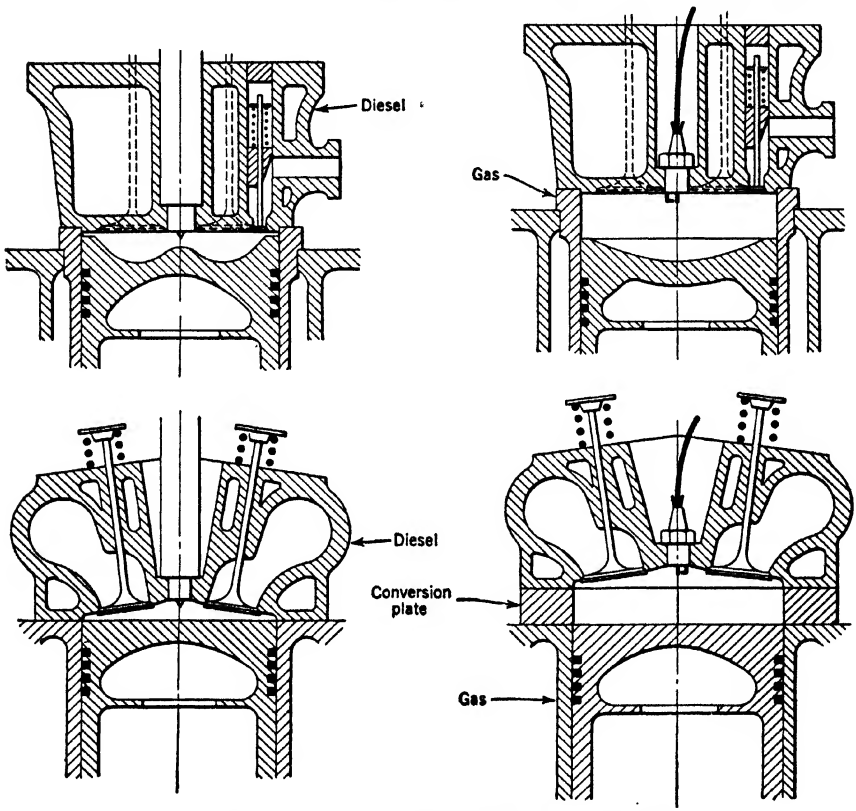


FIG. 94. Typical convertible Diesel-gas engines.

head is similar to that of the gas engine, but uses a spray nozzle in place of the spark plug. Starting of the upper engines is effected with compressed air; see air-starting valve at right side of cylinder heads. The Diesel piston of the upper design has a hemispherical top and is longer above the wrist pin, giving the higher compression of the Diesel cycle. The change in compression pressure is from 120 lb. per sq in. in the gas engine to 450 lb per sq in. in the Diesel engine. This engine is rated at 30 bhp per cylinder at 550 rpm. The gas-mixing valve is of the balanced type, with a single adjusting screw which is rotated to regulate the quantity of gas. Ignition is furnished by magneto or battery.

For the lower designs shown in Fig. 94, the same piston, cylinder, and cylinder head are used for the Diesel and the gas engine. A conversion plate is inserted between the cylinder block and the cylinder head, as indicated. This conversion plate is of the proper thickness to reduce the compression; and, by replacing the fuel pump with a magneto and the fuel nozzle with a spark plug, the conversion is easily made. The spark plug is centrally located and the valves are slightly inclined; for the Diesel engine the fuel nozzle must use a wide-angle spray tip.

The operating characteristics of the high-speed convertible Diesel-gas engine indicate that such designs are commercially practicable, and that economical operation is possible where application of the Otto- or Diesel-cycle conversions of the high-speed engine is desirable.

65. Trends of High-speed Diesel-engine Design and Application. Modern trends in high-speed Diesel-engine designs must be carefully interpreted because the technique is changing so rapidly that the separation of experimental design from practical design may be difficult. Definite trends are toward two cycles, increased rotative and piston speeds, less vibration and noise, weight reduction by use of welding and metal alloys, multiple inlet and exhaust valves, improved scavenging, standardized fuel oil, aluminum-alloy pistons, use of cylinder liners, improved combustion, and tie-rod framing construction.

Controversial factors in design are: supercharging to increase the output of the smaller cylinder and to avoid the effects of valve restriction; the use of aluminum in stressed members and the limiting size of aluminum-framed cylinders; the limit to the size of a cylinder using an auxiliary combustion chamber or air reservoir in the cylinder head or piston head; application of the V-type frame to Diesel design; opposed-piston arrangement; medium-compression spark-ignition pump-injection oil engines; individual fuel pumps, common-rail system, or unit injectors; closed automatic fuel nozzles preferred to mechanically operated or open fuel nozzles; correct limiting size of the four-cycle cylinder and the correct size at which to shift to the two-cycle design; uniflow path of scavenging air; and the use of the sleeve-valve Diesel engine.

Designs of lighter weight are demanded of the Diesel engineer, and the weight reductions must be accomplished without expensive refinements. Efforts to bring the structural weight of the high-speed Diesel engine down to gasoline-engine figures are handicapped by the higher maximum pressures that must be sustained by the cylinder heads, pistons, connecting rods, crankshaft, and frame. A total weight differential of about 20 per cent additional should be sufficient to withstand these higher pressures. Weight saving through the use of aluminum alloys or thin steel castings instead of nickel cast iron will be more generally practiced. Reducing the engine weight per horsepower by increasing the rotative speed involves problems of reducing and balancing inertia forces, bearing design, wear, cooling, and lubrication.

The limit to power increase through an increase of speed is the volumetric efficiency; the solution for this difficulty is supercharging, which is not much used as yet. Mean effective pressure rating is an arbitrary selection, and inherently different for Diesel engines of the same type. An average value of mep based on brake horsepower is 100 lb per sq in.; in certain experimental engines values as high as 125 lb have been reached. These figures will probably be increased by supercharging, improved fuel injection, and better combustion control. In some cases the well-tried features of successful foreign engines, as patented fuel-injection systems and combustion chambers, have been utilized in engines produced in this country, under license arrangements. In other engines, new ideas have been introduced in connection with the combustion process and considerable progress has been made. Although some engine manufacturers prefer the precombustion chamber form of cylinder head, others have adopted the direct-injection plain cylinder head and the turbulent auxiliary chamber types. There is little doubt that the latter is the more efficient, thermally, and can be designed to give easier starting.

The revolutions per minute of commercially successful high-speed Diesel engines at present lie between 600 and 1200; see Table XVII. This is the factor which has in the past confined Diesel engines to applications in which higher speeds were not necessary. Successful development work, however, has proved that higher speeds are practicable. It is justifiable, therefore, to predict that Diesel engines and gasoline engines will soon be competitive in all classes of high-speed service.

Railroads, apparently motivated by the recent successful efforts toward better and faster rail transportation, are now giving serious attention to developments along similar lines, including the use of steam power as well as Diesel engines with electric drive. It may be expected that some of these developments will take a more definite form in the near future. It is quite evident that the demand of the public for faster and cheaper transportation by rail, which has been induced by the enormous growth of motor and air transport, will result in a wider adoption of the principles incorporated in some of the recent successful Diesel-electric locomotives and trains. Larger units of greater capacity to meet the varied requirements of railroad operation may well be developed along similar lines.

The future high-speed Diesel engine will probably develop over 150 hp per cylinder, weigh 10 to 15 lb per hp, run at speeds up to 2500 rpm with piston speeds of 2000 ft per min, and occupy less than $\frac{1}{2}$ cu ft

TABLE XVII

DOMINANT CHARACTERISTICS OF RECENT HIGH-SPEED DIESEL ENGINES*

Location	Service	Brake Horse-power	Rpm	Cylinders		Cycle	Fuel Injection	Pound Fuel per bhp-hr	Horse-power per Cylinder	Piston Speed, ft per min
				No.	Size					
Murray, Kan	Drilling	125	850	6	$5\frac{3}{4} \times 8$	4	Solid	0.43	21	1133
Lima, Ohio	Generator	100	514	5	$7\frac{1}{8} \times 10\frac{1}{2}$	4	Solid	0.40	20	900
Mt. Locke, Texas	Generator	100	600	6	$6\frac{1}{2} \times 8\frac{1}{2}$	4	Solid	0.40	17	850
Abbottstown, Pa.	Generator	85	1200	6	$4\frac{1}{8} \times 6$	4	Solid	....	14	1200
New York, N. Y.	Generator	82	900	6	$5\frac{1}{4} \times 7$	4	Solid	....	14	1050
New Paris, Ohio	Grain elev.	80	600	4	7×9	4	Solid	....	20	900
Los Angeles, Calif	Generator	75	1200	4	$3\frac{1}{2} \times 4\frac{3}{4}$	2†	Solid	....	19	950
Sussex, N. Y.	Compressor	70	900	6	$4\frac{7}{8} \times 6$	4	Solid	....	12	900
Miami, Fla.	Generator	62	1200	6	$4\frac{1}{2} \times 5\frac{3}{4}$	4	Solid	....	10	1150
Stockton, Ill.	Pump	60	1200	6	$4\frac{1}{4} \times 6$	4	Solid	....	10	1200
Foxboro, Mass.	Pump	60	850	4	$5\frac{3}{4} \times 8$	4	Solid	....	15	1133
Port Chester, N. Y.	Compressor	45	900	3	$5\frac{1}{8} \times 7\frac{3}{32}$	4	Solid	....	15	1064
Waterbury, Conn.	Shaft	38	1000	4	$4\frac{1}{2} \times 5\frac{1}{2}$	4	Solid	....	$9\frac{1}{2}$	920
Torrington, Conn.	Generator	18	1200	4	$3\frac{5}{8} \times 4\frac{1}{2}$	4	Solid	....	$4\frac{1}{2}$	900
Medford, Mass.	Generator	12	750	1	$4\frac{15}{16} \times 6\frac{3}{4}$	2	Solid	....	12	845

* *Power*, pages 334 and 335, June, 1937.

† Opposed-piston unit.

per hp. The leading characteristics of representative high-speed Diesel engines recently manufactured in the United States are listed in Table XVII. Choice between four-cycle and two-cycle engines appears to greatly favor the four-cycle type.

The argument that, because the four-stroke cycle has become the generally adopted system for high-speed gasoline engines, it must also be best for high-speed Diesel engines is without value. It is evident that, with the four-cycle method of charging, there is required a certain amount of mechanism which needs considerable care and attention, involving gears, cams, valves, rollers, and rockers; these parts are multiplied by the number of cylinders, as the engines are multi-cylinder. Inasmuch as the simple two-cycle Diesel engine offers great possibilities with a high specific power output for a relatively small floor space, designers will consider more seriously ways and means of improving scavenging methods and lowering the fuel consumption of this type of engine.

In order to insure more complete combustion of Diesel fuel oils in high-speed two-cycle engines, the following points must be considered: higher compression ratios in order to insure a high temperature, high speed, and good turbulence of the incoming fuel charge; shape of the combustion chamber to give rapid heating to the fuel spray; a long free fuel spray; cooling of injection nozzle; use of automatic or unit-injector type of fuel nozzle; and the all-important problem of scavenging. Efficient scavenging is possible with the opposed-piston design in which the unidirectional flow of scavenging air helps to blow the burnt gas thoroughly out of the cylinders with a small amount of surplus air. This compact type of high-speed Diesel engine will undoubtedly be simplified and adopted more generally in the future than in the past.

66. High-speed Diesel Engines versus Gasoline Engines. Because the high-speed Diesel engine is becoming more and more a direct competitor of the modern gasoline engine, it may be of some interest to compare these two engine types, citing the respective advantages and disadvantages of each. The high-speed Diesel engine operating at much higher compression ratios gives higher thermal efficiency values, and, therefore, for a given power output consumes less fuel than the gasoline engine. Moreover, instead of being restricted to the more expensive volatile fuels used in the gasoline engine, it employs lower-grade, cheaper fuels. There is, therefore, a double gain in the matter of reduced fuel costs. A good fuel-consumption value for the average gasoline engine is 0.60 lb per bhp-hr, whereas for the high-speed Diesel engine fuel consumptions of 0.40 lb per bhp-hr are obtained. Oils suitable for Diesel engines cost about one-half the price of commercial gasoline. On this basis the relative fuel costs per brake horsepower-hour of high-speed Diesel and gasoline engines are in the ratio of 1 to 3.

The fuel is supplied to high-speed Diesel engines by positively driven mechanical pumps which meter out the exact quantities of fuel and inject these at precisely the correct times, so that little fuel is wasted and combustion is nearly uniform. With the gasoline engine the mixture of air and gasoline is drawn through the carburetor; and, since the carburetor cannot be said to function satisfactorily under all conditions of load and engine speed, the correct mixture cannot be provided always.

In the gasoline engine there is, at the end of the compression stroke, a homogeneous mixture of gasoline particles, vapor, and air; when ignition occurs combustion must spread from the spark plug point to all parts of the mixture. Although turbulence of the compressed charge plays an important part, nevertheless, there is still a mass of mixture to be ignited. In the Diesel engine having suitable injection and combustion chamber design, each particle of fuel when injected takes up its oxygen supply and becomes a potential center from which combustion may begin; in effect there is an initial mass-ignition instead of the single point ignition method of the gasoline engine.

Because of its higher thermal efficiency, the high-speed Diesel engine loses less heat to the cooling water than a gasoline engine of equal power; the amount of cooling surface may therefore be reduced. These lower cylinder temperatures also reduce the tendency for cylinder, valve, and piston distortion. Except in cases of considerable injection advance the high-speed Diesel engine produces a power stroke of relatively long duration and high average torque over a wide range of speed. The torque-speed curve is much flatter than that of a gasoline engine so that the Diesel engine is more flexible over a wider speed range.

Another advantage of the high-speed Diesel engine is the absence of magneto or coil-ignition apparatus, spark plugs, low- and high-tension cables. The carburetor is eliminated, but against this must be offset the necessity for a fuel pump, distributor, oil pipe lines, and separate injection nozzles in each cylinder. In recent high-speed Diesel engines the electric starting process is so effective that these engines will start readily even in the coldest weather. Unlike the gasoline engine that requires several minutes of warming up before the load can be taken, the load can be applied as soon as the Diesel engine starts. Higher volumetric efficiencies are generally obtained with the high-speed Diesel engine, owing to the absence of the carburetor restrictions, manifold passages, and ports; the air used for combustion enters the cylinders directly from the atmosphere.

Thus far we have considered the various advantages of the high-speed Diesel engine over the gasoline engine. It should not be assumed that the Diesel engine has all the advantages in its favor, because in some respects the gasoline engine is still unrivaled. Some of the Diesel engine designs are not altogether satisfactory in the matter of speed range, control, silence, and clear exhausts. Other disadvantages are greater weight and higher first cost. In the relatively short period of its development, the high-speed Diesel engine has overcome many of its earlier faults and as with the gasoline engine it may be only a matter of sufficient research and design progress until more of its present difficulties are overcome. Superchargers may still further broaden the application of Diesel engines, particularly where weight and space considerations are paramount. This development, already under way in Europe, may prove important in this country, especially in certain marine, railway, and airplane applications.

Up to the present time, manufacturers of high-speed Diesel engines have concentrated upon the industrial and commercial-vehicle types, because of fuel-economy considerations. For use as an automobile engine, where fuel costs are of secondary importance in comparison with performance, high speeds, quietness, flexibility, and lightness, the appeal of the Diesel engine is not an important one at present. If, however, a Diesel engine can be produced which will give the same performance and other advantages of the modern gasoline engine, there is little doubt that it will prove a serious competitor to the gasoline engine, for it will possess the additional advantage of lower fuel consumption, absence of ignition equipment, and the use of a much safer, and probably cheaper, fuel.

The high-speed Diesel engine possesses many inherent qualities that render its application to aircraft purposes not only attractive but also desirable from the points of view of safety and economy. At present, however, Diesel aircraft engines are still in the experimental stage, although their performances are distinctly encouraging. There is little doubt that, given the same attention and development facilities as the gasoline engine, the Diesel aircraft engine will ultimately replace the gasoline engine, at least on all long-distance flights. Some countries are experimenting with Diesel aircraft engines, and in several places production types of engines are in commercial service. Among the more important Diesel aircraft engines special mention should be made of the Beardmore and Phoenix (British); the Junker, Maybach, and Mercedes-Benz (German); Packard, Guiberson, and Deschamps (American); Clerget and Coatalen (French); and Fiat (Italian).

CHAPTER X

INFLUENCE OF SHAPE OF THE DIESEL COMBUSTION CHAMBER

The main purpose of every combustion chamber is to ensure an efficient combustion of the fuel supplied. In air-injection Diesel engines, the fuel is atomized by means of high-pressure air and mixed with air in the combustion chamber during the combustion period, owing to the turbulence created by the injection air. In these engines combustion does not present any great difficulties. With mechanical injection of fuel, the fuel is injected in the shape of a pulverized conical or fan spray, which does not create any turbulence and does not mix readily with compressed air. Hence, some relative motion of air is required to facilitate the mixing of the atomized fuel and the air available for combustion in a Diesel engine that uses mechanical injection.

67. Fundamental Requirements of a Rational Combustion Chamber. Turbulence is created naturally during the compression stroke of the piston and sometimes during the suction stroke in every known type of combustion chamber. In the combustion chamber shown at *A* in Fig. 95, the air is moving parallel to the cylinder axis and in the same direction as the piston during the compression stroke. If the top of the piston were cup-shaped in the center and its sides came up close to the cylinder head, as shown at *B* in Fig. 95, then at the end of compression there would be a considerable movement of air. This air movement would be directed from all sides radially towards the axis of the cylinder, owing to the displacement of the air around the cup-shaped cavity in the center of the piston top.

The relative position of the fuel jet in the combustion chamber should be such that the direction of the air movement is across the fuel jet during the injection period. If the air stream were in the same or opposite direction of the fuel jet, then, as soon as combustion began, products of combustion instead of fresh air would be carried through and mixed with the yet unburned fuel in the jet. This requirement does not apply to various precombustion chamber engines, where other features peculiar to each particular design may be of greater importance. Because of the significance of the relative position and directions of the fuel jet and the air turbulence, the problems of the formation of air streams in combustion chambers during the combustion period should receive careful consideration.

Figure 95*C* shows a typical combustion chamber of a Diesel engine in which the air comes into the chamber through the central throat in a solid stream and, meeting the opposite wall of the chamber, divides into two streams, each curling itself up in opposite directions. Two to four jets of fuel at about 80 degrees included angle in one plane usually give

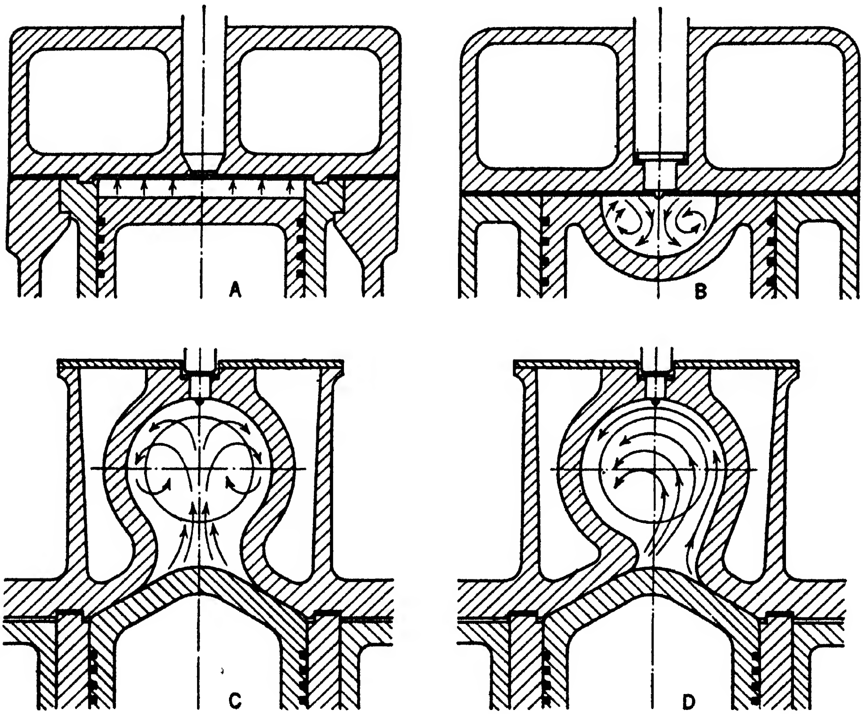


FIG. 95. Combustion chambers showing direction of air movement during compression.

the best results with this combustion chamber, and generally speaking this should be considered a satisfactory design. The throat may be made somewhat wider without departing from the principles of construction.

A similar design is shown in Fig. 95*D*, but with somewhat different arrangement of the throat, which is made tangential to the one side of the circular chamber. In this design the turbulence in the combustion chamber will be of a different type, one continuous vortex as shown. A single fuel jet would probably give the best results with this combustion chamber design and the indicated air turbulence.

In designs *C* and *D*, Fig. 95, the fuel jet is given an excellent opportunity to mix with the air, as all the air available for combustion purposes in the cylinder is concentrated in the combustion chamber. All clearances and air pockets outside the combustion chamber proper should be reduced to a minimum or eliminated. Unless this requirement is considered when designing the combustion chamber, difficulties may be experienced in obtaining a high mean effective pressure in the cylinder, although the fuel consumption may be satisfactory at lower mean effective pressures.

Efficient combustion of the fuel in a Diesel engine is dependent upon several fundamental requirements which may be summarized as follows: concentration of all the air available in the cylinder in one space, as compact as possible, and of a shape suitable for distribution of jets radiating from one point; use of as few fuel jets as possible, radiating from one point in the combustion space in order to be in a favorable relative position to the air movement during the compression stroke; combustion space so shaped as to insure a definite movement of air, preferably of moderate velocity and at right angles to the fuel jets, at the end of the compression stroke; combustion space not so narrow or flat as to cause the jets to touch the walls.

Even though the combustion chamber of an engine may fulfil the above requirements, the fuel-injection system may be improperly designed and unsatisfactory results obtained. With the open type of combustion chamber, designers find that better results are obtained by using as many fuel jets as practicable. With a flat type of combustion chamber, as shown at *A* in Fig. 95, and with a wide angle between jets, there is no advantage in using more than five jets, for then the expanding cones of the jets would overlap. Another frequent fault in the design of fuel equipment is too long a period of injection. The time of fuel injection is closely correlated with the turbulence in the combustion chamber, because in many designs the air movement is reversed as soon as the piston starts on the down stroke. If a long injection period were used, then as soon as the turbulence in the combustion chamber is reversed the products of combustion and not the fresh air would be passing through the fuel jet, thus retarding the rapid and complete combustion of the fuel.

68. Heat Loss to Walls during Combustion. Heat losses during combustion from the burning fuel to combustion-chamber walls are generally due to two causes, direct radiation and direct contact with the hot gases. In the losses from direct radiation temperature plays an important part, for the amount of heat radiated is proportional to T^4, the absolute temperature of combustion. The time element is also important. It is well known that if the combustion of a fuel is slow and inefficient, the heat losses to the cylinder walls may be excessive, although the temperature of combustion may be lower than with a more efficient combustion process. Thus, a high combustion temperature, which is a sign of efficient combustion, is very desirable.

Heat losses by direct contact between hot gases and metal walls depend primarily on the velocity of hot gases in relation to cooled walls. Hence, in all combustion chambers having high velocity turbulence, the heat losses from hot gases to combustion-chamber walls are relatively high, between 30 and 40 per cent of the heat of combustion of the fuel supplied. The same reasoning applies to the compression of the air. In all combustion chambers with high velocity turbulence heat is lost from air to cylinder walls during compression, and consequently a higher compression ratio or electric heating plugs may be necessary to facilitate starting.

The amount of the heat losses from the hot gases to cylinder walls does not affect appreciably the over-all thermal efficiency of the combustion process. A temperature exceeding 3500 F at the end of combustion seems to have a retarding effect on further combustion of the fuel. Although excessive heat losses from gases to combustion-chamber walls during combustion are not detrimental to the thermal efficiency of an engine, they may produce considerable heat stresses in the cylinder-head walls and eventually result in cracks. Heat losses during the compression stroke are undesirable, for the starting of the cold engine becomes more difficult, and the running of the engine may become uneven. From this point of view the combustion chambers of the "open" type are much more advantageous.

To overcome the difficulties of high heat stresses in combustion chambers with high turbulence velocity, designers now use metals for cylinder-head castings having better heat conductivity and lower modulus of elasticity. The importance of this will be seen from an approximate equation for determining the heat stresses in water-cooled combustion-chamber walls.

$$S = C \times E \times c \times h \times w \times F \times Ihp \times \frac{t}{D^2}(1 + R) \div k \qquad (54)$$

where S = heat stresses in wall, pounds per square inch.
C = coefficient, depending upon units chosen.
E = modulus of elasticity, pounds per square inch.
c = coefficient of linear expansion of metal of the walls.
h = heat of combustion absorbed by walls, 25 to 35 per cent.
w = fuel consumption in pounds per indicated horsepower per hour.
F = heating value of fuel, Btu per pound.

Ihp = indicated horsepower.
t = thickness of wall, inches.
D = cylinder diameter, inches.
R = stroke to diameter ratio.
k = thermal conductivity, Btu per square inch per inch thickness per degree F temperature difference per hour.

Whatever the numerical value of the heat stress is, from equation 54, the following conclusions may be made: the greater the modulus of elasticity of the metal, the higher the stress; the greater the coefficient of linear expansion, the higher the stress; the thicker the walls, the higher the stress; and the greater the thermal conductivity of the metal of the wall, the smaller the stress, owing to the smaller temperature drop through the wall. Consequently for engines with high power output and heavy heat losses to the walls a metal with a small modulus of elasticity and of high conductivity is desirable to reduce heat stresses. Some aluminum alloys are suitable for this purpose.

The variation of temperature in the combustion-chamber walls is much greater near the inner surface, and it changes very little on the cooled side. In the early oil engines with mechanical injection of fuel it was considered desirable to have part of the cylinder head hot, to assist combustion. As the knowledge of combustion and fuel injection advanced, it was found that hot surfaces in combustion chambers were generally unnecessary. A hot surface in the path of the oil jet may be harmful, for the oil striking the hot surface "cracks" and produces a smoky exhaust. In the older types of low-compression oil engines, to avoid fuel cracking, it was found necessary to inject a quantity of water into the hot bulb to keep its temperature within desirable limits. This cooling water, which was lost with the exhaust, was sometimes as much as ten times the quantity of fuel. The effect of water injection on the cleanness of the exhaust and engine efficiency was, however, very marked.

69. Spherical Combustion Chamber Preferable to Flat Combustion Chamber. In designing a combustion chamber, the shape of every portion of it should be very carefully considered as it appears not only when the piston is in its top dead center position but also when the piston is in several intermediate positions from the beginning of the injection to the end of the combustion period.

The approximate velocities and expected directions of the air streams in the combustion chamber should be plotted and the influences of these streams on the shape of the fuel jets should be carefully studied. If the air stream is across the fuel jets, it is reasonable to expect that the fuel jets will be either deflected or at least bent to conform to the direction of the air flow. As soon as the piston gets over its dead center, the direction of its movement is reversed; this is bound to affect the turbulence in the combustion chamber in one way or another.

It is also important to consider carefully the possible influence of the shape of combustion chamber on the heat losses by modifying the air velocities in various passages and also the ratio of the combustion-chamber surface exposed to the heat of combustion to its volume. Consider two very simple types of combustion chambers for the same engine, one perfectly flat and cylindrical, the other—spherical. Let D = cylinder diameter, inches, L = length of stroke, inches, $\frac{L}{D} = R$, and compression ratio = 14 to 1.

Flat cylindrical combustion chamber:

Displacement volume, $$V = 0.7854 \times D^2 \times L = 0.7854D^3R \tag{55}$$

Combustion-chamber volume, $$v = 0.7854D^3 \times \frac{R}{13} = 0.06D^3R \tag{56}$$

Combustion-chamber surface, $$A = (0.7854 \times 2 \times D^2) + \left(\pi D^2 \times \frac{R}{13}\right)$$

$$A = 1.57D^2 + 0.242D^2R \tag{57}$$

Surface to volume ratio, $$R'_f = \frac{1.57D^2 + 0.242D^2R}{0.06D^3R} = \frac{26.2 + 4.03R}{DR} \tag{58}$$

Assume $R = 1.25$, a frequent ratio for $\frac{L}{D}$; then $R'_f = \frac{25}{D}$ (59)

Spherical Combustion Chamber. The volume of the spherical chamber will have to be the same as for the flat type of chamber, for a given compression ratio, in this case,

$$v = 0.06D^3R \tag{60}$$

But volume of a sphere, $$v' = \frac{\pi d^3}{6} \tag{61}$$

Where d = diameter of spherical combustion chamber.

$$\frac{\pi d^3}{6} = 0.06D^3R$$

$$d = 0.485D\sqrt[3]{R} \tag{62}$$

Combustion-chamber surface, $$A' = \pi d^2 = 0.74D^2R^{2/3} \tag{63}$$

Surface to volume ratio, $$R'_s = \frac{0.74D^2R^{2/3}}{0.06D^3R} = \frac{12.33}{D\sqrt[3]{R}} \tag{64}$$

Assume $R = 1.25$, as before; then $R'_s = \frac{11.5}{D}$ (65)

It may be seen by comparison of equations 59 and 65, that the surface exposed to the direct flame of combustion in the spherical combustion chamber is only about one-half of that in a flat combustion chamber, for the same stroke-bore ratio and the same compression ratio. For this reason, smaller total heat losses to combustion-chamber walls may be expected in a spherical chamber during combustion. This condition may be completely changed, however, if a high velocity turbulence is provided in the spherical chamber.

A flat cylindrical combustion chamber is obviously not rational, for it does not provide for the expansion of the fuel jets, and because of its flatness it is almost impossible to prevent the jets from grazing the top of the piston. The modified form of it, as shown in Fig. 97*B*, is superior, and is well adapted for the type of rotational turbulence used in it. From the point of view of effective combustion, the spherical chamber appears to be more suitable, for it provides a more concentrated volume of air for combustion and permits a better distribution of the fuel jets. The open type of combustion chamber as shown in *B*, Fig. 95, should also be satisfactory, and with an efficient injection system could probably be used even without the air swirl.

A combustion chamber approaching a sphere in shape but with a wide throat connecting it to the cylinder should also be considered an "open" type chamber; but it would be wrong to insist that a perfect spherical shape is the best for a combustion chamber from the point of view of efficiency. If the shape of the fuel jets and the natural air turbulence in it are considered, the best shape for a chamber of this type would be that of a double-spherical lens with a different curvature for each side. A combustion chamber of this type is shown in Fig. 96; the top half is almost half a sphere and the bottom half is described by a radius from a center located in the fuel valve. The probable air movement is indicated by the curved lines and its influence on the shape of the jets is shown by the shaded area. The throat of this combustion chamber is fairly wide, so that little heat is lost by the air during combustion and the starting of the engine should be relatively easy. The difference in curvature in the top and bottom halves of this combustion chamber is entirely due to considerations of the shape of the fuel jets and air turbulence.

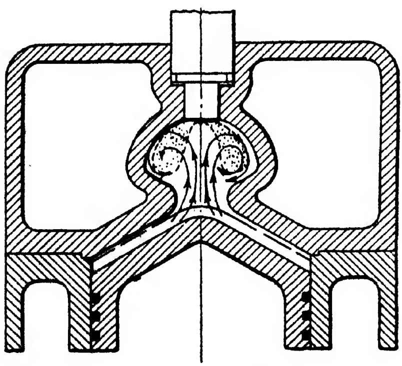


FIG. 96. Spherical shaped combustion chamber.

70. Effectiveness of Fuel Injection. Fuel injection efficiency is of the greatest importance in all combustion chambers of the open type or those having low velocity turbulence. All early oil engines with mechanical injection of fuel were either of the so-called hot-bulb type or of the precombustion-chamber type. In hot-bulb engines the fuel was not injected in the atomized state but was simply forced through a small hole onto an incandescent surface of the hot bulb, and the hot bulb was relied upon partly to atomize and partly to vaporize the fuel, as described in Art. 43. This early method of fuel injection could not be used at present in any open-type combustion chamber with hope of even moderate success. Such important factors as injection pressure, rate of fuel delivery, constant spray velocity, length of injection period, etc., were probably unknown or not considered. The hot-bulb cylinder design was reasonably satisfactory for the requirements of its time. Even modern fuel-injection equipment presents such technically complicated problems that many manufacturers are still using precombustion chambers which do not require a high injection efficiency.

Apart from exacting technical requirements of the modern fuel-injection equipment, there are some very important points which must be considered in conjunction with the cylinder-head design and the type of turbulence used in the combustion chamber. The most important factor is the length of the fuel-injection period in terms of crank angle; if the injection period is too long, part of the fuel will be injected after the top dead center. This should be considered highly unsatisfactory. It is important in open-type combustion spaces that all the fuel at full load should be injected before top dead center and the injection completed at least 5 degrees before it. If this requirement is not satisfied, the power developed will be reduced and after-burning, smoky exhaust, and lowered thermal efficiency will result. This is largely due to the reversal of the direction of turbulence after the piston passes top dead center. The length of the fuel-injection period should not be more than 10 to 15 degrees crank angle for slow-speed engines and 20 to 30 degrees for high-speed engines. This condition may not be easy to satisfy in four-cycle engines, as it will require large fuel-pump plungers and rapid cams. With precombustion-chamber engines or in combustion chambers with high velocity turbulence the length of the fuel-injection period is not of great importance, as the preliminary combustion in the precombustion chamber is more like an explosion and occurs after all or most of the fuel has already been injected.

The question of ignition quality, as explained in Art. 15, has attracted much attention since the small high-speed Diesel engine began its rapid development several years ago. In the early days of the high-speed engine the opinion was sometimes expressed that the upper limit to the speed of rotation would be controlled by lack of sufficient time for the combustion of Diesel fuels at high engine speeds. Actually, no such limit has been reached, although small Diesel engines have been run up to speeds of 4000 rpm. But a difficulty not connected with rotational speed has persisted in regard to the combustion of fuel oils in the high-speed engine—namely, ignition delay. Between the start of injection of the fuel into the combustion chamber and the instant at which the fuel begins to burn there is an interval—the delay period—which for a given engine, fuel, and injection timing is approximately constant when measured in degrees of crank angle. A long ignition delay allows a large amount of fuel to be injected before burning starts and, in consequence, the rate of pressure rise is high immediately after combustion. The effect of this is to produce rough running and Diesel "knock." On the other hand, if burning starts too soon after the beginning of injection, although combustion is relatively smooth and shockless, another difficulty may arise, that of inadequate mixing of the fuel and air before combustion, with resultant power reduction and increased fuel consumption.

Conditions tending to retard combustion naturally lengthen the delay period and accentuate "knock" in a Diesel engine. Thus, reduction in the density of the ingoing air by throttling increases the delay and knock. Conversely, supercharging tends to reduce the delay period and to promote smooth running. Since the method of mixing the air and fuel and their physical condition during the mixing process have an important effect on the initiation of combustion, it is evident that engine design must have a great influence on ignition lag.

The position of the injector in the combustion chamber and the dispersion (atomization and spreading) of the fuel jets are also of considerable importance. With the combustion chamber shown in Fig. 96, the best position is in the center of the top of the combustion chamber, with the jets issuing at about 70 degrees included total angle for a small cylinder bore. The best size and shape of the holes in the fuel nozzle are generally determined on an engine by tests, as judged from fuel economy, visibility of the exhaust gases, and torque.

The designer must have some knowledge of the shape of the fuel jet when it has fully developed in the combustion chamber; or differently expressed, knowledge of the shape of the hole which is bored into the compressed air by the jet. This shape is, for the ordinary round hole, of sufficient bore to length ratio; a cone with a top angle of about 20 degrees ending in a hemisphere, as shown in Fig. 97. The several cones should so fill the combustion chamber that they do not reach the walls of cylinder or piston, and that there is sufficient room around each of them to give the air and fuel ample opportunity to come into contact with each other. The number and the required length of the cones can be determined; and as this length is primarily dependent on the size and length of the hole, these dimensions can be fixed.

Another important factor to consider with regard to fuel injection is the penetrating power of the fuel jet. In the past many experiments have been carried out for determining the factors which are mainly responsible for the greater or smaller penetration, but the results of these tests appear to be contradictory. It is, of course, quite natural to expect that a fuel jet issuing from a large diameter orifice will have a

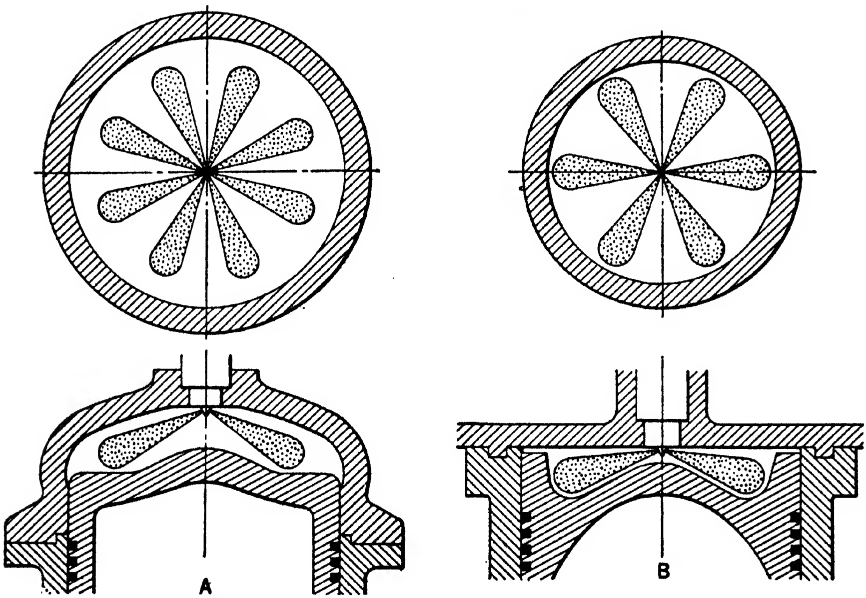


FIG. 97. Combustion chambers showing distribution and shape of fuel jets.

greater penetrating power than a jet from a smaller diameter orifice; this obvious fact has been confirmed by experimental data. With regard to other factors, the results appear to be confusing; for instance, a higher injection pressure does not always improve the penetration, for usually the fuel gets better atomized and thus the jet more quickly loses its penetrating power. Higher compression pressure in the cylinder also certainly reduces the penetration possibilities of the fuel spray. The combustion-chamber turbulence may affect penetration, for it can deflect or curve the jet. The quantity of fuel injected per nozzle affects the penetration very considerably, as does also the chattering of the needle valve in the fuel injector. The increased length of the holes in the nozzle reduces the expansion angle of the jet and increases its penetrating power.

It is erroneous to compare the jet of fuel—as is often done—to shot of very small caliber leaving the muzzle of a shot gun. The fuel leaves the orifice of the hole in solid formation, as can be proved by injecting into a vacuum. As soon, however, as the jet meets air in its course, it is broken up into small particles, which will be smaller the higher the muzzle velocity and the greater the spreading. Theories about penetration and atomization have been presented on the assumption that the first particles leaving the orifice go farthest. These theories have been proved fallacious, because the first particles emerging are immediately swept off their course by the surrounding air of comparatively high density. The emerging jet has been compared to an advancing army leaving a fortress. The soldiers in the front line are continually being killed off and their places taken by others pushing up from behind, and from above we see a wedge of soldiers slowly advancing into the enemy's territory. Tests have been made to determine the distance over which the ultimate leader will advance, but it is better to determine the spray velocity at various distances from the orifice for various diameters, and at constant fuel pressure and air density. For a hole of 0.025-in. diameter the spray velocity is about 22 per cent of the muzzle velocity at 3-in. distance, for a hole of 0.007-in. it is about 7 per cent; in each case the length of the hole is twice the diameter.

It is impossible to give any general recommendations with regard to fuel injection for either the precombustion-chamber type or for the combustion chamber with high velocity turbulence, for in both these types the combustion is much more dependent on the air turbulence than on the injection efficiency. In general, for both types of combustion chambers the exhaust is never very clean, and the fuel consumption never as low as can be obtained in the open-type combustion chambers with low velocity turbulence. Both the precombustion and the high-velocity turbulence types of combustion chambers should be considered as using fuel injection and combustion methods which do not meet the complete technical requirements of existing high-pressure direct-injection systems.

71. Combustion Chamber Efficiency. Many engineers make the mistake of judging the efficiency of the combustion chambers of internal-combustion engines by the results based on the delivered or brake horsepower. These results have little connection with the combustion-chamber efficiencies in different engines, and can only be used for comparison of combustion efficiency in the same engine and under the same conditions of operation. This is true because the mechanical losses in various engines differ considerably; and these losses have a far greater influence on the engine output than any possible variations in combustion efficiency, provided the fuel-injection equipment and combustion chamber are not too poorly designed and correlated. The mechanical losses in any engine depend on many factors, of which the following are the most important: stroke to bore ratio, length and tightness of bearings, friction of pistons and rings on cylinder walls, indicated mean effective pressure, size of valves in four-cycle engines, engine speed, ratio of connecting-rod length to the crank, weight of reciprocating parts, balancing of cranks, and type of drive used for camshaft.

Engine manufacturers are not concerned about the indicated fuel consumption of an engine, because in general the buyers of engines are interested only in the engine output and the corresponding fuel consumption. This is a most unwise attitude on the part of both the engine builders and the buyers, for the life of the engine depends more on the quality of its combustion than on its mechanical efficiency. Low mechanical efficiency often is due to long and heavy pistons, too many piston rings, long bearings, etc., all of which can contribute only to long life of the engine, whereas poor fuel combustion efficiency often results in overheated pistons, carbonized piston rings, wear on cylinder liners, carbonized cylinder heads, contaminated lubricating oil, wear on bearings, etc. Yet, two engines with very different characteristics may have exactly the same fuel consumption, and an engine with the lower mechanical efficiency may have a worse fuel consumption per brake horsepower than an engine with a high mechanical efficiency and a poor combustion efficiency.

Hence, it is essential that the engineer know the engine mechanical efficiency and its fuel-combustion efficiency, and judge the efficiency of fuel injection and the effectiveness of the combustion chamber by the fuel consumption per indicated horsepower, figures which are not influenced by the mechanical efficiency. In other words, the fuel consumption figures per indicated horsepower give direct information on the cylinder's injection and combustion efficiency, which is the most important factor contributing to continued satisfactory performance of the engine.

72. Fuel and Its Influence on the Combustion Chamber. Gasoline engines are designed to operate at the highest possible compression ratio consistent with freedom from detonation and pre-ignition under any circumstances. Diesel engines are designed to operate at the lowest compression ratio which will effect combustion under all conditions. In the gasoline engine and with the average fuels available the designer is compelled, except with very small cylinders, to limit the compression ratio to about 6.5 to 1. In the Diesel engine, to effect combustion without an unduly prolonged ignition-delay period under the most adverse circumstances, the designer is limited to a compression ratio of about 14 to 1 despite the fact that the heavier petroleum products may have a lower self-ignition temperature than gasoline. Between these two limits lies a wide neutral zone unattainable by either type of engine except by the use of anti-detonating compounds on the one hand or pro-detonating compounds, hot plates, etc., on the other hand; however, the use of these compounds may produce evils greater than those which they are designed to overcome.

It used to be, and to some extent still is, argued that the combustion process in a Diesel engine, including the vaporization of the liquid particles, the subsequent admixture of this vapor with sufficient air for combustion, and the combustion of the air-vapor mixture thus formed must necessarily take so long a period of time as to render the high-speed Diesel engine either impracticable or so inefficient as to be of very little service. Such arguments appear to be entirely unfounded; not only have they been refuted by practical results but they also appear to be based on false premises. In fact, with high-turbulence combustion chambers it has been found that ignition lag in degrees crank angle is constant (or may even decrease) with engine speed because the turbulence effect increases with speed. Except that a very minute quantity of vapor must be present to provide the initial nucleus of flame, there does not appear to be the slightest basis for supposing that the rest of the fuel must be vaporized completely before combustion can take place. On the contrary, it appears far more probable that combustion actually takes place from the surface of the liquid droplet, and that it is supplied with fresh oxygen and the burnt products are scoured away by the rapid passage of this droplet through the air. The really essential condition for high speed is sufficient relative motion between the drops of fuel and the air—a condition which can be met by rapid movement either of the fuel or of the air, or both (provided, of course, their direction is not the same).

It does appear, however, that for any given fuel the relative proportion of the ignition delay period must increase as the engine speed is raised, that the extent of this period in relation to the rate of injection determines the maximum pressure, and that, therefore, the higher the speed the higher should be the maximum pressure. In other words, the higher the speed the greater the departure from the constant-pressure and the nearer the approach to the constant-volume cycle.

The ignition delay period can be shortened by employing a higher compression ratio, thus increasing both the temperature difference between the highly compressed air and the ignition temperature of the fuel, and also the density and therefore the intimacy of the two. Increase of compression, however, is not altogether a satisfactory method. Although it reduces the ignition delay and, therefore, also the rise of pressure above that of compression, these are almost completely compensated for by the higher initial compression pressure, and the slightly higher efficiency due to the increased expansion ratio is almost counterbalanced by the increased friction and heat losses.

73. Types of Combustion Chambers. It is impossible to describe all the existing combustion chambers used in various engines. A number of typical combustion chambers have been chosen, representing the several lines of development. The general principles of design rather than the combustion chambers of any particular engines are illustrated and briefly described. *A* in Fig. 98 shows a modernized modification of the old Brons design. The main difference is that the precombustion chamber is arranged in the center of the cylinder head, and fuel is injected into it by a modern injector. A number of precombustion chambers like this are used now, all differing slightly in internal shape, otherwise very similar; see Fig. 68. The general characteristic of this type is that the combustion chamber is divided into two parts, one into which the fuel is injected and where it explodes, and the other where the combustion is completed. The volume of the precombustion chamber varies approximately from one-fifth to one-sixth of the total volume of the compression space.

B in Fig. 98 shows a typical combustion chamber of the air-cell type. The fuel ignites as it is injected into the clearance space. As the combustion space increases the air flows from the cell into the cylinder to continuously support combustion, thus assuring more nearly constant pressure; see also Fig. 84. The "Comet" engine shown in Fig. 82 has the spherical combustion chamber located in the cylinder head just to the left and above the cylinder block. A number of other similar type heads are also used under different names, differing slightly in shape and location of the air cell.

B in Fig. 97 shows the Hesselman combustion chamber. It is the first open-type combustion chamber with a rotational turbulence produced by masked valves. This design has been extensively imitated, and the rotational turbulence is used now in a number of other open-type combustion chambers. Figure 62 shows the latest development of Hesselman's low-compression oil engine with rotational turbulence and spark-plug electric ignition. *B* in Fig. 95 shows a hemispherical combustion chamber in the top of the piston. Rotational turbulence of the air produced radially towards the axis of the cylinder and across the fuel jets materially aids combustion. Further development of the open-type combustion chamber is shown at *C* in Fig. 98; the rotational and radial turbulence in it combine to form an effectual dual type turbulence in the combustion chamber. At *D* in Fig. 98 is shown a partly conical, partly spherical type of combustion chamber. The turbulence in it is not very satisfactory and combustion is uncertain because of the inability to obtain the proper relative directions of the air and fuel jet.

All chambers of the precombustion type have the advantage that they do not require efficient fuel injection. But they may require heating plugs for starting and usually the combustion is not perfect. Consequently the exhaust is not very clean, and the amount of carbon deposited in the cylinders is greater than it is with open-type combustion chambers. The high velocity turbulence chambers shown at *B* in Fig. 98 and Fig. 82 are a compromise between the precombustion-chamber

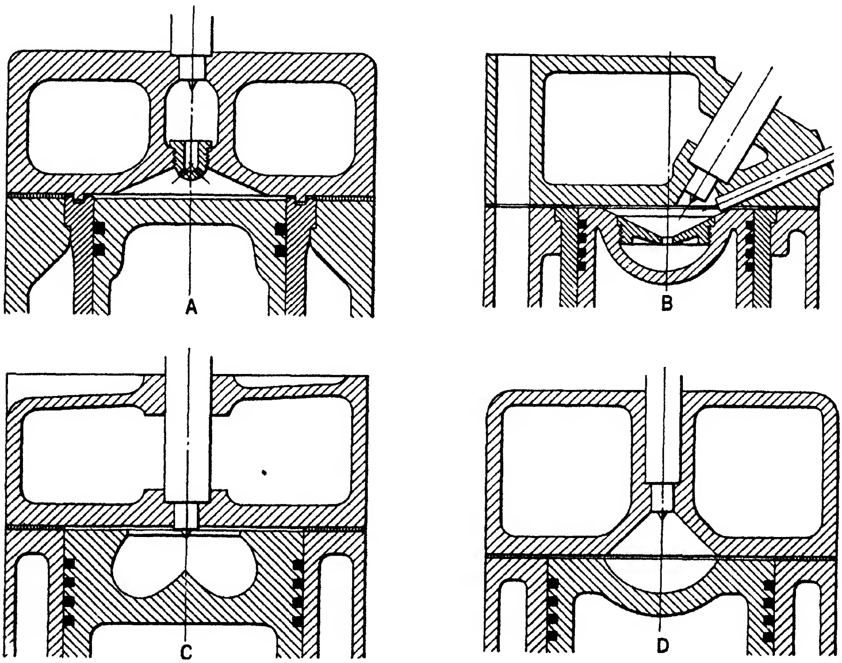


FIG. 98. Four types of combustion chambers for oil engines.

and the open-combustion chamber types. Although they require heating plugs for starting from cold, the combustion process in them is of a much higher efficiency and of a better quality than in the precombustion-chamber type. A reasonably clear exhaust and a high mean effective pressure can be obtained.

The open-type combustion chambers, shown at *B* in Fig. 95 and *B* in Fig. 97, require very efficient fuel-injection equipment and design to get the best performance. Although this type is not capable of very high indicated mean effective pressures, 110–120 lb per sq in. being probably the upper limit, excellent combustion with invisible exhaust can be obtained in them particularly with rotational turbulence of moderate velocity. The resultant fuel consumption should be about 0.30 lb. per ihp-hr. The cylinder-head depression type of open combustion chamber, illustrated at *C* in Fig. 95, has all the advantages of the piston-depression type of combustion chamber, and the type of fuel injection is easier to choose because the combustion chamber turbulence is more favorable.

Combustion chambers of the open type shown in Fig. 96 are somewhat similar to the last discussed type, but are more rationalized. The internal shape is designed to be particularly suitable for the normal natural turbulence in them and to conform to the shape of the fuel spray. The shape of the throat should be adapted to form an air stream of the most suitable shape. High mean effective pressures, invisible exhaust, and low fuel consumption per developed horsepower are possible with this design. The fuel injection arrangement is simple; two or three jets would probably be sufficient. Other advantages of this type are freedom from Diesel fuel knock and the fact that it is not as sensitive to changes in injection timing as many of the other types of combustion chambers.

CHAPTER XI

DIESEL FUEL INJECTION

The heart of the Diesel engine is the fuel-injection system. The maximum speed and power output of an engine are governed largely by the design of its fuel-injection system; to a certain extent, also, the acceleration, regular running, and fuel economy depend upon this important unit of the Diesel engine. The Diesel process begins in the injection pump, where hydraulic phenomena of various types occur. Another series of hydraulic effects occurs in the fuel-injection nozzle and at some distance beyond it; the size of the spray orifice reacts on the delivery characteristics of the injection pump. It is often stated that the Diesel engine is simpler than the gasoline engine because it eliminates the carburetor and ignition system. However, the fuel-injection system on the Diesel, which must be substituted for these parts, brings up many problems; it is usually more costly and difficult to make and maintain than the gasoline engine parts which it replaces.

74. Fuel-injection Requirements and Methods. It is generally assumed that the sole function of the fuel system is to measure out quantities of fuel, no attention being given to other vital factors. In practice, the fuel must not only be measured out accurately at all loads but must also be delivered into the combustion chamber at the time, in the form, and at the rate suitable for the conditions under which the engine is required to operate. With any type of Diesel engine, whether high or low speed, the major problems are to bring the fuel and air within the cylinder in intimate contact with each other and to maintain a high relative speed between the fuel particles and the air, in order constantly to sweep away the products of combustion as they are formed and to replace them with fresh air. The jet or jets of fuel must penetrate far enough through the compressed air in the combustion chamber to reach the farthest outlying air, and at the same time the spray must be fine enough to spread out on all sides. This condition can be accomplished readily when high-pressure air is used for injecting the fuel, because the air stream then serves both to atomize the fuel and, by creating additional and violent turbulence, to distribute the finely pulverized fuel more or less uniformly throughout the combustion chamber.

It is worth while to establish the fact that in solid injection it is compressed air which also causes the atomization—the air in the compression space of the engine cylinder. The difference between the two injection methods, therefore, is that, in air injection the fuel is atomized by the air mixed with it before entering the cylinder, whereas in mechanical injection the atomization takes place by the air in the cylinder. To call a mechanical injector an atomizer is therefore a misnomer. Indeed, there is no atomization when there is no air; and a mechanical injector, injecting into a vacuum, gives a solid jet without any dispersion over a long distance.

With mechanical injection of Diesel fuel the problem is more difficult than with air injection and the conditions are more conflicting. On the one hand, in order to reach the outlying air, there must be a fairly "hard" or penetrating jet; on the other hand, to give adequate dispersion of the particles of fuel, it is necessary to create a "soft" or atomizing jet. Thus mechanical injection starts with the disadvantage of having two apparently irreconcilable conditions to fulfil; and it is in the attempts to fulfil or circumvent these conditions that much ingenuity has been displayed in recent years. No complete reconciliation has yet been found, but compromises have been attained which render the mechanical-injection more efficient than the air-injection engine and thus make possible the high-speed self-contained oil engine of light weight. The mechanical-injection Diesel engine generally shows a lower fuel consumption than the carburetor and the air-injection engine. The mechanical-injection engine develops a lower mean effective pressure than the other two types. Because of the high thermal efficiency of its cycle and the absence of power-consuming accessories, the mechanical-injection engine possesses an inherent potentiality for a high mean effective pressure. The reason for this deficiency is the lack of satisfactory combustion control which offsets the advantage of its superior engine cycle by necessitating the use of a large percentage of excess air. Future progress of the mechanical-injection engine is dependent upon better combustion control, with which the processes of turbulence, atomization, dispersion, and penetration are closely interrelated.

Of the two types of fuel injection, air and mechanical, the mechanical has historically the first rights, as it was well known before Diesel gave the watchword, air injection. Some of Diesel's predecessors used mechanical injection, and it can even be upheld that Diesel retarded progress by impressing on the technical world the axiomatic principle that no satisfactory fuel injection was possible without the use of compressed air. It seems that Diesel's positive opinion so overawed the designers and experimenters during his lifetime that very few attempts were made to disprove his opinion during the years 1893 to 1913. These few attempts were rewarded with some measure of success, not sufficient, however, to induce other designers to develop mechanical injection further. Ten years passed, including four years of the World War, before mechanical injection was taken up seriously by the foremost Diesel engine builders, and almost twenty years before some of the best-known makers of large engines adopted it.

The *air injection* of fuel oil was used in the early experimental Diesel engines, and is being used in some large present-day engines because it has certain advantages not possessed by the solid-injection system; see Arts. 47 and 58. The compressed air is at a pressure of about 1000 lb per sq in. and thus provides an excellent means of forcing the oil through small passages into the cylinder when the needle valve opens. The expansive power of the air, as it drops in pressure from 1000 to 500 lb upon entering the cylinder, increases its volume and thus aids in further atomizing the fuel oil. A well-designed air-injection nozzle atomizes the oil so completely that it does not wet the hand, but floats as mist.

In the *mechanical-injection* process, the energy for atomization and dispersion is contained solely in the oil as a result of its pressure. Storing in the fuel oil the same amount of energy as is used in the air-injection process would involve pressures exceeding 10,000 lb per sq in. There are serious objections to the use of such pressures in engineering practice on account of the high stresses and ensuing wear in the pump mechanism and nozzle parts. Even if applied, the result would not be the same as with air injection, because of the difference in the atomization process. In many engines, oil pressures of 3000 to 6000 lb per sq in. are used. There is a tendency to employ lower injection pressures, making easier manufacture of pump and nozzle parts, thus prolonging the life of the engine and its component parts.

75. Air Injection of Diesel Fuels. Diesel engines which inject the fuel with the aid of high-pressure air require a compression pressure of 500 to 525 lb per sq in. and an air-injection pressure of 800 to 1000 lb per sq in. At the beginning of the power stroke the high-pressure air is admitted through the injection nozzle for a period of about 10 per cent of the stroke. The oil previously (or simultaneously) deposited in the injection nozzle is picked up by this injection air and gradually sprayed into the combustion chamber, where combustion occurs at constant pressure during the period of injection. There is no valve after the oil and air are mixed, thus avoiding cut valve seats. There are few restricted areas and sharp changes in direction to avoid excessively high air pressures and to prevent clogging of the fuel valve passages with dirt, asphalt, or carbonized oil. Atomization may be effected by a device that does not require a close relation between the size of openings and the amount of oil; the maximum power of the engine is limited only by the amount of oxygen (air) available for combustion and not by the capacity of the nozzle. This gives a remarkable flexibility for varying loads.

Air-injection fuel nozzles, also called fuel valves, may be divided into two classes, open and closed. The open-nozzle fuel valve has no special means to stop the flow of fuel from the orifice, and the fuel delivery is controlled by the fuel pump. The closed-nozzle fuel valve has a spring-loaded valve at the discharge orifice, and the fuel delivery may be controlled either hydraulically by a pump, or mechanically by cams, eccentrics, and levers.

The freedom from clogging permits the use of the lowest grades of fuel oil with the *open-nozzle fuel valve* without interruptions for cleaning and grinding the valve. After-dripping of fuel oil into the cylinder may be a serious disadvantage. An open-nozzle fuel valve is used with the Allis-Chalmers air-injection engine shown in Fig. 63; the method of fuel injection is described on page 158.

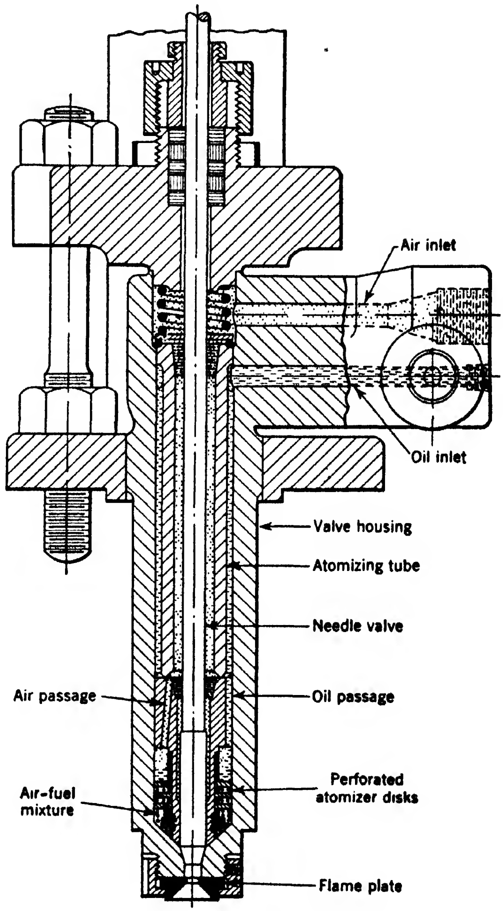


Fig. 99. Nordberg air-injection fuel nozzle.

Figure 99 shows a mechanically operated air-injection *fuel nozzle* of the *closed type.* The nozzle housing fits into an opening in the cylinder head; the water is sealed off by a thin bushing. Within this housing is placed a long atomizing tube which carries a series of perforated disks near the lower end. The fuel oil flows down the clearance space between the housing and the atomizing tube, whereas the air flows between the atomizing tube and the needle valve, as indicated in Fig. 99.

Thus the air and oil are forced down into the same concentric area just above the perforated atomizer disks. The fuel valve is constantly exposed to the high pressure of the injection air, whereas the fuel is deposited above the disks just slightly before the needle valve opens.

At the proper instant of the engine's cycle the needle valve, Fig. 99, is mechanically opened (raised) by means of a rocker-arm connection to the camshaft. The high-pressure air then rushes toward the cylinder, carrying the oil charge with it. The oil is forced through the small openings in the three atomizer disks and subdivided into successively finer particles as it moves downward to the conical space just below the disks. The air-fuel mixture is injected into the combustion space through a central orifice contained in the flame plate. The flame plate carries one central opening with flaring exit walls, thus enabling the hollow cone of atomized oil, formed by the flow along the conical end of the needle valve, to spread throughout the combustion space.

Fuel pumps for air-injection Diesel engines must deliver oil to the fuel nozzles at not more than 1000 lb per sq in., whereas fuel pumps for mechanical-injection engines may be required to deliver oil at pressures of 4000 to 6000 lb per sq in. During the power stroke of the engine, the air-injection fuel pump requires a longer time to inject a given amount of fuel than does the mechanical-injection pump. In general, the air-injection fuel pump is simpler and easier to build; however, it must not leak and must measure the fuel accurately before it is forced into the fuel nozzle. For varying loads on a Diesel engine, the speed is kept constant by the correlated action of the governor and the fuel pump by one of the following methods: changing the stroke of the pump plunger, holding open the suction valve during the first part of the discharge stroke, or by-passing part of the oil handled by the pump back into the suction line during part of the discharge stroke.

The use of air injection insures adaptability to various grades of Diesel fuel oils; efficient turbulence is created by injecting the fuel oil gradually with a large amount of high pressure air. The expanding of the air-fuel mixture from its high pressure (about 1000 lb) to the existing cylinder pressure (about 500 lb) also aids turbulence and assures complete combustion. This excellent atomization, distribution, and combustion of the fuel charge results in high mean effective pressures and good fuel economy for air-injection Diesel engines. Disadvantages include complication of the engine by a direct-connected multi-stage air compressor which uses a considerable amount of developed power and often is a source of trouble. Recent improvements in mechanical-injection Diesel engines have, in general, made them simpler and more efficient than the air-injection type.

76. Mechanical (Solid) Injection of Diesel Fuels. The first satisfactory mechanical-injection, also called solid-injection, system for supplying fuel oil to a Diesel engine was used in the year 1914. The types of mechanical-injection systems may be classified as follows: precombustion-chamber, constant-pressure (also called monorail or common-rail), and individual-pump (also called jerk-pump). Mechanical-injection fuel nozzles are of either the open or the closed type. A nozzle is of the open type if no means are provided within or near the nozzle to stop the flow of the fuel. A nozzle is called closed if it contains a spring-loaded valve near or at the orifice. With the open type, the injection is controlled entirely by the fuel pump, whereas the closed type is controlled mechanically by cam action or hydraulically by the fuel pressure.

When *precombustion-chamber* injection is used, the compression space is subdivided into two parts. One part is formed by the space between the top of the piston and the bottom of the cylinder head; the other, formed by a special recess in the cylinder head, is connected with the main cylinder space by a restricted passage; see Fig. 68. The fuel is injected into the precombustion chamber during the compression stroke; the smaller the passage between the precombustion chamber and cylinder, the earlier the injection. In this way the fuel is given time to vaporize before it ignites in the precombustion chamber, as explained on pages 164 and 165. Because of the insufficient amount of air, only partial combustion occurs in the precombustion chamber; but the pressure rises and forces out (injects) the remainder of the fuel into the cylinder, where combustion is completed.

In precombustion Diesel engines, the pump-injection pressures seldom exceed 1000 lb per sq in. The advantages of the precombustion system of fuel injection are that timing of injection is not required to be precise; fuel charge is given longer time to be prepared for combustion; and efficient turbulence is created by the precombustion charge. These features are very beneficial in variable- and high-speed engines, as explained in Art. 61. Disadvantages of precombustion chambers are: poor combustion control, probable high-explosion pressures, and unsuitability for large cylinder sizes.

Some years ago, when it became desirable to get away from the use of high-pressure air for fuel injection, the *constant-pressure,* or what is frequently called the common-rail, system of mechanical injection was developed, as shown in Fig. 100. This system was quite logical, since it resembled the air-injection system in that it employed a constant-pressure fuel manifold or header. In one case the header carries high-pressure air; in the other it is of much smaller volume and carries fuel oil under a constant high pressure.

The *constant-pressure* or common-rail system of fuel injection is shown schematically by the drawing on the left in Fig. 100. *A* represents a fuel pump which is driven by the engine but need not be timed. *B* is a high-pressure tube which serves for carrying fuel to all cylinders and *C* represents the fuel-injection nozzle. Governing is accomplished by means of timing this fuel-spray valve. The drawing on the right in Fig. 100 shows the general setup for intermittent injection, jerk-pump injection, or *individual-pump* injection as it is variously termed. In this sketch, *A* represents a quick-acting pump and *B* the injection nozzle. It is necessary to have one pump per cylinder whether these pumps are grouped or distributed along the engine. The high velocity necessary for atomization is derived from the very high injection pressure

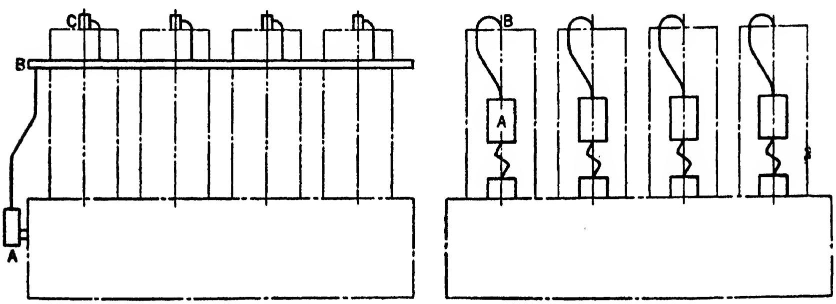


FIG. 100. Schematic arrangements of Diesel mechanical-injection fuel systems: left, constant-pressure (common-rail) injection system; right, individual pump (jerk-pump) injection system.

through the small orifices in the nozzle during the time this pump delivers its measured amount of fuel. Governing is by means of limiting the filling of the pump or by opening a by-pass valve at some point in the pump stroke to make the remainder of the stroke ineffective.

With the common-rail system the pump speed and phasing have no definite relationship to that of the engine crankshaft apart from the necessity of maintaining the required pressure in the fuel system for all conditions of load. It is in fact unnecessary for the fuel pump to be driven by the engine it is designed to feed. The fuel is admitted to the engine through a mechanically operated fuel nozzle, like Fig. 101, which is operated and suitably timed from the engine camshaft, the quantity of fuel being controlled by the lift, area of holes, and time of opening of the fuel-spray valve and the pressure in the accumulator pipe.

An advantage of the common-rail system is its simplicity; the proper injection pressure is available at the fuel-spray valve throughout the entire injection period. This insures constant injection conditions at all times, and produces regular and efficient combustion. The disadvantages are that very accurate construction and adjustment are required in order that the control of the fuel-valve lift may be exact and the load equally divided between the cylinders, and that the pressure is on the fuel valve at all times. A leaky fuel-injection valve produces dribbling or even injection at other than the correct time; this might prove a serious matter. Dribbling may be eliminated by interposing a control valve between the common rail and the fuel nozzle, thus relieving the nozzle of all pressure except during the injection period. This method substitutes an open nozzle in the cylinder head for the mechanically operated fuel valve. A check valve prevents the compression pressure in the cylinder from backing up into the fuel line. The common-rail system is not well adapted to speeds above 800 rpm because of the heavy impact of the valve gear on its cam and the difficulties of accurate control.

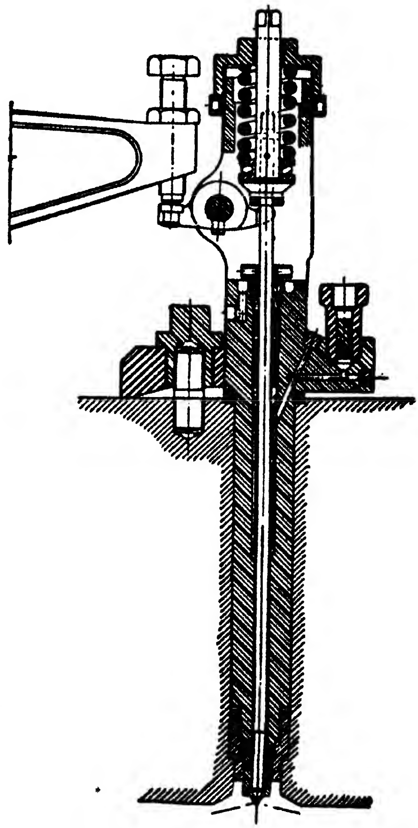


Fig. 101. Winton injection nozzle for common-rail fuel system.

The mechanically operated (closed) type of fuel-injection nozzle is never used except with the common-rail fuel system (Winton four-cycle and Cooper-Bessemer) and distributor system (Cummins and Ingersoll-Rand). A typical mechanically operated nozzle is shown in Fig. 101. It is spring-loaded to hold it tightly on its seat against the compression pressure of the cylinder and the fuel pressure. This necessarily makes the spring load rather a heavy one, which in turn puts a heavy duty on the valve-operating mechanism; nevertheless the control must be very accurate because the lift is small. The problem is accentuated when small engine cylinders are used.

The distributor system of fuel injection is used on the Cummins Diesel engine. Figures 86 and 87 show sectional views of the engine and Fig. 102 shows the general layout of the fuel metering and delivery to the mechanically operated injection nozzle. All the important parts have been suitably annotated; the distributor shown on the extreme right side is for a six-cylinder engine.

The Cummins fuel-injection nozzle consists of a cast body with an accurately fitting plunger; the latter is operated by a rocker arm and push rod working from the camshaft located in the cylinder block. Mounted on the lower end of the injector body and seating in the cylinder head is the fuel cup and adapter. A line from the fuel pump forms a fuel passage from the metering plunger in the fuel pump and distributor to the fuel hole running through the injector body. A groove is turned in the adapter face seating on the lower end of the injector body which connects with the bottom of the fuel passage. A series of

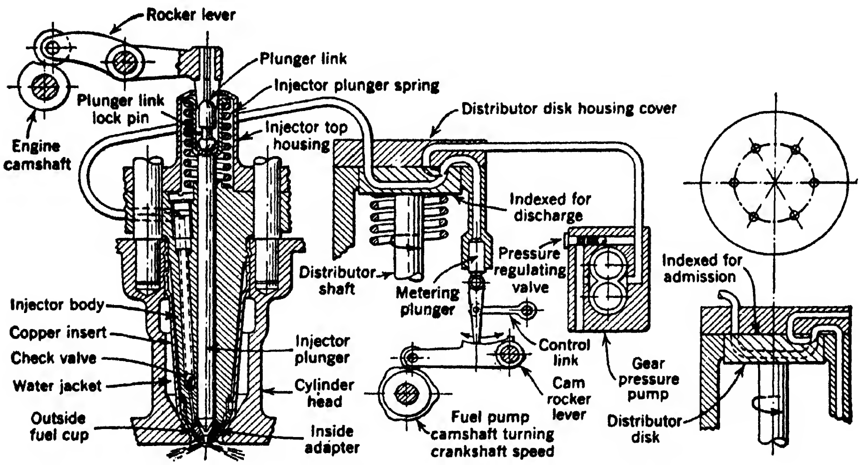


FIG. 102. Diagram of Cummins distributor-type low-pressure fuel-injection system.

small holes leads from this groove in the adapter to the annular space between the cup and adapter. Another series of holes leads from this annular space to the chamber under the injector plunger. The action of the Cummins fuel-injection nozzle in delivering fuel to the cylinder and the subsequent combustion are described in the last paragraph of Art. 61.

The Cummins injection system employs a single-plunger fuel pump for measuring and delivering the fuel at relatively low pressures, about 100 lb per sq in. to all the cylinders, irrespective of the number of cylinders and regardless of the setting or adjustment of the fuel pump. The fuel is distributed by a rotary type distributor disk to each of the cylinders in turn. The single-plunger pump is subject to a variable stroke, the latter being controlled by hand throttle or governor; the maximum stroke is such that no more fuel is delivered than can be burned in the cylinder with the available air charge. Valves are eliminated in this type of pump because of the distributor disk with proper ports which index as they revolve in correct time with the suction and discharge of the metering plunger.

In order to operate satisfactorily at high speed, it is necessary to deliver the fuel to the Cummins pumping and distributing mechanism under a certain pressure. A small gear pump with pressure regulator takes the fuel from the main tanks and delivers it at 90 to 120 lb per sq in. pressure to the plunger chamber. In this way the use of high fuel pressures has been avoided. The advantages of this method of fuel delivery and injection are that it is a "valveless" system, exactly the same amount of fuel being delivered to each and every cylinder, and that low fuel pressures are used on the delivery side of the system.

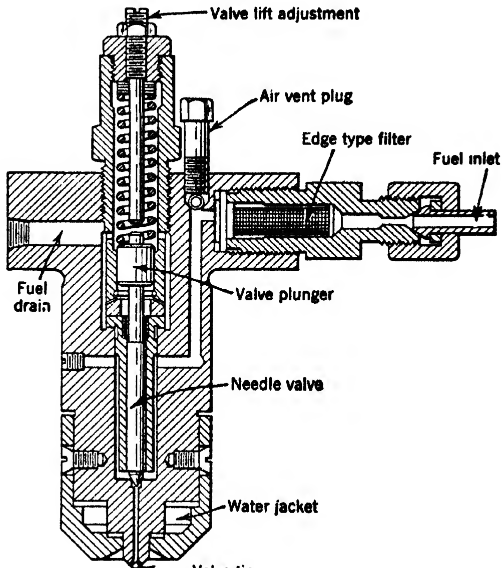


FIG. 103. Fairbanks-Morse closed-type fuel-injection nozzle.

The fuel-injection nozzle shown in Fig. 103 is of the closed type. It has a spring-loaded needle valve which is hydraulically opened by the fuel pressure and an *individual fuel pump* is required for each nozzle. Through drilled passages in the nozzle body and passages formed by external fluting on the bushing the fuel reaches the valve tip, the passage through which is closed by the needle valve. When the pressure at this point reaches a predetermined value the needle is lifted off its seat against spring compression and fuel is forced through the passage in the tip and into the cylinder through multiple orifices. As soon as the needle valve opens, the fluid pressure immediately drops and the needle valve reseats itself, thus cutting off the flow of fuel to the cylinder. The reader should note the water-cooling space in the lower part of the nozzle body around the valve tip.

The Bosch fuel-injection system, Fig. 104, is used by a large number of manufacturers of mechanical-injection Diesel engines, because of its simplicity and reliable performance. Between the pump and the fuel-

…on nozzle is the connecting piping. The best arrangement is to …ve the pump and nozzle adjacent so that the piping becomes a short passage, but it is not always convenient to do so. In small engines especially, it is usual to group the injection pumps with their drive into one unit; see Fig. 112. When a single pump with distributor is used, the individual fuel piping may also become relatively long. The pump must compress all the fuel in the connecting pipes to the injection pressure before injection can begin. Frequently a moderate pressure is

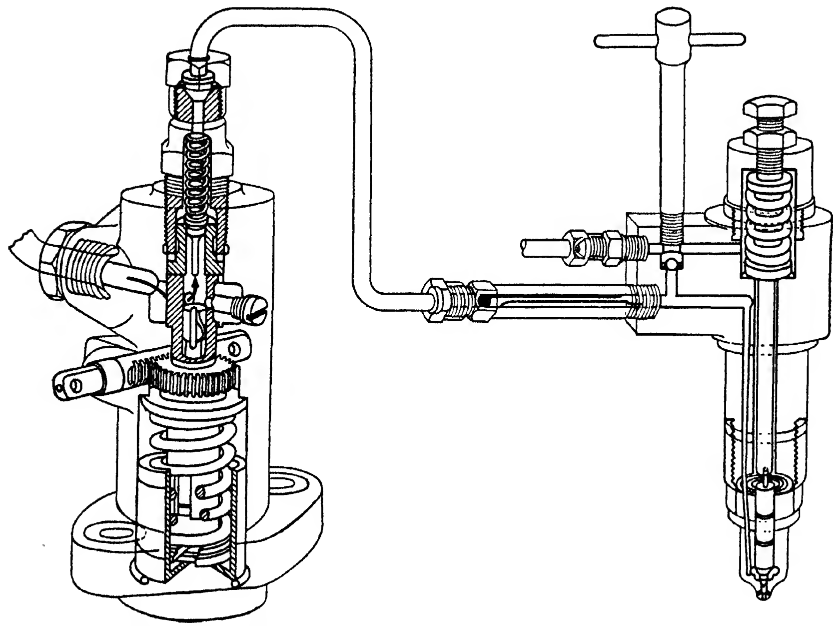


FIG. 104. Essential parts of Bosch fuel-injection system, multiple-hole nozzle.

retained on the fuel in the piping so as to reduce the time required to build up the injection pressure. The fuel is compressible; hence the volume must be a minimum and the piping connection should be as short and the bore as small as possible, limited only by the ability to withstand the high injection pressure with a minimum of expansion of the tubing.

In pump-injection systems the fuel nozzle is usually operated *hydraulically*. The pump controls the exact amount of fuel and the timing of injection, as explained in Art. 79. A Bosch fuel-injection nozzle is shown, sectionally, in Fig. 105. Another style is shown in Fig. 104. The nozzle tip may be of either the multiple-hole or pintle type; see details in Fig. 106. For direct-injection Diesel engines having flat, tapered, or combustion space of other type, with or without whirling of the charge, introduction of the fuel through one hole is inadequate. It is necessary to use the multiple-hole nozzle tip to distribute the fuel uniformly in the combustion space. Nozzle tips of pintle type are fitted with valves whose ends are furnished with a thin shank or pin, the shape of which is made according to the spray angle desired. The pin extends into the orifice nozzle so that it forms an annular space. By suitably shaping this pin we can form either a hollow cylindrical jet of high power of penetration or a tapered spray with an angle varying from a few degrees to about 60 degrees. The wider the angle, the better the atomization; but there is a sacrifice of penetration.

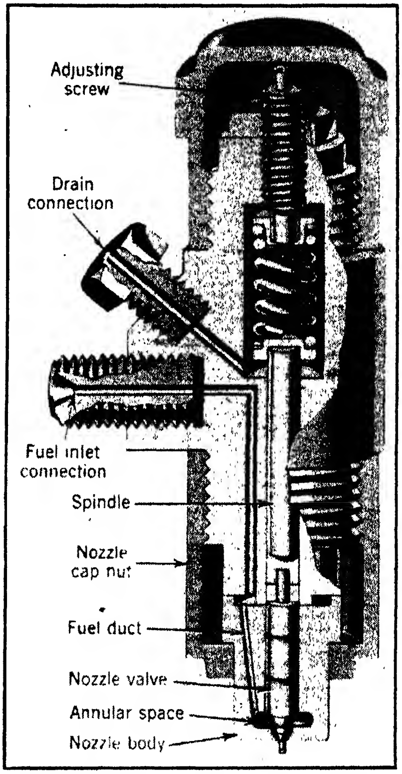


FIG. 105. Bosch closed-type fuel-injection nozzle.

It will be seen, therefore, that the pintle nozzle permits fine variations in the characteristics of the fuel spray; for instance, the cross section of the orifice can be opened gradually during the lifting of the valve stem if the pin is tapered, or in steps if the pin is made in two cylindrical steps. In this way only a small quantity of fuel is injected at first, followed by the greater part later. This procedure prevents a sudden pressure increase in the cylinder and gives smoothness to the running of the engine. The double-throttling effect of the nozzle, first at the valve seat and then at the nozzle orifice, is responsible for the fact that the pin nozzle functions uniformly and accurately. In addition, the motion of the pin in the nozzle orifice prevents the formation of a carbon deposit.

The Bosch fuel-injection nozzle, Figs. 104 and 105, is hydraulically operated. The fuel delivered by the fuel-injection pump reaches the injection nozzle through the fuel inlet connection and the fuel ducts of the nozzle holder, the nozzle being held in place against the highly ground and lapped surface at the shank end of the nozzle holder by means of a cap nut. In the center bore of the nozzle body, the hydraulically operated nozzle valve is placed and spring-loaded as shown. In order to vary the opening pressure of the nozzle, the tension of this spring may be changed by means of an adjusting screw. The small quantities of fuel leaking past the nozzle valve rise through the clearance space around the spindle, leave the holder through the drain connection, and are returned to the fuel supply tank. When the pump on its delivery stroke builds up sufficient pressure on the oil in the space around the bottom of the nozzle valve to overcome the tension of the valve spring, the nozzle valve is forced upward and the fuel is sprayed directly into the cylinder.

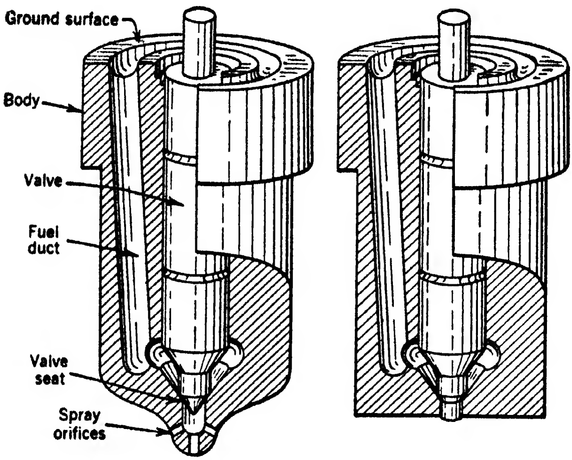


FIG. 106. Multiple-hole and pintle-type nozzle tips for Bosch fuel-injection nozzle.

The Hesselman low-pressure fuel-injection nozzle of open type, Fig. 107, employs a triple-ball spring-loaded check valve to cut off the fuel spray sharply when injection ends, and to prevent combustion pressures from backing up into the fuel pressure lines. The nozzle tip, made of nitralloy stainless steel, has two spray holes (0.015 to 0.025 in.),

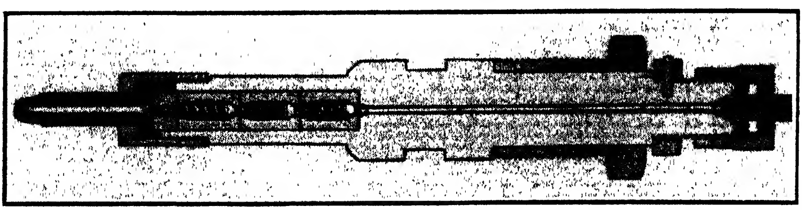


FIG. 107. Hesselman open-type low-pressure mechanical-injection fuel nozzle.

one pointing (about 30 degrees from center line) towards the whirling current of turbulent air in the cylinder and the other pointing in the opposite direction. The size of the holes and the angles for each nozzle depend upon the engine model. This type of valve is used with low-compression spark-ignition oil engines of high speed, as explained on page 153. The fuel is forced down twin grooves in the nozzle tip insert; the ends of the grooves flare at 30 degrees to direct the fuel toward each spray hole.

The springs in the Hesselman fuel-injection valve of smaller engines are adjusted to open with an oil pressure of 650 lb per sq in. whereas the larger engines require a pressure of 1000 lb per sq in. This is considered a low injection pressure, possible only in engines which have a high air turbulence in the combustion chamber to accomplish a thorough mixing of fuel and air. Moderate injection pressures are desirable because they increase the life of the injection equipment just as the use of moderate compression and combustion pressures increase the life of the engine itself.

77. Design Features of Fuel Injectors for Diesel Engines. The valve in a closed injection nozzle is sometimes operated mechanically, but it is thought that this type of valve will ultimately disappear, the automatic hydraulically operated valve being much simpler and having proved to be entirely reliable.

The most important requirement of a fuel-injection nozzle is distribution of the fuel over the induction air. The process of distribution can be divided into two functions, penetration and dispersion. The dispersion function can again be divided into atomization and spreading. The purpose of an injection nozzle is therefore more explicitly to give the penetration, atomization, and spreading of the fuel most appropriate for a certain type of engine, as discussed in Art. 70. It is also required that the beginning and end of the injection period be sharply defined and that the proper distribution continue over the full period of injection.

By penetration is generally meant the distance which the end of the fuel jet travels in the compressed air inside the cylinder. It is determined by the pressure on the fuel, the pressure and temperature of the compressed air in the cylinder, and the nature of the hole through which the injection takes place. It is natural, though frequently erroneous, to think that increased fuel pressure materially enhances the penetration as defined above.

The velocity of the fuel at the orifice increases with the square root of the fuel pressure, but at the same time the resistance in the air increases with the nth power of the velocity, n being a figure between 1 and 2. The influence of the higher pressure is therefore almost counterbalanced by the greater resistance of the air; and increasing the pressure proves an inefficient means of getting better penetration, although it aids atomization. Experiments on penetration have shown that doubling the injection pressure gave an increase of penetration of only about 17 per cent. The penetration changes with the air density, but by no means proportionally. As the compression pressure is generally determined by other desired Diesel engine features—ignition quality of the fuel, for instance—varying this pressure cannot be used as an effective means to influence the penetration.

The nature of the hole through which fuel injection takes place has a considerable amount of influence on the penetration and on the character of the injection, i.e., the distribution generally. It is primarily the diameter of the hole which matters; the length of the hole is of less importance. Up to a certain ratio between diameter and length of hole (about 1 to 4) penetration increases, but proportionally longer holes do not give any better penetration. A jet ejected from a hole of larger diameter, however, will travel farther. The fuel particles in the center of the jet meet with less resistance; they are of larger size, and consequently have more momentum and encounter less specific resistance. This implies that the atomization is impeded. That the atomization is impeded is generally of minor consequence, however, as the highly penetrating jet is usually required on engines of large size, where, on account of the slower speed and the ampleness of the time available for the combustion, a very fine atomization is not so essential.

Experimenters have shown what happens when the fuel is injected into air at rest. The whole picture will be quite different when the air is in motion and when the fuel is ignited after entering an engine cylinder. To study the behavior of the jet under actual conditions is far more difficult but not impossible. A recent article by Dr. Otto Holfelder [1] gives some excellent photographs of the fuel spray, its ignition and combustion under various conditions. These photographs, taken at a rate of more than 400 per second, show the shape and development of the jet, the time at which the spray ignited, and how the combustion proceeds as the fuel is injected. These results were the first, it is claimed, which can be accepted as comparable to actual operating conditions.

It is most interesting to study what happens in the fuel injector and how the fuel behaves after it leaves the nozzle. It is a curious fact, however, that, although tremendous changes in views regarding these points have occurred, there have been very few recent changes in the design of injectors. This is true probably because the theory has lagged behind practice and the best designs had already been determined experimentally, by trial and error. The best size and shape of the holes in the nozzle have been found on the engine by results as judged from fuel economy, visibility of the exhaust gases, and torque.

The total area a of the holes in a fuel-injection nozzle can be determined for an engine with a cylinder volume of V to develop the horse-

[1] Zündung and Flammenbildung bei der Diesel-Brennstoff-Einspritzung," O. Holfelder, *Forschungsheft* (supplement to *Forschung aus dem Gebiete des Ingenieurwesens*), September–October, 1935.

power, hp, at N revolutions per minute and a fuel consumption of w per working stroke. If t be the time taken for the injection of the fuel, v the velocity of the fuel as caused by the initial injection pressure, and c the coefficient of discharge,

$$a = \frac{w}{cvt} \tag{66}$$

The product of the velocity v and the discharge coefficient c is the apparent nozzle exit velocity v_e, and if the initial injection pressure be assumed constant for this particular class of engine, v_e will also have a constant value. Hence equation 66 can be written

$$a = \text{const}\,\frac{w}{t} \tag{67}$$

The fuel is injected over a certain crank angle of x degrees, and the time taken by the injection is

$$t = \frac{x}{360} \cdot \frac{60}{N} = \frac{x}{6N}$$

This angle x has an approximately constant value at full load for engines of one class, therefore

$$t = \text{const}\,\frac{1}{N} \tag{68}$$

If the fuel consumption per effective hp for engines of this class at full load is constant, the quantity of fuel w is proportional to the cylinder volume V.

$$w = \text{const}\ V$$

and as under these same conditions

$$hp = \text{const}\ VN$$

it follows that

$$hp = \text{const}\ wN$$

or

$$w = \text{const}\,\frac{hp}{N} \tag{69}$$

If the values for w and t from equations 68 and 69 are introduced in equation 67, it is found that the speed N disappears and

$$a = \text{const}\ hp \tag{70}$$

This simple equation shows that for a certain class of engines the total area of the holes in the fuel nozzle is directly proportional to the horsepower developed, independent of the speed. How many holes should be used and how these should be arranged is a matter of experiment and experience. Here the volume and shape of the combustion chamber are the principal factors to be considered, as explained in Chapter IX.

As a rule, a certain amount of compromise is necessary in order to get the required total area and the correct angle at which the holes are drilled in the nozzle; the final adjustments must be made during actual engine tests because the shape of the combustion chamber and the volume change during the injection of the fuel. *A* in Fig. 97 shows the combustion chamber of a large Diesel engine of the four-cycle supercharged type of 500 bhp per cylinder. When the first engine of this type was tested several nozzles were tried out and it was found that an 8-hole nozzle was the only suitable one giving smokeless exhaust. With a 12-hole nozzle of equal total area the engine could not be made smokeless, thus proving that the cones were too close together and too short. The form of the combustion chamber should be very closely adapted to the shape of the fuel jets, as shown at *B* in Fig. 97.

Good combustion is much easier to achieve with high-speed engines if the air in the compression space is in motion during the injection of the fuel. The distribution of the fuel must take place in an exceedingly short time, of the order of one-thousandth of a second, and this is made considerably easier if the air is being swept past the fuel orifice. The proper distribution is obtained by bringing the air to the fuel instead of bringing the fuel to the air. The method of bringing the fuel to the air necessitates injectors with small holes, the size, number, and direction of which must be adapted to a certain most advantageous shape of the combustion chamber. Given a combustion chamber which causes an air swirl to be developed, the adaptation of the nozzle to the circumstances is much easier. The type of nozzle used is the pintle type, see Fig. 105, which gives a tubular jet, straight or conical as is best suited to the engine. These pintle nozzles have the advantage of functioning at different speeds with constant pressure as the needle valve carrying the pintle will lift higher and the area of injection will be increased at higher speeds, and vice versa. A number of successful methods have been used to produce this air swirl, as illustrated and explained in Chapters VIII and IX.

It is important that the amount of fuel in the injection nozzle be as small as possible, which is essential for that part of the fuel entrapped below the valve seat. This fuel can eventually drip out and may carbonize or cause after-burning. The valve should be small and have a low lift, and yet resistance of the fuel in the valve should be kept at a reasonable figure. There should be no air pockets in the injector. The entire apparatus should be sufficiently stiff and of symmetrical shape, so that there can be no distortion by heat or other stresses which can cause misalignment of seat and valve guide. These should, therefore, be very close together, or the end of the needle valve should be so elastic that it will adapt itself to the conical valve seat. Sometimes the valve seat is made flat instead of conical, in order to counteract the danger of misalignment of seat and guide.

The cooling of the injection nozzle is an extremely important feature, and the injector should be so designed that the cooling is efficient, down to the parts which protrude into the cylinder and are subjected to the heat of combustion. If the nozzle is insufficiently cooled the drops of fuel which are formed after the closing of the valve will carbonize. This will impair the development of the proper jet, will mar the distribution of the fuel, and eventually will clog up the holes. Some, otherwise excellent, Diesel fuels have properties which make cooling of the nozzle imperative. If kept below a certain temperature—say below 300 F—the fuel will not carbonize, but will form a soft deposit which is easily blown off by the spray; but if the nozzle is too hot, carbon deposits will form in the shape of hollow cones, which, hindering atomization and spreading, keep the air from reaching the fuel particles at the proper place and time, leading in time to smoky exhaust.

The three media which come into consideration for cooling fuel injectors are water, lubricating oil, and fuel. Each has its disadvantages and it is difficult to give a general rule for the use of any of them. Cooling with water has frequently a curious erosion effect on the material of the injector, the cause of which is hard to detect. When oil or fuel is used, no corroding or eroding action is present; but both oil and fuel have other disadvantages. With oil, a leaky injector may cause contamination of the lubricating oil with fuel, which may become dangerous if not detected. With fuel, a leak in the cooling space or in the seating of the injector may allow fuel to enter into the cylinder during the suction or scavenging stroke, and this may eventually cause backfiring.

A well-designed fuel-injection nozzle which embodies the various requirements enumerated in this article is a simple apparatus. But in order to function satisfactorily it should be made with the utmost precision, and of the best and most suitable materials. It lends itself well to normalization, and it is therefore natural that the manufacture of injectors is being done by specialists, who manufacture them in large numbers and at comparatively low prices. Many builders of Diesel engines are now using these normal types of injectors.

78. The Unit Fuel Injector. The unit injector combines all the elements required for fuel injection into one unit, Fig. 108; one of these individual injector units is fitted to each cylinder of the Diesel engine. There are no long high-pressure pipe lines between the fuel pumps and the injection nozzle. As shown in Fig. 108, fuel, supplied at a pressure of about 20 lb per sq in., enters near the top at the right-hand side of the body. After passing through the filter in the supply line, the fuel fills the annular supply chamber around the bushing. Surplus fuel supplied flows out through the overflow passage on the left side. The fuel pressure in the injector is maintained by discharging the overflow

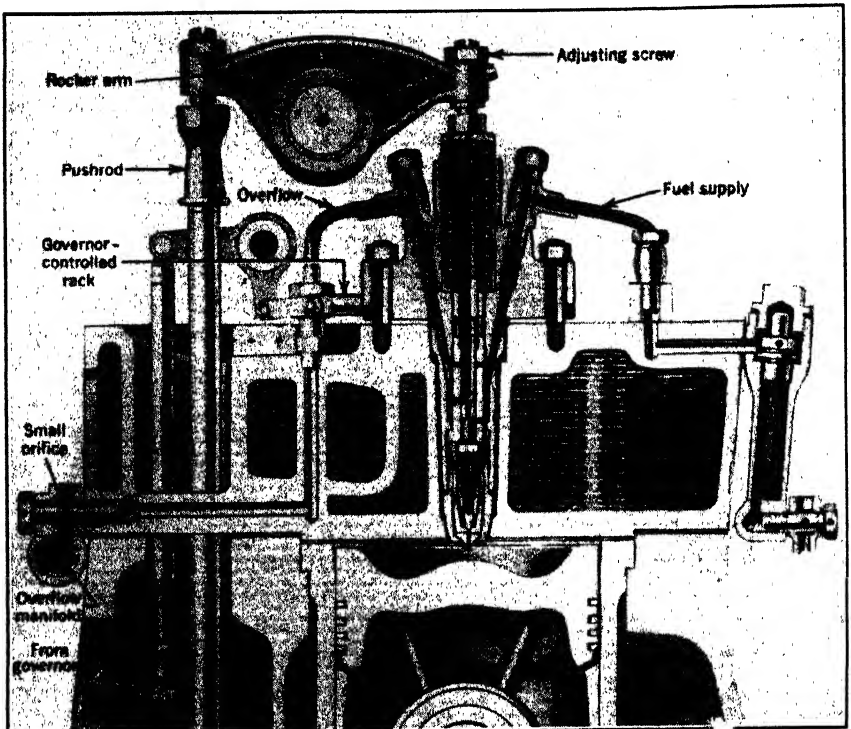


FIG. 108. Section through unit fuel injector, cylinder head, and piston of Winton two-stroke cycle engine.

fuel through a small orifice in a restriction plug, which sets up the required resistance.

As the plunger, Fig. 108, is moved downward by the rocker arm (cam and push rod actuated), fuel in the bushing chamber is first displaced through the lower port into the supply chamber until the lower edge of the plunger closes this port. Fuel is then displaced through the central and transverse holes in the plunger and through the upper port into the supply chamber. Continued downward movement of the plunger causes the helical upper lip to cover the upper port, and the fuel in the bushing chamber is then forced down through the passages in the spring cage and the needle guide to the annular space around the spring-loaded needle valve. Acting on the stepped area of the needle valve, the fuel pressure builds up (above 10,000 lb) until it lifts the needle valve from its seat, Fig. 109, when the fuel passes into the spray tip and is forced through the orifices into the cylinder.

Injection continues until the lower lip on the plunger uncovers the lower port in the bushing, and then the fuel by-passes up through the holes in the plunger and through the lower port into the supply chamber. This releases the pressure of the fuel in the bushing chamber, and the needle valve spring closes the valve. On the return stroke, the upward movement of the plunger fills the bushing chamber with fuel oil which flows from the supply chamber through the lower port.

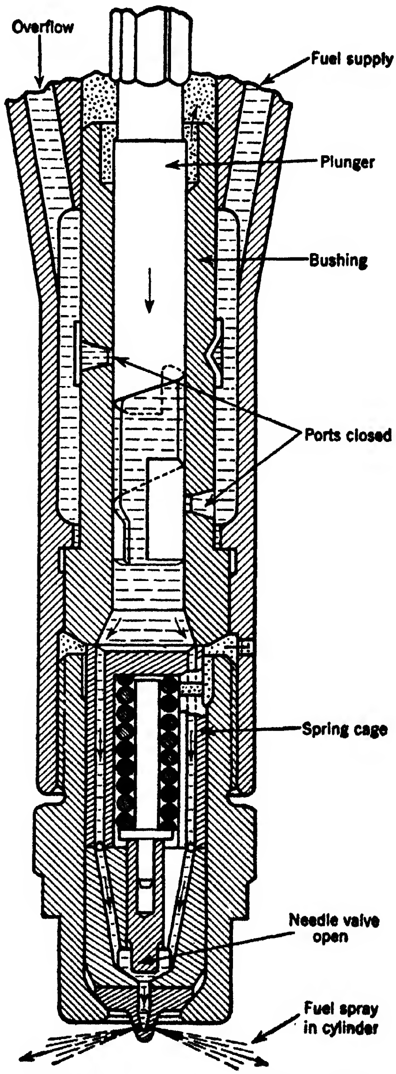


Fig. 109. Pumping period of Winton unit fuel injector.

The plunger has two motions, reciprocating motion to pump the fuel and a rotary motion to vary the amount of fuel according to the load. In Fig. 110, the upper diagrams show the stroke positions and the lower diagrams the rotary positions. The upper series of diagrams show that the total downward stroke of the plunger is divided into three parts. The first or upper part allows acceleration to the required plunger velocity before pumping begins. The second or pumping part of the stroke controls the quantity of fuel injected. The third or over-travel part slows down the plunger velocity to the end of the stroke. The lower diagrams in Fig. 110 show how the distance from the lower lip to the lower port is varied by rotating the plunger, with the resulting variation in the effective pumping stroke.

A gear on the reciprocating plunger and a rack connected to the governor, as indicated on Fig. 108, rotate the plunger and control the fuel supply.

Figures 109 and 110 show that the pumping part of the stroke begins when the helical upper lip on the plunger covers the lower edge of the upper port. Rotating the plunger, so as to increase the effective pumping stroke, also makes the effective pumping stroke start earlier during the cycle. When the load is increased, the plunger rotation automatically advances the beginning of injection.

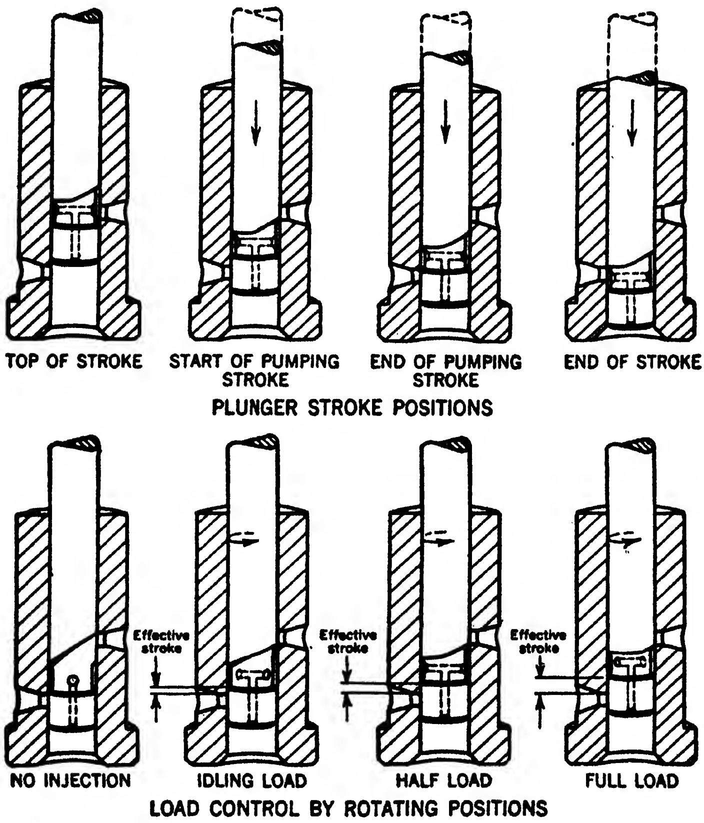


FIG. 110. Bushing of Winton unit fuel injector with various plunger positions.

The spring-loaded needle valve performs three important functions in the operation of the unit injector. First, it serves as a check valve between the combustion chamber and the injector. Second, it gives positive control for the start and stop of the fuel spray into the cylinder. Third, it provides the necessary atomization of the fuel at low speeds.

Atomization is a function of the relative velocity between the fuel and air, i.e., a function of the velocity of the fuel through the orifices. Since the needle cannot open until the fuel pressure has been built up high enough to compress the spring, it follows that the initial fuel spray velocity is always sufficient for good atomization, even at low speed and light load. This fine atomization of the first part of the spray insures prompt ignition of the fuel, so that the remainder of the fuel spray burns as it enters the cylinder. Thus, the combustion pressure can be controlled by the shape of the cam which operates the injector plunger.

79. Fuel-injection Pumps. The pump is probably the most important unit of the fuel-injection system for it must be designed to function for long periods without wear or leakage under very high oil pressures. Moreover, it must deliver an exceedingly small metered quantity of fuel at each stroke and at precisely the correct moment in relation to the piston's movement on the compression stroke. Thus in an engine running at 2000 rpm, the injection period of only 20 to 30 degrees takes place in about 0.002 sec. The quantity of fuel injected per stroke for a 50-hp cylinder is of the order of 0.004 to 0.05 cu in. at normal loads and 0.001 to 0.012 cu in. at light loads. There should be a fairly sharp beginning of the fuel injection, a constant or (preferably) moderately rising pressure of the fuel during injection, and a very sharp ending of the fuel injection. The pressures used for pump injection range from as low as 800 to as high as 10,000 lb per sq in.

Precise metering of the fuel must be maintained by the pump for variations in load, speed, and reasonable changes in viscosity of the fuel. It is important to have each injection pipe of the same length and size, so far as possible, in multi-cylinder engines; otherwise the values of the pressures and the nature of their variations will differ in each pipe with the result that the injection pressures and volumes will also differ. There should be sufficient pressure on the pump suction to insure filling on each stroke.

The ideal pressure requirements for the fuel-pump system are a low initial rate of injection with an increasing rate up to a maximum at the point when injection ceases. The rate of fuel injection is dependent upon the fuel cam design; the cam must be steep enough to accelerate fuel injection and should have an easy beginning if it is to operate quietly. The increase of pressure in the fuel at the beginning of the injection sets up pressure waves in the column of fuel in the fuel piping. The longer the piping and the greater the quantity of fuel contained, the more likely it is that these waves will become troublesome. The higher the speed and the smaller the engine, the more essential it is that the connection between the pump and the fuel-injection nozzle be reduced to a minimum.

Fuel pumps for mechanical-injection Diesel engines are controlled by one of three methods: variable-stroke pump plungers, governor-controlled suction valves, or governor-controlled by-pass valves. The fuel pump shown in Fig. 111 is the two-cylinder plunger type used with the Winton common-rail system of fuel injection. The pump forms an individual unit, and combines a built-in hand pump for high pressure

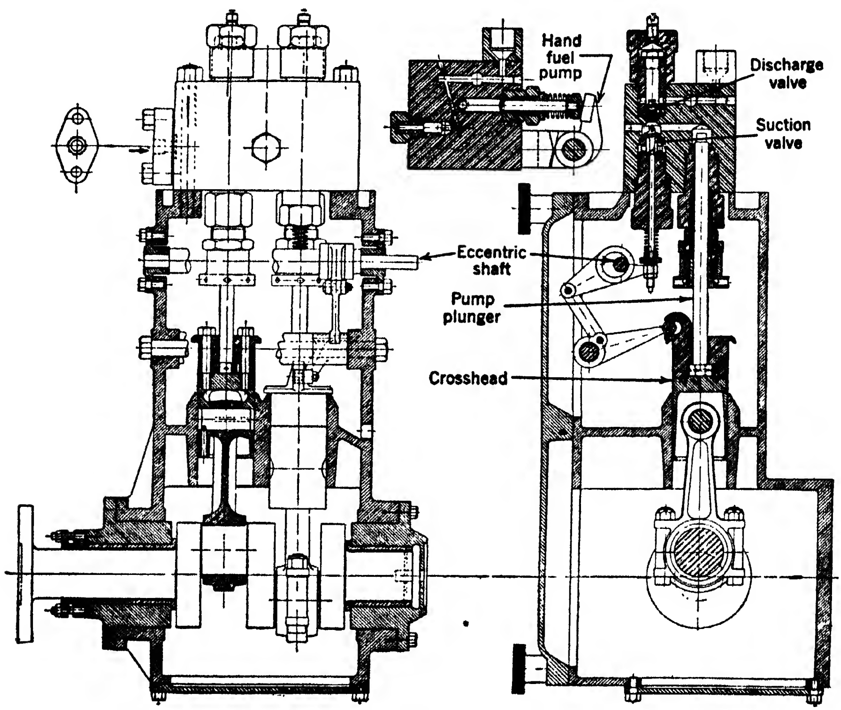


FIG. 111. Winton fuel pump for common-rail fuel injection; governor-controlled suction valve.

priming for starting the engine. The plungers, hardened alloy steel, are operated through crossheads from a crankshaft. Leakage past the plunger is prevented by the use of a special long packing box and adjustable gland. The steel block on top of the fuel-pump housing contains the plunger cylinder, suction and discharge valves, hand pump and safety valve.

Fuel is drawn into the plunger chamber through the suction valve and is discharged through the discharge valve and manifold into the common rail. The duration of opening of the suction valve is controlled by the governor through a system of links and levers operated from the plunger crosshead. The governor lever which is connected to the eccentric shaft regulates the amount of fuel discharged from the plunger chamber into the discharge manifold by increasing or decreasing the clearance between the top of an adjusting screw and the head near the lower end of the suction valve; see Fig. 111. As this fuel pump is of the constant displacement type, the excess fuel is by-passed back through the suction valve into the fuel inlet manifold before the suction valve is permitted to seat. The suction valve is so timed that the valve is open when the plunger is at the bottom of the suction stroke and may close at such a position of the discharge stroke of the plunger as is determined

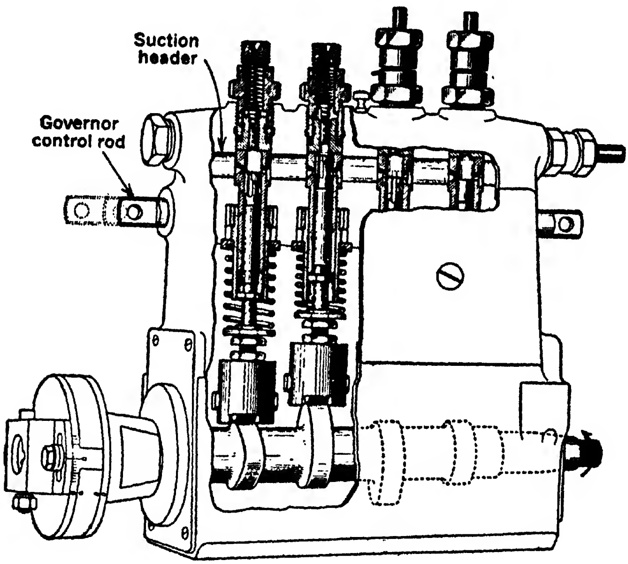


FIG. 112. Cutaway drawing of Bosch quadruple pump showing driving mechanism.

by the position of the governor control which regulates the amount of fuel discharged into the engine common rail.

The hand fuel pump, consisting of a spring return plunger actuated by a lever and reach rod from the control station on the engine, utilizes the suction and discharge valves of one of the pump cylinders and is used for hand priming the common rail when starting the engine. The hand pump plunger is made similar to a poppet valve with a seat on the underside of the head which seals against any possible leakage past the plunger when the engine is in operation. The block of the fuel pump discharge valve also carries a safety valve which is adjusted to a predetermined pressure ranging from 6000 to 10,000 lb per sq in. If the pressure in the discharge manifold exceeds the amount for which the safety valve is set, the excess fuel by-passes back into the suction manifold.

The Bosch fuel pump belongs to the single plunger per cylinder constant-stroke type. Figure 112 shows a cutaway drawing of a Bosch pump assembly for a four-cylinder engine. Each pump element, see details in Fig. 113, consists of a steel plunger which is a piston fit in the barrel. The plunger is provided with a suitable guide and tappet roller (Fig. 112) which rests on the cam shown. There is also a discharge valve fitted at the upper end of the barrel. Fuel supplied by gravity from a tank, above the pump, flows through a suitable filter to the fuel inlet connection and keeps the suction space in the pump casing always flooded with clean fuel, whence it can readily be drawn into the pump barrels of the various elements through the two small ports provided. The pump plunger moves vertically in the barrel with a constant stroke of 10 mm. The plunger diameters vary in different models from 5 to 10 mm; special single-cylinder units are also made with plunger diameters up to 20 mm and strokes to 30 mm. The 10-mm plunger with 10-mm stroke has a delivery of 0.0171 cu in. per stroke.

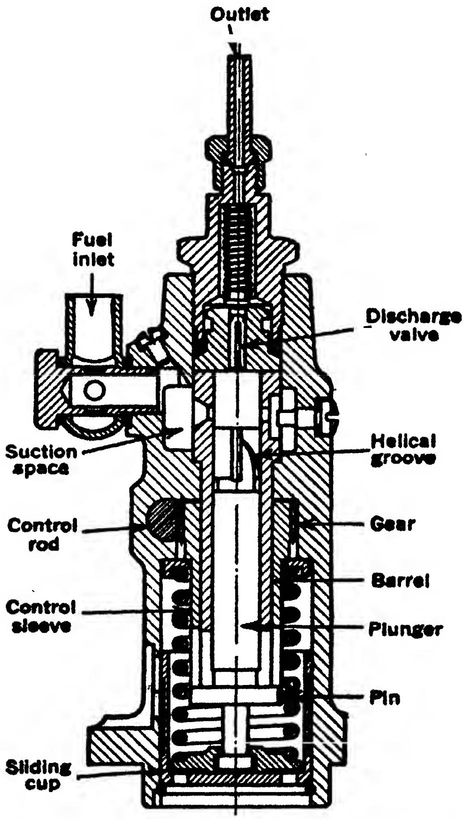


FIG. 113. Bosch individual fuel pump; plunger acts as bypass valve.

To enable the pump to vary the quantity of fuel delivered per stroke, the plunger is provided with a vertical channel, see enlarged view in Fig. 114, extending from its top edge A to an annular groove (the upper edge B of which is formed as a helix) partly down the plunger length. By means of the rack on the governor control rod and toothed control sleeve the plunger can be rotated slightly in its barrel during vertical movement. The pump element on the left in Fig. 112 is shown at the bottom of its suction stroke when the two small ports are open through which fuel has already been drawn, thus filling the pump barrel. On the next up or delivery stroke (second pump element from left) the plunger displaces fuel back through the two small ports until its top edge A (Fig. 114) covers them, so that the remaining fuel is forced out through the discharge valve to the fuel outlet. Thus it may be seen that since the plunger is of constant stroke its top edge will always cover the ports in the pump barrel at the same position of cam rotation, so that the beginning of fuel injection at the nozzle will always be the same, relative to the position of the engine crank.

The pump will continue to inject fuel at the nozzle as long as the ports are kept covered by the plunger, but reference to drawing 2 in Fig. 114 shows that before the plunger reaches the top of its stroke, the helical edge B of its annular groove uncovers the right-hand port. This enables the enclosed fuel to take the path of least resistance, from the vertical

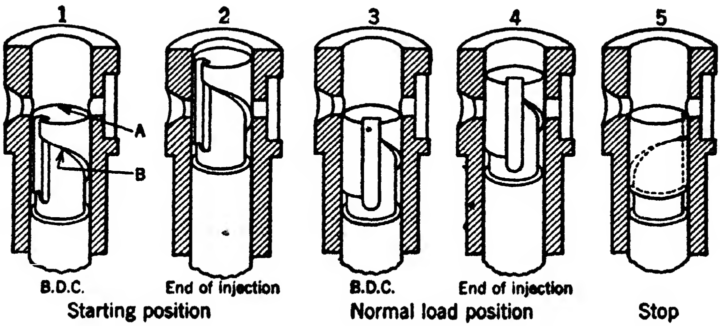


FIG. 114. Plunger of Bosch fuel pump in various positions.

channel and annular groove in the plunger, back through the port in the barrel to the suction space. The position of the plunger stroke, at which the helical edge B will uncover the port, is adjustable by rotating the plunger through a certain angle axially by means of the rack on the governor control rack and geared control sleeve shown in Figs. 112 and 113.

A gear surrounds the upper end of the control sleeve, Fig. 113; the lower end of the control sleeve is slotted to accommodate a pin, which is fastened to the plunger. The governor moves the control rack which turns the control sleeve, thus turning the plunger through the interconnecting pin. The amount of rotation controls the instant of pressure release by helical edge B, the amount of fuel delivered through the discharge valve, and the amount returned (by-passed) to the suction space.

When the helical edge B of the pump plunger uncovers the port in the pump barrel near the end of the delivery stroke, the pressure of the fuel is immediately reduced so that the discharge valve at once drops on its seat, thus cutting off communication between the pump and the fuel-injection nozzle at the engine cylinder. The discharge valve also performs another function, that of releasing the pressure in the fuel discharge piping. The valve in question is designed specially for this double purpose. It effects a certain increase in volume in the delivery pipe system, so as to reduce the pressure practically to atmosphere. The effect of this is to cause the fuel valve in the injection nozzle to close quickly, thus suddenly terminating the spray of fuel and eliminating after-dribble.

The selection of a suitable fuel-injection system for a Diesel engine depends upon the service required, the type and size of engine, and the design of the combustion chamber. Engines with quiescent or semi-turbulent combustion chambers generally use the common-rail fuel system with mechanically operated injection nozzles. Multi-hole spray tips are necessary to atomize and distribute the fuel properly. Pump-injection systems may be applied to all types and sizes of engines and the various forms of combustion chambers; however, some special system of pump injection is always associated with a certain type of combustion chamber and size of engine. Either the open or closed type of fuel nozzle may be used with the pump-injection system. The open nozzle is used where a relatively coarse spray is satisfactory, owing to the form of combustion chamber or the speed of the engine. The closed pintle-type nozzle is employed with a turbulent combustion chamber. The closed multi-hole-type nozzle is applicable to a combustion chamber which requires the fuel to be atomized and distributed as it is injected into the cylinder.

80. Regulating the Delivery from a Diesel Fuel Pump. The amount of fuel injected into each cylinder per working stroke governs the power output from each cylinder, the maximum power being obtained when the quantity of fuel injected is sufficient to utilize the maximum percentage of oxygen in the air compressed in the combustion chamber. The fuel supply from the pump may be either manual or governor controlled; in some cases a combined manual and governor control arrangement is used. In the governor control, it is so arranged that only a very small force is necessary to operate the fuel-control device; this enables a relatively small governor to be employed. This feature is especially important in connection with fuel-injection pumps for variable speed engines. Actual operation has shown that the governing of the maximum permissible speed of the engine is not sufficient and that control of the engine under no-load condition is just as important. The operator should be able to control the amount of fuel injected for all intermediate speeds, by means of hand or foot control. Figure 115 shows a convenient arrangement suitable for obtaining the above control with engines using the Bosch fuel pump.

The method of controlling the quantity of fuel is fundamental in the design of the injection pump. This regulation may be effected in one of the three ways shown in Fig. 116. (1) The stroke or lift of the pump

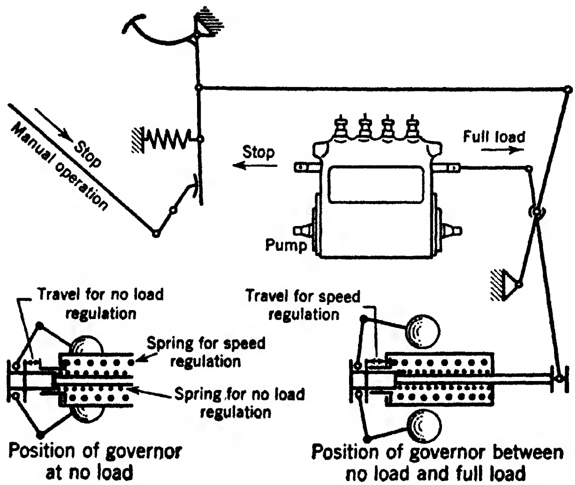


Fig. 115. Arrangement of governor linkage to provide regulation for both speed and no-load conditions.

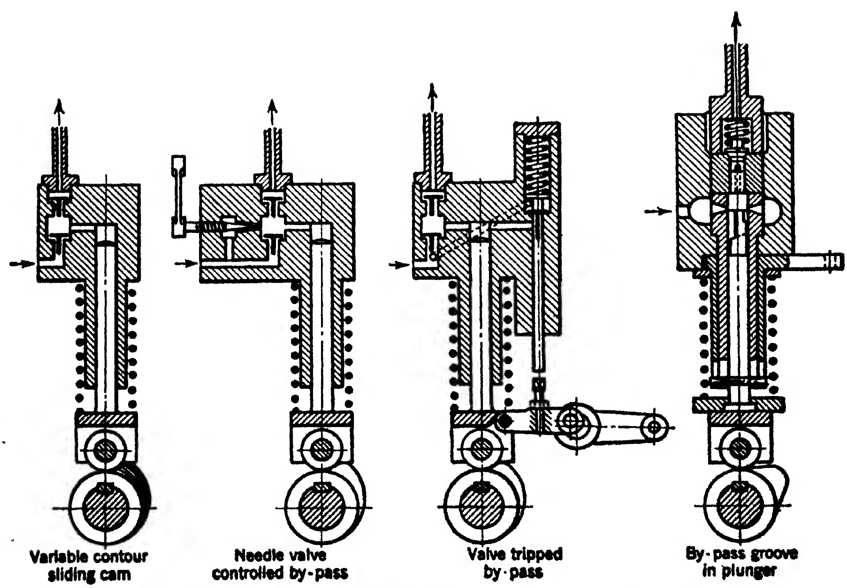


Fig. 116. Methods of regulating fuel delivery from an injection pump.

plunger is varied by means of a *sliding cam* of suitably varied outline. (2) Part of the fuel entrapped in the pump during the working stroke of the plunger is permitted to escape through a by-pass to the fuel intake of the pump. The quantity by-passed, and consequently also the amount of fuel which is permitted to reach the nozzle, is regulated by a *needle valve.* (3) With *by-pass regulation* a by-pass port is opened after the plunger has completed a definite part of its stroke; the port returns the excess fuel to the fuel intake. In practice, this is usually done by opening a special by-pass valve or by opening the inlet valve of the pump for a longer or shorter period before the end of the working stroke of the plunger. Pumps without suction valves, in which the plunger

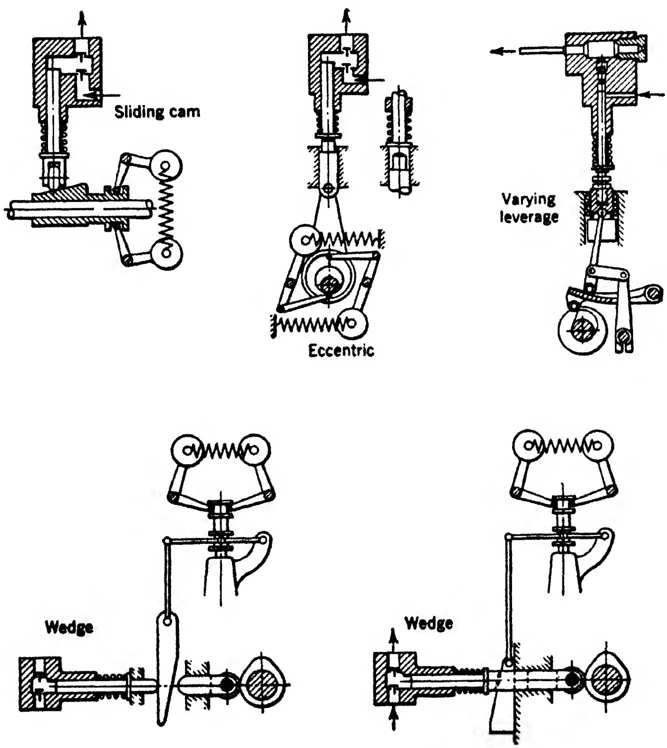


FIG. 117. Methods for varying stroke of pump plunger to control fuel delivery.

controls the intake port, employ a sloping groove in either the plunger or a plunger sleeve which registers with an opening in the plunger sleeve or plunger earlier or later in the working stroke, according to the relative angular position of these parts. The excess amount of fuel can then escape from that part of the pump space which is under pressure.

Several methods of regulating the quantity of fuel by a variable stroke of the pump plunger are illustrated in Fig. 117. The method which includes a sliding cam makes possible a very simple construction for the pump. The shifting may be either an axial sliding or a double-eccentric arrangement. The camshaft of such a pump requires very precise workmanship to obtain uniform contours and accurate spacing of the cams for a multi-cylinder engine. Another method of regulation inserts a wedge in the tappet mechanism to produce a variable stroke. The force necessary to change the stroke of the plunger is rather large in this method, because of the pressure against which the plunger must move; therefore it is seldom applied to pumps requiring pressures above 1000 lb per sq in., because of the excessively heavy governor weights that would be required. With a shifting eccentric, only a small part of the lift of the eccentric can be utilized. Eccentric regulation is applied only to small two-cycle engines. The other method of regula-

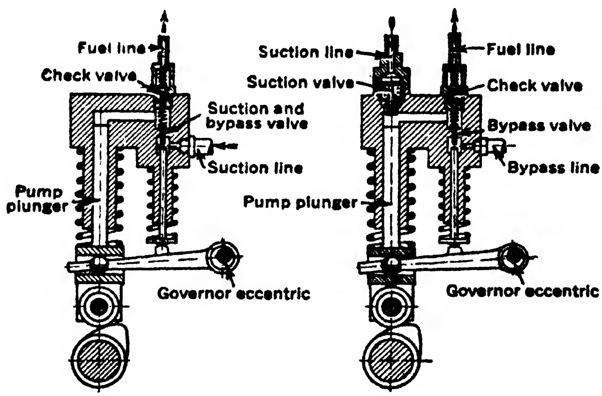


FIG. 118. Valve-regulated by-pass methods to control delivery of fuel pump.

tion shown in Fig. 117 is by variable leverage. The stroke of the pump is changed by sliding the lower end of the push rod on the rocking lever.

For the needle-valve control shown in Fig. 116, the stroke of the plunger remains constant and the fuel which is not needed for injection returns, either to the intake space or by a separate line to the fuel tank, during the upward stroke of the plunger. This type of pump is being replaced largely by pumps having mechanically operated by-pass control. By-passes can be controlled in different ways. Pumps equipped with inlet and outlet valves use the inlet valve for this purpose or a separate by-pass valve is provided. Pumps having plunger valves are provided either with a separate by-pass plunger (Fig. 111) or the plunger itself (Fig. 113) is used for controlling the quantity of fuel; these pumps employ a constant stroke of the plunger.

Diagrams of two fuel pumps having by-pass valves are shown in Fig. 118. The same valve is used for both inlet and by-pass in the diagram at the left, being pushed open during the pressure stroke of the plunger, at a time controlled by the governor, to permit the return flow of the fuel not required. The pump shown at the right functions in the same way except that a separate inlet valve is provided and the excess fuel escaping through the by-pass valve is returned by a separate line to the fuel tank. The advantage claimed for this construction is that any surges which may occur in the suction line will have no effect on the delivery side of the pump. Control of the fuel by-pass by means of the pump plunger, as used with the Winton unit injector (Art. 78) and with the Bosch fuel pump (Art. 79), has found considerable favor and is being used extensively. Typical construction features of the by-pass plunger type of fuel pumps are shown in Figs. 109 and 113.

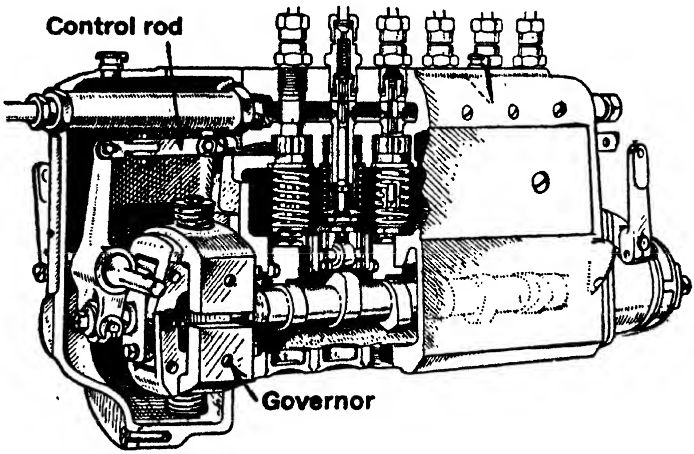


FIG. 119. Bosch governor integral with fuel-injection pump.

With Diesel engines of variable speed it is frequently desirable to have the engine operate under control of the governor at minimum speed so that it will not stop, and at maximum speed so that it will not race; between these two extremes the speed is controlled by the throttle or accelerator. The Bosch governor, operating upon this principle, is housed within the casing of the fuel pump at the fuel inlet end, as shown in Fig. 119. The governor weights are carried on bell-crank levers which are provided with adjustable springs at top and bottom. There is an outer spring to control the idling speed and two strong inner springs for the maximum speed. When idling, the former spring bears on the weight, but as the engine speed is increased the increased centrifugal force causes the governor weights to bear against the two stronger springs. Throughout the normal speed range the weights are held in this position as the centrifugal force is not enough to compress them. When the engine attains or exceeds maximum speed, the centrifugal force becomes sufficiently great to overcome the spring compression and the control rod is moved towards the "stop" position, the fuel to the engine is reduced, and the desired maximum engine speed is not exceeded.

The Woodward Type-IC hydraulic governor is shown schematically

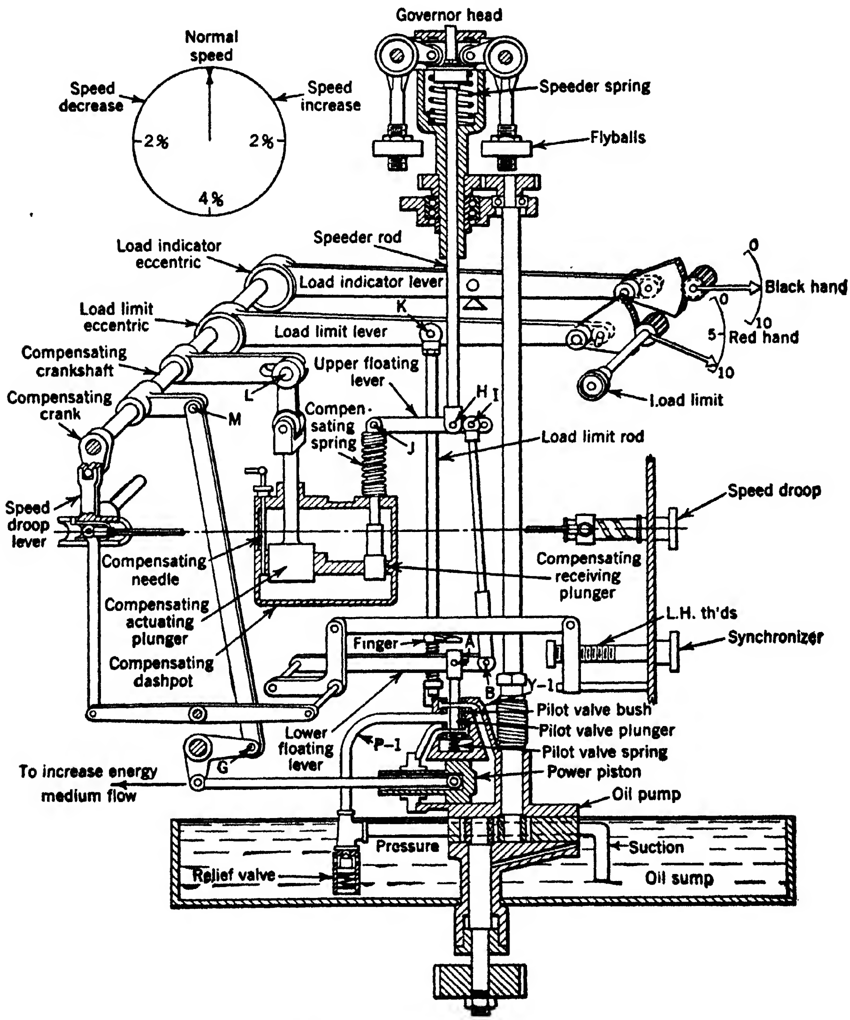


FIG. 120. Woodward Type-IC hydraulic governor.

in Fig. 120; it maintains constant speed of the engine regardless of variations in load. This governor employs the usual control element of weights in which centrifugal force is opposed by a spring. This motion is not transmitted directly to the fuel-pump control linkage; but it acts indirectly through a pilot valve that controls the position of the governor power piston. Oil for actuating the power piston comes from a small rotary pump that maintains a constant supply of oil at the proper pressure controlled by a relief valve.

The action of the governor, when load is added to the engine, is as follows. The speed starts to decrease, the fly balls move in, pivots H, I, B, and A are raised, the pilot valve plunger is raised, oil under pressure is admitted from P–1 to Y–1, and the power piston moves to the left to increase the flow of energy medium. As the power piston moves to the left, pivots G, M, and L are lowered, the compensating actuating plunger is forced downward, the compensating receiving plunger is forced upward, raising pivot J and lowering pivots I, B, and A until the pilot-valve plunger centers, stops the flow of pressure oil into port Y–1 and stops the movement of the power piston at the exact position corresponding to the increased load of the engine. As the energy medium flow is corrected, the speed of the engine returns to normal. In unison with the return of the speed to normal and consequently the return of the fly balls to their central position, the compensating receiving plunger returns to its central position, owing to the centering action of the compensating spring which forces the oil through the compensating needle valve. While awaiting another momentary change in speed, resulting from a change in load, the pilot valve plunger and the power piston remain stationary.

The time required for the compensating receiving plunger to return to its normal position is controlled by the small compensating needle and must be so timed that the pilot plunger remains stationary and keeps the port in the pilot valve bushing closed, while the fly balls return to their normal position. Thus compensation is a follow-up action; but the net effect is that, whereas the power piston has a different position for each engine speed, the governing element returns to the same position after the fuel control has been adjusted to suit the new load conditions. In this way, the engine continues to have the same speed regardless of load and the governor is of the isochronous type. The power required to move the balanced pilot valve is very small, hence the governor is very sensitive.

The governor is provided with auxiliary devices to accommodate any desired type of engine or parallel operation with other engines. Load adjustment, when operating in parallel with other units, and speed adjustment, when operating alone or for synchronizing, are obtained by means of the synchronizer, either manually or by electrical remote control. A graduated indicator on the front of the governor shows the setting of the synchronizer both for complete turns and for fractions of turns, reading directly to 1/100 of one turn of the synchronizer knob.

Basically isochronous, the Woodward governor does not require either a drooping or rising speed characteristic for proper isolated unit operation. An adjustable drooping characteristic is added, however, as a standard auxiliary for parallel operation of alternating-current generator drives. Rotation of the speed-droop graduated dial on the front of the governor causes a change in the length of the connection between the fly balls and the governor oil-control valve as a linear function of movement of the engine fuel-control shaft from the zero fuel position. The droop is adjustable from 0 to 6 per cent.

A load limit (red hand) is provided on the front of the governor to limit the maximum load on the engine to any predetermined value. The load-limit adjusting knob may be used to shut the engine down by revolving the knob until the load-limit pointer reaches zero. A second pointer (black hand) on the load-limit dial is provided to indicate the actual position of the engine fuel-control shaft. This second, or fuel-indicating pointer, is useful as an indicator of economical engine operation, for if at the same kilowatt output the indicator shows increased fuel consumption, there must either be something wrong in the engine or scavenger, or the grade of fuel oil is inferior.

CHAPTER XII

DESIGN OF MAJOR ENGINE PARTS

The subject matter presented in this chapter must necessarily be limited to basic considerations for the design of the more important parts of internal-combustion engines. All engine designs are based on compromises; it is upon the soundness of judgment by which these compromises are made that the success of the engine ultimately depends. At times there are several ways in which a design may be worked out, and each may appear to be equally logical. Usually the best procedure is the one involving the fewest assumptions; however, some assumptions are necessary in every design.

81. Selection of Engine Parts and Materials. There is a prevalent belief that the reliability and even the efficiency of an engine are, to a large extent, dependent on the number of parts it contains. In general, this belief is a fallacy. It is obvious that the number of parts should be reduced to a minimum without sacrificing efficiency, mechanical correctness, and dependability; but this reduction can also be overdone. Frequently few parts denote an excess of compromise. Many of the parts of an internal-combustion engine are subjected to complicated stresses, both thermal and mechanical; these stresses should be considered separately, and additional parts used as may be required to reduce and simplify the stresses. No part should be subjected to compound stresses if by the provision of additional members the stresses can be divided; for example, when a member is subjected to combined torsion and bending it may be preferable to replace it by separate members, one designed for bending and free from torsion, another subject to torsion only. This may require more parts in the engine, but the safety and reliability would be greatly improved.

That an increased number of parts involves increased care and maintenance of the engine by the user is not necessarily true. Fewness of engine parts saves manufacturing costs to some extent, but it certainly confers no benefits on the user. It is far better to use ten parts if necessary to comply with the laws of mechanics than to defy these laws with only one part. The internal-combustion engine has received considerable condemnation because of its actual or supposed unreliability, and the numerous unusual occurrences which may incapacitate the engine. First cost and economy of operation are generally the most important factors considered in the selection of an engine; but of equal importance for consideration should be the reliability and durability of its principal parts, and a decrease of repairs and costs of maintenance.

It is commonly supposed that to be successful an internal-combustion engine must be made from very carefully selected and highly specialized materials. In general, the higher the quality of the materials, the better is the engine; but with appropriate design, the ordinary materials available will be found to give satisfactory results. Surface hardness is frequently of more importance than the tensile strength of wearing parts. Designing for extreme rigidity may require the use of comparatively heavy parts, hence tensile strength may be of secondary importance. When the weight is to be a minimum, as for an airplane engine, or when a high unit output or high speed is desired, the use of materials of high strength and light weight is warranted.

Design problems of the modern high-speed engine would have been insurmountable if metallurgy had not provided alloys of high strength and light weight to be used for pistons, connecting rods, bearings, crankshafts, and other highly stressed important parts of the engine. Nickel iron or molychrome iron may be used for frames, crankcases, and cylinder blocks; high-strength aluminum alloys containing copper, magnesium, and nickel may be used for pistons; and chrome-nickel steels for connecting rods, crankshafts, and many small interconnecting parts.

82. Cylinders. A special grade of iron generally containing nickel, molybdenum, or about 40 per cent steel is used in casting the cylinder. These metals produce hard, close-grained, uniform castings, which have good wearing qualities. The cylinder liner may be cast integral with (Fig. 74) or separate from (Fig. 66) the cylinder block. The former design requires a more liberal wall thickness than is usual with a separate liner. With thinner walls conduction of heat is more rapid and stresses due to heating are lower. Most internal-combustion engine cylinders are completely water-cooled with jacket spaces, which usually extend the entire length of the cylinder. Cylinders and jackets are often cast in one piece; in small engines and in some large engines the cylinder, jacket, and frame may be combined into one casting. Figures 65, 81, and 83 show engines in which a separate liner has been used, the liner forming the cylinder proper and the space between the liner and jacket providing the cooling-water space. A separate liner has the advantage that a specially suitable grade of iron can be used, and the jacket and frame may be cast more easily. Also the wear on the cylinder may be taken care of more readily and expansion due to temperature variations is possible without imposing dangerous stresses on cylinder and jacket. To allow this free expansion, the cylinder liner is held rigidly at the top and the bottom is allowed to move vertically.

When the wall of a cylinder, which is subjected to an internal fluid pressure, is thick relative to the internal diameter, the stress in the wall is not uniformly distributed over the cross section. In such cylinders the stress is greater at the inner surface and decreases to a minimum at the outer surface; this stress is tensile. The cylinder of an internal-combustion engine is generally considered to be a thick cylinder and subject to that design analysis. A number of equations have been deduced to express the relations between pressure, stress, and cylinder thickness; of these, the equations of Lamé, Birnie, and Clavarino are the best known. Lamé's equation, which is still used for cast-iron cylinders, neglects the factor of lateral contraction (Poisson's ratio). Birnie's equation assumes that the ends of the cylinder are open and that, as a consequence, there is no longitudinal stress. When the ends are closed, Clavarino's adaptation of Lamé's equation is preferable for the design of steel cylinders.

// The thickness of the cylinder should be determined to withstand the internal pressure of the explosion and the stresses produced by the connecting rod thrust in a trunk-piston engine. The internal pressure stresses the cylinder barrel in an axial and radial direction. The cylinder wall proper takes all the radial load, and all the axial load if the water jacket is not continuous and cast integral with the cylinder. For brittle materials like cast iron, the following slightly modified form of Lamé's *thick-cylinder equation* gives results which agree very closely with practice.

$$t = \frac{D}{2}\left[\sqrt{\frac{S+p}{S-p}} - 1\right] + t_a \tag{71}$$

where t = thickness of cylinder wall, inches.

D = diameter (internal) of cylinder, inches.

S = allowable stress; not more than 4000 lb per sq in. for cast iron, 8000 for steel cylinders.

p = max. cylinder pressure, pounds per square inch.

t_a = added wall thickness for reboring or regrinding; $\frac{1}{8}$ in. for cylinders of 4 to 8 in.; $\frac{1}{4}$ in. for 10 to 20 in.; $\frac{3}{8}$ in. for 22 to 32 in.; and $\frac{1}{2}$ in. for 34 to 48 in.

All cylinders subject to high temperatures, as those for internal-combustion engines, should be as thin and flexible as possible. The allowable stress should be about 4000 lb per sq in. in order to get a construction that is not too rigid against expansion by heat and to obtain more rapid heat transfer. For a first approximation the use of the *thin-shell equation* $pD = 2tS$ is frequently permissible. In any case a special allowance must be added to the wall thickness for reboring.

or regrinding, as indicated in equation 71. Moreover, with trunk pistons, the cylinder is subject to sidewise deflection due to the angularity of the connecting rod. Finally, there are cases where the weight of an overhanging cylinder, or of heavy attached combustion spaces or valve housings, have a serious influence on the stresses in the cylinder. For these reasons strength calculations can give only a lower limit of thickness, and many engineers design cylinders by rule-of-thumb methods based on experience.

The stress on the jacket wall depends upon the arrangement of cylinder and jacket walls. When the cylinder and jacket are cast in one piece, as in Fig. 74, the jacket takes part of the axial thrust due to the explosion pressure. If the arrangement were as shown in Fig. 66 the cylinder wall would be depended on to take all loads due to the explosion. Generally, the thickness of the jacket wall should be from two-thirds to three-fourths of the thickness of the cylinder wall. The width of the water space may be made from 1 in. for a 6-in. cylinder to 3 in. for a 30-in. cylinder. The water should be introduced at the lowest point and discharged at the highest point; there should be no dead corners which eventually might form air pockets or steam pockets, and the water circulation should completely encircle the cylinder walls. Accretions of metal and projecting thin edges may lead to cracking and ultimate rupture.

Example. Find the wall thickness of a 17 × 25 in. Diesel-engine cylinder with a maximum internal pressure of 600 lb per sq in. Assume a working stress of 4000 lb per sq in. Use (*a*) thick-cylinder equation and (*b*) thin-shell equation. Also estimate (*c*) the jacket wall thickness and (*d*) the width of the water space.

Solution. (*a*) $t = \frac{D}{2}\left[\sqrt{\frac{S+p}{S-p}} - 1\right] + t_a$

$$t = \frac{17}{2}\left[\sqrt{\frac{4000+600}{4000-600}} - 1\right] + \frac{1}{4}$$

$t = 1.64$ in.; use $1\frac{3}{4}$-in. thickness.

(*b*) $t = \frac{pD}{2S} + \frac{1}{4}$

$$t = \frac{600 \times 17}{2 \times 4000} + \frac{1}{4} = 1.53 \text{ in. (too small)}$$

(*c*) $\frac{2}{3} \times 1\frac{3}{4}$ in. $= 1\frac{3}{16}$ in. thick jacket wall

(*d*) About $2\frac{1}{2}$ in. width of water space.

83. Cylinder Heads, Valves, and Valve Gear. Cylinder heads are largely empirical designs as there are no satisfactory rational equations for determining the thickness of the metal. The material generally used is cast iron, cast iron containing about 40 per cent steel, or aluminum alloy. The accurate determination of the thickness is usually difficult because of the complicated shape of most cylinder heads, but a flat circular-plate equation may be used as a check.

$$t' = KD\sqrt{\frac{p}{S}} \tag{72}$$

in which t' = thickness of cylinder head, inches.

K = constant; 0.33 to 0.40 for cast iron, 0.30 for semi-steel and steel, and 0.28 for aluminum alloy.

D = diameter of cylinder, inches.

p = max. cylinder pressure, pounds per square inch.

S = allowable stress; not more than 4000 lb per sq in. for cast iron, 6000 for semi-steel, 8000 for steel, and 5000 for aluminum alloy.

Frequently the cylinder head is made the same thickness as the walls of the cylinder. In most cases the shape of the head is very irregular because of the valve pockets, gas passages, fuel-injection nozzles (or spark plugs), and water jackets. The metal around the passages, provided for these parts, produces a girder-like structure and the thickness of the cylinder head may be reduced. The cylinder head is generally a one-piece casting and (except for L-head engines) symmetrical about the center; in two-cycle designs the inner surface may be made spherical—the strongest structure possible—thus permitting a minimum thickness of metal with consequent ready flow of heat to the cooling water.

Example. Find the thickness of a plain cylinder head for a 17 × 25 in. Diesel engine with a maximum cylinder pressure of 600 lb per sq in.

Solution.

$$t' = KD\sqrt{\frac{p}{S}}$$

$$t' = 0.33 \times 17\sqrt{\frac{600}{4000}} = 2.17 \text{ in.}$$

NOTE. Compare result with cylinder thickness obtained for the same engine in the previous example.

In the design of *valves*, it is necessary to remember that the objects are to provide easy entry and exit for the gases and to keep the valves as small as possible when consistent with the first condition. The flow of the gases on either side of the valve port should be as free as possible with a minimum of abrupt bends or changes of section. The use of unduly large valves, and particularly large valves with reduced lifts, should be avoided for the following reasons: the provision of relatively large valves will invariably be detrimental to the compactness of the combustion chamber; when the valves are large, a lower lift is required and the gas velocity may be lower, resulting in less turbulence, power, and efficiency; the larger the valve, the higher will be its temperature, the lower will be the volumetric efficiency and the density of the charge, and the higher the temperature of the cycle; also large valves may produce unnecessarily heavy stresses on the valve-operating gear.

That valves should be as light as possible and at the same time consistent with mechanical strength and heat dissipation is obvious; but there is a tendency in many high-speed engine designs to reduce the weight of valves too much, with the result that stretching, distortion, and overheating may occur. When the entire reciprocating mass of the valve and its valve gear are taken into account, it will be found that the weight of the valve head alone forms but a very small part of the whole, and it is generally poor practice to save metal in the valve head and stem for the sake of the relatively small saving in weight. A valve transfers its heat largely through its seat, hence a fairly wide seat should be used in order to provide a sufficient area of contact when the valve is at rest. Narrow valve seats save some expense, both in first cost and subsequent grinding; but the use of wide seats tends to a higher efficiency and better sustained performance. In some cases, warping of valve heads may offset the advantages of wide seats.

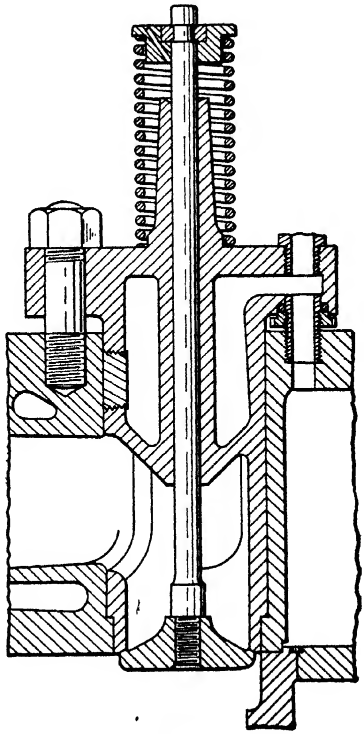


Fig. 121. Typical valve and valve-cage construction for a large engine.

Valve cages and valve seats, as required for large engines, should be made of close-grained cast iron; the valve head may be made of hard cast iron, threaded and riveted to the medium-carbon steel valve stem, as shown in Fig. 121. The inlet and exhaust valves for four-cycle engines are usually interchangeable and located symmetrically in the cylinder heads; the exhaust-valve cage should be water-jacketed, carrying the jacket as far down as possible. Valve-seat inserts for the exhaust valves may be made separate, from nickel cast iron, stellite, copper-tungsten, or cast aluminum-bronze. They are pressed or shrunk into the valve cages or cylinder head. If these seats are to be reliable a high degree of skill in design and manufacture is essential, because of the differences in thermal expansion of the dissimilar adjacent materials used. For high-speed engines, the exhaust valve is made of a heat-resisting material; many of these are available. Hollow valve stems may be used to reduce the weight of the valve and to provide better cooling. The hollow stem is frequently filled with a salt which melts at valve-stem temperatures and carries the heat from the head toward the lower end of the valve stem. Exhaust temperatures of the Diesel engine are lower than those of the gas engine; hence materials do not show such marked differences in performance. These metals are mostly chrome-nickel irons or steels, sometimes alloyed with silicon, tungsten, copper, manganese, etc. The inlet valve is usually made from one of the forging varieties of medium-carbon, chrome-nickel steels.

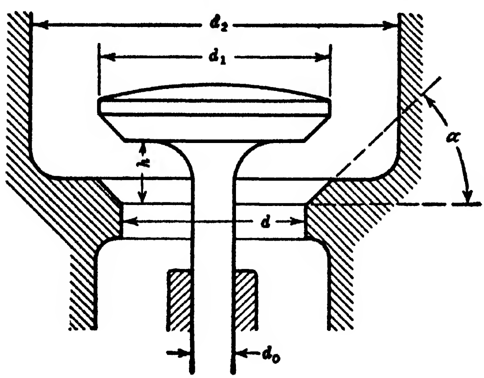


FIG. 122. Poppet valve proportions.

The so-called poppet or mushroom valve, see Fig. 122, is used almost exclusively in internal-combustion engines; valve ports may be controlled by the edges of the piston, as in two-cycle engines. Valve dimensions are based on the "mean gas velocity" through the valve opening and the piston displacement, then

$$v_g \times a = v_p \times A$$

$$v_g = \frac{A}{a} v_p = \left(\frac{D}{d}\right)^2 v_p \tag{73}$$

and

$$d = D\sqrt{\frac{v_p}{v_g}} \tag{74}$$

where d = diameter of valve, see Fig. 122, inches.
D = diameter of piston, inches.
v_p = mean velocity of piston, feet per second, $\frac{2LN}{60}$.
v_g = mean velocity of gas through valve, feet per second.

The mean value of v_o may be assumed as 140 to 180 ft per sec for stationary engines, and 175 to 225 ft per sec for automotive and aircraft engines. The angle α of the valve seat is usually 45 degrees. The main advantage of a conical seat is its self-centering tendency. The valve lift for small moderate-speed engines is $\frac{1}{5}$ to $\frac{1}{4}$ of the valve diameter. In large engines questions of valve design and in high-speed engines timing of the closure of the inlet valve make it necessary to decrease the valve lift; h in Fig. 122. In actual engines the lift is made between 0.10d to 0.25d. A smaller lift reduces the noise due to hammering of the valve against its seat when closing. Other valve dimensions may be determined by these equations; refer to Fig. 122.

$$d_o = \frac{d}{8} + \frac{1}{8}\text{ in. to } d_o = \frac{d}{8} + \frac{3}{8}\text{ in.} \tag{75}$$

$$d_2 \geqq \sqrt{d^2 + d_1^2} \tag{76}$$

For equations 73 and 74 the valve area (conical area of opening) is taken at full lift, and the pressure difference between outside air and cylinder is ignored; thus the expressions lack mathematical accuracy but their use is quite as satisfactory—or more so—as the involved method of plotting piston velocities and corresponding valve areas to draw a curve of gas velocities. The latter is never realized in practice because of pressure changes and particularly because of the inertia of the gas columns in the intake pipe and exhaust pipe. Tests have shown that there is no advantage in reducing the mean gas velocity below 150 ft per sec; a decrease in efficiency and power is noticeable when the velocity exceeds 200 ft per sec. Cylinder-head ports should be increased, outside the valve throat, the area being increased 50 per cent if practicable without introducing casting difficulties. At high speeds large valve areas are required and are best obtained by the use of dual inlet and exhaust valves. The smaller dual valves offer many advantages, as previously stated in this article.

The exhaust valve should close after the piston has moved away from top dead center position, so that a little wear, expansion, or finishing tolerance will not cause it to close before the exhaust stroke is completed. The exhaust valve should close not less than 10 degrees after upper dead center for slow-speed engines and up to 25 degrees for high (rotational) speed engines to allow for wiredrawing losses and clearing of exhaust gases from the cylinder by the inertia of the gases. The opening of the exhaust valve is governed by starting conditions in air-starting stationary engines. The duration of starting-air admission should not be prolonged so that both starting and exhaust valves are open at the same time; this would result in the needless loss of compressed starting-air. For most engines, low speed and high speed, the exhaust valve opens at 35 to 50 degrees before lower dead center position of the piston.

It is customary to open the inlet valve 5 to 20 degrees before upper dead center; thus both inlet and exhaust valves are open at the same time and scavenging of the cylinder may be effected. Tests have shown this procedure to be economical; but where a long exhaust pipe is used or where the exhaust must be muffled, it may be advisable to open the inlet very little before dead center position of the piston; see Figs. 41 and 44. Closing of the inlet valve occurs at 30 to 60 degrees after lower dead center position of the piston, for slow-speed engines 30 degrees lag may be enough, but for high-speed Diesel and automobile engines 45 to 60 degrees lag is necessary.

The object of the early opening and late closing of the valves is to reduce the gas velocities and the flow resistance through the ports. The opening of the inlet valve occurs only slightly before upper dead center, but the closing is retarded considerably to utilize the high velocity of the gases induced in the intake passage to force a larger charge into the cylinder. On the other hand, the exhaust valve closes near the upper dead center but should open considerably before the lower dead center to reduce the pressure in the cylinder as near to atmospheric pressure as possible before the piston begins its upward (exhaust) stroke. Timing of the valves is a function of the engine speed; as the speed increases the intake valve closes later and the exhaust valve opens earlier. However, the actual time of opening and closing is not as important as the amount (length of time) of valve opening. The best valve timing can be determined only by actual tests, as the design of the intake and exhaust passages is an important correlated factor. For example, the best time for closure of the inlet valve can be established experimentally by plotting the compression pressure with each valve setting; the setting that gives the highest compression provides the best charging of the cylinder and should be used.

Valves are usually actuated by means of cams, with interposed valve-operating parts; the cam contour is formed by two arcs joined by two flanks which may be straight lines (roller-follower type only, Figs. 82 and 126) or curves tangent to both arcs; see Fig. 83. The cam transmits its motion to the valve by means of a cam follower which is generally one of four types: the roller-follower type, Figs. 62, 82, and 126; the pivoted follower, Fig. 34; the mushroom follower, Figs. 40 and 83; or the barrel-type follower, Figs. 43 and 123. The cams should preferably be made integral with the camshaft; the camshaft should be drop-forged, then case-hardened at the cams and journals. The cams may be copper-plated after grinding to prevent scoring the followers, also called lifters or tappets. The camshaft is supported in the crankcase or overhead in steel-backed babbitt or bronze bushings and driven from the crankshaft by gears, Figs. 34 and 67, or by a "silent" or roller chain.

There are many successful types of valve-gear mechanisms used on internal-combustion engines; several typical designs will be briefly described. In Fig. 34 is shown a valve gear with the inlet and exhaust valves in the cylinder head. The camshaft, which is above the level of the cylinder head, is driven by a vertical shaft through bevel gears. The operating (or rocker) arms are short; hence the valve stems have long guides to prevent wear and to assure continuity of accurate valve seating. Figures 42, 43, 62, 82, 83, 123, and 126 show valve gears with the inlet and exhaust valves in the cylinder head; the camshaft is located in the base of the engine and each valve has its own cam. The valves open downward and are operated by long pushrods through short rocker arms placed on top of the cylinder. The valve lifter (follower) in Fig. 123 is a single-piece iron casting with chill-hardened and ground outer surfaces; cored openings in the side wall of the lifter decrease its weight and allow for ample lubrication. These lifters are carried in guide holes reamed in the crankcase, in a compartment at one side of the cylinder block. This compartment is open below allowing oil from the rocker arms to drain back into the crankcase.

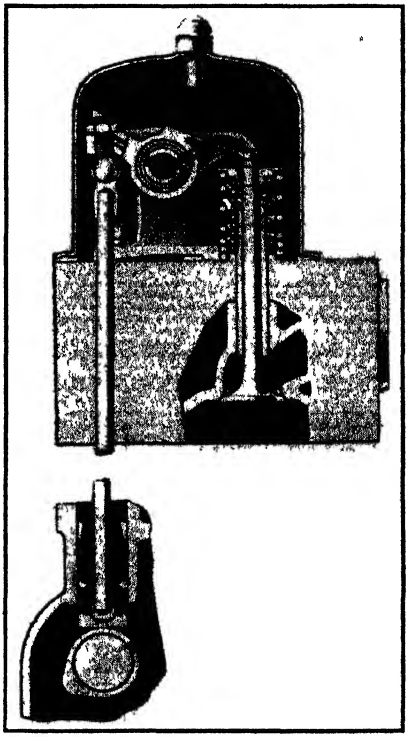


FIG. 123. Valve and valve-operating mechanism for overhead-valve automobile engine.

Figure 39 shows another arrangement of valves, called the L-head design, for a vertical multi-cylinder engine. The valves open upward and the valve gear is simpler than for overhead valves because of the elimination of rocker arms and the use of shorter pushrods. A complete description of the L-head valve design and operation is given on page 122. Figure 57 illustrates a valve gear for a single-acting horizontal engine. The valves are near the bottom of the combustion chamber, and open upward. Both valves are operated through levers, passing under the cylinder, from cams on the camshaft, which is located at the side and below the center line of the engine. The exhaust cam is so designed that, for air starting, the compression can be relieved by shifting the cam on the shaft. In some large horizontal gas engines the valves are driven from the same eccentric, the desired lift of the valves being obtained by rolling levers.

84. Pistons and Piston Rings. In any internal-combustion engine of normal design, piston friction forms by far the largest of the mechanical losses. Of the many engine parts the piston is probably the most difficult of design in that it must fulfil so many conditions, some of which may be contradictory. The piston receives the energy impulse from the expanding gases in the cylinder and transmits this energy through the connecting rod to the crankshaft. The piston must also transmit a large amount of heat from the combustion chamber to the cylinder walls. With the older types of slow-speed engines having a low power capacity per cubic inch of piston displacement, the design problems were not as important as they are today with the high-speed and high-capacity engines. Some modern engines develop their maximum power at 3000 to 4000 rpm, and heavy reciprocating pistons would develop high inertia forces which are undesirable. The main objects in the design of a piston are: minimum weight, strength to withstand fluid pressures and inertia forces, bearing area sufficient to prevent undue wear, gas and oil sealing of the cylinder, high-speed reciprocation without noise, dispersion of the heat of combustion, resistance to thermal and mechanical distortion, and adequate support for the piston pin.

Trunk pistons are generally used for internal-combustion engines. These pistons, Figs. 39, 43, 61, 62, 81, 83, and 126, consist of a head to withstand the cylinder pressure and heat, a skirt to act as a bearing for the connecting-rod side thrust, a piston pin to connect the piston with the connecting rod, and piston rings to prevent leakage of cylinder gases. These pistons, used in single-acting engines, are open at one end. Cast iron or steel is used for the pistons of large stationary engines, whereas cast iron or aluminum alloys are used for the smaller high-speed internal-combustion engines. Cast iron and steel have the high strength desired, good wearing qualities at the required temperatures, low thermal expansion, but relatively low heat conductivity. The chief advantage of aluminum alloys is their light weight (40 per cent of cast iron). The strength at low temperatures is good but they lose about half their strength at temperatures above 600 F. The thermal expansion of the aluminum alloys is nearly two and one-half times that of cast iron and the resistance to abrasion is low at high temperatures; however, aluminum has a very high heat conductivity, three and one-half times that of cast iron.

The quantity of heat flow to the piston head can be estimated from

$$h = \frac{H_f \times a \times W_f \times bhp}{A} \tag{77}$$

where h = heat flow to piston head, Btu per square inch per hour.
H_f = high heat value of fuel supplied, Btu per pound.
a = part of heat supplied that was absorbed by piston, 0.05 for most engines.
W_f = weight of fuel supplied, pounds per brake horsepower per hour.
bhp = brake horsepower per cylinder.
A = piston head area, square inches.

Heat absorbed by the piston head flows through the head toward the cylinder wall. This condition necessitates an assumption of a temperature drop from the center toward the edge. For a plain disk of even thickness, which condition approximates that of a conventional flat-head piston, the heat flow equation will be

$$T_c - T_e = \frac{h \times D^2}{16 \times t_p \times K} \tag{78}$$

and

$$t_p = \frac{h \times D^2}{16K(T_c - T_e)} \tag{79}$$

where t_p = thickness of piston head, inches (*thermal stresses*).
h = from equation 77.
D = diameter of piston head, inches.
K = heat conductivity, Btu per cubic inch per hour per degree temperature difference; 2.2 for cast iron and 7.7 for aluminum.
T_c and T_e = temperature at center and edge of piston head, degrees F.

Experiments on successful piston designs indicate that $(T_c - T_e)$ is about 400 F with cast iron, and T_c is 800 F; for aluminum pistons $(T_c - T_e)$ is about 130 F, and T_c is 500 F.

The piston section as determined for heat transfer should be checked and modified for structural efficiency. The piston head may be treated as a flat plate with a uniform load and rigidly supported at the outer edge.

$$t'_p = \frac{D'}{2}\sqrt{\frac{p}{S}} \tag{80}$$

where t'_p = thickness of piston head, inches (strength).
D' = inside diameter of piston (under) head, inches; D' $D - 2t'_p$, approximately.
p = maximum cylinder pressure, pounds per square inch.
S = allowable stress in tension; 4000 lb per sq in. for cast iron, 5000 for aluminum.

If the thickness calculated for strength exceeds that calculated for thermal stresses, the piston head may be reenforced by a longitudinal rib or ribs, Figs. 62, 65, 74, 81, and 83, of a depth in excess of t_p by $\frac{1}{8}$ in. or more. The rib or ribs should be cast with as big radii as is consistent with the size of the piston; this reduces the high cooling stresses which are likely to occur with sharp changes in section.

Empirical equations frequently used for determining other piston dimensions are:

$$\text{Thickness of wall under rings} = t_p \text{ or } t'_p \tag{81}$$

$$\text{Length of piston} = 1.25D \text{ to } 2D \tag{82}$$

$$\text{Number of piston rings} = 2\sqrt{D} \tag{83}$$

$$\text{Width of piston rings, small engines} = \frac{D}{25} \tag{84}$$

$$\text{Width of piston rings, large engines} = \frac{D}{32} \tag{85}$$

$$\text{Thickness of concentric piston rings} = \frac{D}{32} \tag{86}$$

$$\text{Thickness of eccentric rings, opposite joint} = \frac{D}{27.5} \tag{87}$$

$$\text{Thickness of eccentric rings, at joint} = \frac{D}{55} \tag{88}$$

The length of the piston skirt below the ring section should be such that the pressure due to the side thrust of the connecting rod does not exceed 25 lb per sq in. during the expansion stroke. The center of the piston pin should be from 0.02 D to 0.04 D above the center of the skirt to offset the turning effect of friction. The diameter of the piston pin is determined by allowing 1200 to 4000 lb per sq in. bearing pressure due to the maximum cylinder pressure; see equation 97 and Table XVIII. The length of the ring section is determined by the number and size of rings; small internal-combustion engines use two or three compression rings and one oil ring, engines of larger size use up to six compression rings and two oil rings. Since the heat transfer from the rings to the cylinder wall is better than the transfer from the piston wall to the rings, it is desirable to use many narrow deep rings in preference to a few wide shallow rings. In order to prevent sticking of the topmost ring, this ring should be located at a distance from the end about equal to the thickness of the piston head. Side thrust from the connecting rod is transmitted through the piston-pin bosses to the skirt; and a stiffening rib should be provided at the center line of the boss and extend around the skirt to distribute this load and to prevent distortion of the cylindrical skirt. Short triangular ribs, Fig. 124, should extend from the lower edge of the ring section to the bosses to carry the cylinder pressure. In some designs these ribs are extended across the head, a construction which may cause the head expansion to force the skirt out of round. At the open end of the piston one or more concentric flanges are provided, see Fig. 126, for stiffening and to locate the piston during machining operations. In high-speed engines the inertia of the piston at the upper end of the stroke tends to throw the head and ring section away from the piston pin; hence the thickness of the wall section under the rings must be sufficient to carry this inertia force.

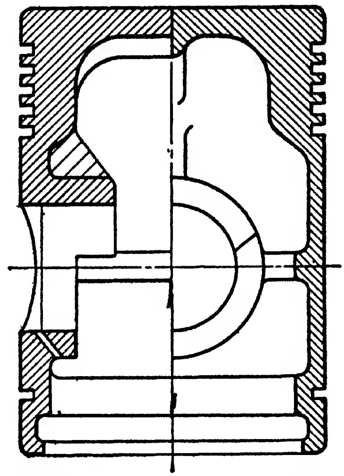


FIG. 124. Typical piston design.

85. Connecting Rods. Connecting rods may be made of cast steel, forged steel, or aluminum alloys. Cast-steel connecting rods may be used for low-speed gas engines, forged steel (carbon or mild nickel) is generally used for most internal-combustion engines, and aluminum-alloy connecting rods are being used in some high-speed engines. The wrist- or piston-pin end is usually provided with a bronze bearing since space is limited and it is not possible to use a large bearing. The crank end of the connecting rod is made adjustable so as to permit taking up wear; the typical design for small engines is shown in Fig. 125; for larger engines the crankpin end is of the marine type and consists of the two cast-steel boxes fastened as shown in Fig. 126. Connecting rods for single-acting engines with trunk pistons are generally not made adjustable because of their inaccessability. The rod-end design depends upon the crankpin and piston-pin sizes, which are determined by their bearing capacities.

The chief considerations in the design of a connecting rod are that it vibration; that it should be as light as possible; that the crankpin end of the rod should be sufficiently rigid to ensure adequate support to the bearing. It is clearly most important to keep the weight of the rod as low as possible, consistent with fulfilling the other requirements. When I-section rods (Fig. 125) are used, and this is probably the most desirable form for high-speed engines from the point of view of manufacture and disposition of the material, the section must have sufficient width to

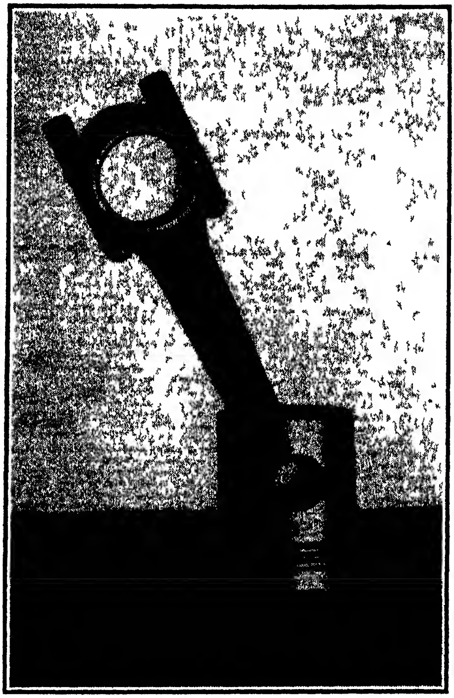


FIG. 125. Steel connecting rod with I section, for small engine.

resist vibration in the plane of the crankshaft, as well as sufficient depth to resist bending.

The body of the rod is considered as a column with pin ends subjected to an additional bending load because of the inertia of the oscillating rod. The stress due to the column action is

$$S_c = \frac{F_a}{a\left[1 - \frac{S_y}{4m\pi^2 E}\left(\frac{L}{\rho}\right)^2\right]} \tag{89}$$

where F_a = axial force on rod due to gas pressure, and inertia of piston and other reciprocating parts, pounds; see equation 90.

S_y = yield point (elastic limit) of material; 35,000 lb per sq in. for carbon steel, 50,000 for nickel steel, 20,000 for aluminum.

a = section area of rod, square inches.

m = column-end factor, 1 for pin ends, 2 for one-end pinned and other-end guided.

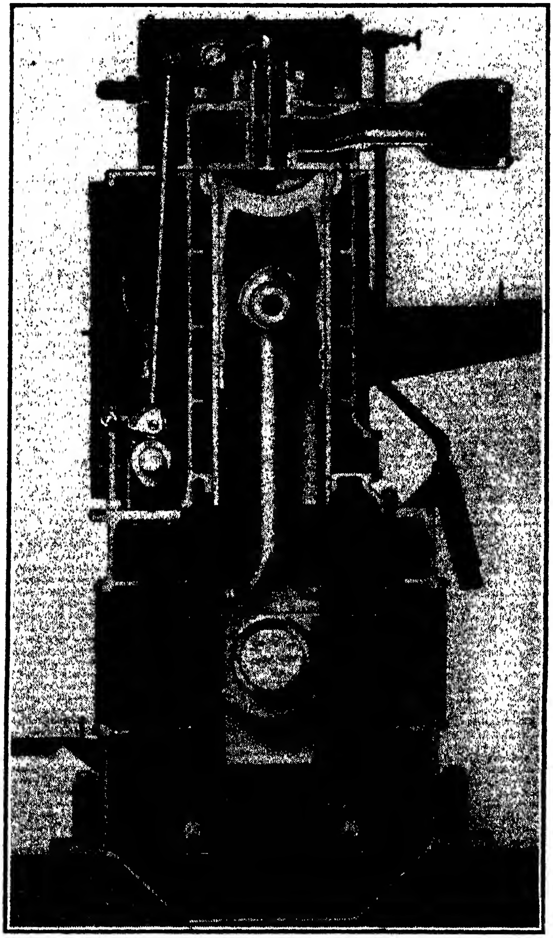


FIG. 126. Steel connecting rod with circular section, machined all over, for larg engine. Also note cylinder, valve, valve-operating mechanism, and piston design.

E = modulus of elasticity; 30,000,000 lb per sq in. for steel, 10,000,000 for aluminum.

L = length of connecting rod, center to center, inches.

ρ = *least* radius of gyration of the rod section.

The value of F_a in equation 89 is

$$F_a = \frac{F}{\sqrt{1 - \frac{R^2 \sin^2 \theta}{L^2}}} \tag{90}$$

in which F = maximum load on piston, pounds.
R = radius of crank, inches.
θ = angle between crank and cylinder center line, measured from head-end dead center position.

Since the cylinder pressure drops rapidly during the early part of the expansion stroke, several values of the piston pressure, corrected for inertia effects, must be used with the corresponding values of θ to determine the maximum load F_a on the rod.

The connecting rod may be assumed to be uniform in section, without an appreciable error; in which case the resultant inertia force, due to the acceleration of the oscillating rod, will act at a distance of $\frac{2}{3}L$ from the piston-pin end and will have a magnitude of

$$F_i = \frac{12Wv^2}{2gR} \sin \theta \tag{91}$$

where W = weight of connecting rod, pounds.
v = velocity of crank pin, feet per second.

The maximum bending moment produced by this force will be at a distance of $0.577L$ from the wrist pin and its value will be

$$M = \frac{12LWv^2}{15.5gR} \sin \theta \tag{92}$$

hence

$$S_b = \frac{Mc}{I} = \frac{0.77LWv^2}{gR}\left(\frac{c}{I}\right) \sin \theta \tag{93}$$

in which $\frac{c}{I}$ = inverse of section modulus; $\frac{10}{d^3}$ for round section, $\frac{6}{bh^2}$ for rectangular section, and $\frac{6H}{BH^3 - bh^3}$ for I section. See also any standard engineer's handbook.

The stresses due to column action and bending, equations 89 and 93, will both be a maximum near the center of the rod and must be found at several values of θ until their sum is a maximum. The sum of these stresses must not exceed the allowable stress in tension. For single-acting engines the allowable stress should not exceed one-third of the yield (elastic limit) stress; see S_y under equation 89.

When the rectangular or I section is used, the radius of gyration for transverse bending will be less than that in the plane of rotation. However, the pins restrain the bending in this direction and experience indicates that the column-end factor, m in equation 89, may be taken as 1. Using this value and making the column equally strong in both directions, the column equations indicate that the ratio of width to depth for the rectangular section should be approximately 0.7. In high-speed engines where the bending stresses are high, the ratio may be as low as 0.5. It is necessary to assume proportions for I-section rods which will provide sufficient stiffness to prevent local buckling even when the column and bending equations indicate sufficient strength. A satisfactory section will be obtained when the width is 0.6 to 0.7 times the depth, and the flange and web thicknesses are from 0.15 to 0.20 times the depth. Although the rod is assumed to be of uniform section when making the computations indicated, most rods are tapered so that the crank-pin end is 1.10 to 1.15 times the depth computed for the center section of the rod.

The stress where the tapered rod joins the piston-pin end should be investigated. The direct stress due to the piston pressure should be added to the bending stress in the same manner as the stresses at the center of the rod. The bending moment at any section of the rod is

$$M' = \frac{F_i x}{3L^2}(L^2 - x^2) \tag{94}$$

where F_i = inertia force from equation 91.
x = distance from the piston pin, inches.

hence

$$S'_b = \frac{M'c}{I} = \frac{F_i x}{3L^2}(L^2 - x^2)\frac{c}{I} \tag{95}$$

The area of the piston-pin end of the connecting rod will be the total force on the piston divided by the allowable stress for steel in alternate tension and compression. The following equation may be used to get the diameter

$$\frac{F}{S} = \frac{\pi d^2}{4}$$

$$S = \frac{4F}{\pi d^2} \tag{96}$$

and

$$d = \sqrt{\frac{4F}{\pi S}} \tag{97}$$

where d = diameter of piston-pin end of rod, inches.
F = maximum load on piston, pounds.
S = allowable alternate stress.

The size of the crankpin end of the connecting rod is discussed in the next article.

86. The Crankshaft. Crankshafts may be divided into two types, shafts with side cranks and shafts with center cranks. The shafts with center cranks constitute by far the majority and can be further subdivided into single-crank and multi-crank shafts. For multi-cylinder engines the construction of the crankshaft is intimately related to the firing order, valve timing, and the distribution of the charge. In choosing a crankshaft, three important things should be kept in mind: running balance, power distribution, and manufacturing ease.

Center crankshafts of medium and large engines generally have a bearing between each pair of cranks and a bearing on each end of the shaft. For automobile engines, in order to make the engine shorter and lighter, the number of bearings may be decreased. Hence a crankshaft for a four-cylinder engine may have either five or three or even two bearings; a crankshaft for a six-cylinder engine may have seven, four, or three; and so on. In V-type engines each crankpin serves two opposite cylinders, and in radial engines one crankpin serves all the cylinders, as shown in Figs. 49 and 50.

To obtain uniform rotation in multi-cylinder engines, the power impulses should be equally spaced. This consideration of power distribution, in addition to running balance, determines the conventional arrangements of crankshaft throws and firing orders, as shown in Fig. 127*a*. The engine cylinders are generally identified by number, the highest number being given to the cylinder nearest the flywheel. Viewed from the front, most engines rotate clockwise.

One of the desirable characteristics in a crankshaft is that the directional grain flow of the metal follow the contour of the section and be continuous throughout its length. Such a condition eliminates the stressing of the material transverse to the grain. Single piece forgings are preferable to built-up sections because of the better stress distribution at the changes in section. Failures, when they occur, are usually due to lack of adequate fillets at the crankpins and journals, or are the result of critical vibrations. The stresses developed in this manner are frequently more than any metal can be expected to withstand. In order to obtain the necessary rigidity in a crankshaft, the section must be large and hence the working stresses are low; consequently, the question of strength is not always a serious matter.

Bearing wear is one of the important considerations in selecting the type of material to use. A straight carbon open-hearth steel containing 0.45 per cent carbon fulfils all the requisite conditions quite satisfac-

TWO-STROKE CYCLE ENGINES		No. of cylinders	FOUR-STROKE CYCLE ENGINES	
Firing order	Arrangement of cranks		Arrangement of cranks	Firing order
1-2		2		1-2 1-2
1-2-3		3		1-3-2
1-4-2-3		4		1-2-4-3 or 1-3-4-2
1-4-3-2-5		5		1-3-5-4-2
1-4-5-2-3-6		6		1-5-3-6-2-4
1-6-2-4-3-5		6		1-4-3-6-2-5
1-6-4-7 2-5-3-8		8		1-5-2-6 8-4-7-3
1-8-6-4 2-7-5-3		8		1-6-2-8 4-7-3-5

FIG. 127*a*. Arrangement of cranks and firing order sequence for multi-cylinder engines.

torily for most industrial engines; clean, homogeneous material is more important than special composition. On some of the higher speed units a harder material may be desirable. In such cases the bearings may be

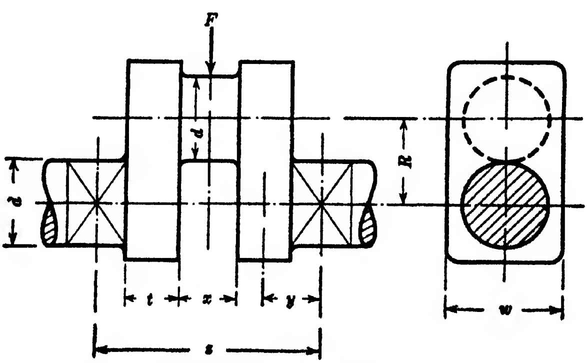


FIG. 127*b*. One section of a crankshaft for a multi-cylinder engine.

hardened by case-carburizing or nitriding; the latter, although more expensive, is the preferable method. Automobile- and aircraft-engine crankshafts are usually made of chrome-nickel steels having a tensile strength of 125,000 to 140,000 lb per sq in., elastic limit of about 100,000 lb per sq in., and an elongation of 15 to 20 per cent.

The simplest assumption in designing a crankshaft is that each section is a *beam* with the piston-pressure load imposed at its center and that the supports are at the middle of each crankshaft bearing. From basic considerations and referring to Fig. 127*b*,

$$S_b = \frac{Mc}{I} = \frac{Fz}{4} \times \frac{32}{\pi d^3}$$

but

$$F = \frac{\pi D^2}{4} \times p$$

hence

$$S_b = \frac{2D^2pz}{d^3} \tag{98}$$

and

$$d = \sqrt[3]{\frac{2D^2pz}{S_b}} \tag{99}$$

where d = diameter of crankpins and journals,[1] inches.
D = diameter of engine cylinder, inches.
p = maximum cylinder pressure, pounds per square inch.
z = center to center of crankshaft bearings; also center to center of cylinders, usually $1.5D$ to $2D$, inches.
S_b = allowable stress due to bending; 8000 lb per sq in. for 0.45 carbon steel, 12,000 for chrome-nickel steel shafts.

When the bearing is relatively long, the amount of deflection frequently becomes a more important consideration than the stress due to bending. The deflection (between crankarms) should not exceed 0.001 in. as computed by the simple beam equation

$$f = \frac{Fx^3}{48EI} \tag{100}$$

where f = deflection for length x, inches.
F = maximum load on piston, pounds.
x = distance between crankarms, inches, see Fig. 127*b*.
E = modulus of elasticity; 30,000,000 lb per sq in. for steel.
I = moment of inertia, $\frac{\pi d^4}{64}$.

[1] This is generally done when a journal bearing is used between two crankpin bearings. If fewer main crankshaft bearings are used, the diameter will be larger because of the increased load on each bearing.

In multi-cylinder engines the torque from each cylinder is transmitted along the crankshaft and subjects intervening crankpins to torsional shear. Thus the dimensions of the crankshaft should be checked for strength in torsion, especially for the Diesel engine with its nearly constant-pressure combustion. The maximum tangential pressure on the crankpin is about half the maximum cylinder pressure, and the torsional moment is

$$M_t = \frac{\pi D^2}{4} \times \frac{p}{2} \times R \tag{101}$$

where R = crank radius, half length of stroke, inches.

The section modulus for torsion is $\frac{I_p}{a} = \frac{\pi}{16} d^3$, I_p being the polar moment of inertia and a the cross-sectional area of the crankpin. Hence

$$S_t = M_t \times \frac{a}{I_p} = \frac{\frac{\pi D^2}{4} \times \frac{p}{2} \times R}{0.2d^3}$$

$$S_t = \frac{\pi D^2 pR}{1.6d^3} \tag{102}$$

and

$$d = \sqrt[3]{\frac{2D^2pR}{S_t}} \tag{103}$$

where S_t = allowable stress due to torsion; 4000 lb per sq in. for 0.45 carbon steel, 6000 for chrome-nickel steel shafts.

The bearing lengths of a crankshaft depend upon the allowable bearing pressures. The permissible pressures depend upon the maximum cylinder pressure, frequency of change of direction of the bearing pressure, material used, journal velocity, and method of lubrication. Table XVIII lists maximum allowable bearing pressures for typical internal-combustion engines using force-feed lubrication. The bearing metals are babbitt for the main journal and crankpin bearings and bronze for the piston-pin bearing. The length of bearing may be determined as follows:

$$S' = \frac{F}{A_p} = \frac{\frac{\pi D^2}{4} \times p}{dx} \tag{104}$$

$$x = \frac{\pi D^2 p}{4S'd} \tag{105}$$

where x = length of bearing, inches; $\frac{x}{d}$ for crankpin bearing 1 to $1\frac{1}{4}$,
for main bearing $1\frac{1}{4}$ to 2.
S' = allowable bearing pressure, Table XVIII.
A_p = projected area of bearing, square inches.
Other symbols as for previous equations.

TABLE XVIII

MAXIMUM ALLOWABLE BEARING PRESSURES WITH FORCE-FEED LUBRICATION

Type of Internal-combustion Engine	Main Journal Pressure, lb per sq in. of Projected Area	Crankpin Pressure, lb per sq in of Projected Area	Piston-pin Pressure, lb per sq in of Projected Area
Automobile and aircraft...	600–1800	1500–2500	2000–4000
Gas engine, 4 cycle.......	600–1000	1200–1800	1500–2000
Gas engine, 2 cycle.......	800–1000	1200–1500	1200–1800
Diesel engine, 4 cycle.....	500–800	1000–1500	1200–1800
Diesel engine, 2 cycle.....	400–600	1000–1200	1200–1600
High-speed Diesel, 4 cycle.	600–1200	1200–1800	1500–2000
High-speed Diesel, 2 cycle.	500–1000	1000–1500	1200–1800
Double-acting, 2 cycle....	600–1000	1200–1800	1500–2000

The moment-arm y (Fig. 127b) in bending is measured from the center of the crankshaft bearing to the neutral axis of the crankarm. Since the crankpin and journal bearing lengths are directly proportional to the diameter of the cylinder as well as the total length z between each pair of journal bearings, it follows that the space remaining for t, the thickness of the crankarm, must also be proportional to D and d; in general

$$t \geqq 0.4d \tag{106}$$

where d = diameter of crankpin, inches.
and for the width of the crankarm

$$w \geqq 2.5t \tag{107}$$

The dimensions of the crankarms calculated by equations 106 and 107 should be checked for strength in bending; in crankshafts for multi-cylinder engines the stresses due to torsional shear are relatively small and may be neglected. For single-cylinder engines, however, the torsional stresses must also be calculated,[2] as these stresses may be of considerable magnitude.

[2] "Applied Elasticity," Timoshenko and Lessells; "Combined Bending and Twist," Chapter VIII, First Edition, 1925, Westinghouse Technical Night School Press.

The stresses in the crankarms are more complicated with the crankshaft in the position of maximum turning moment than in any other position. The section where the crankarm joins the shaft, at A in Fig. 128, is the most dangerous as at that point there will be direct compression due to the radial force F_r, bending due to F_r, and bending due to F_t; also torsional shear stresses due to F_t, which may be neglected

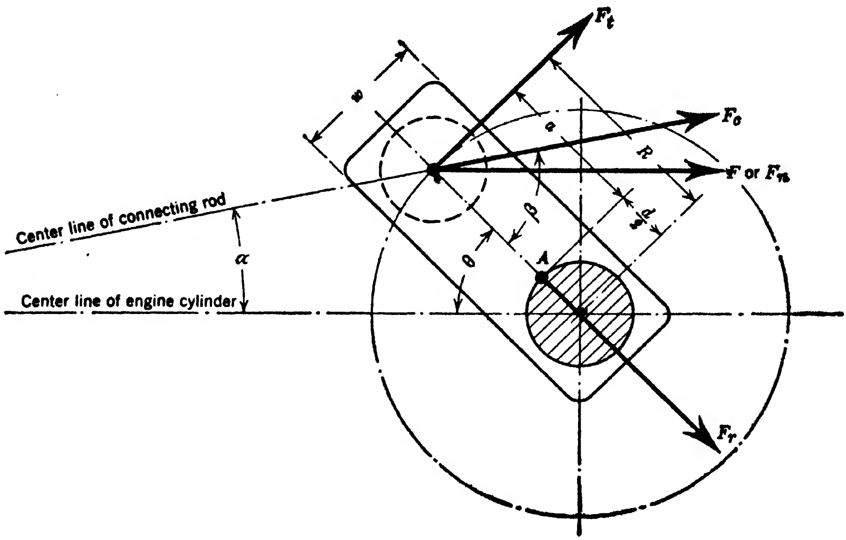


FIG. 128. Forces acting on a crankshaft at angle, 30 to 40 deg, of maximum torque.

in multi-cylinder engines. The direct compression will be, refer to Fig. 128,

$$S_c = \frac{F_r}{wt} \tag{108}$$

the bending stress due to F_t will be

$$S_b = \frac{6F_t \times a}{tw^2} \tag{109}$$

and that due to F_r will be, refer to Figs. 127*b* and 128,

$$S'_b = \frac{6F_r\left(\frac{x}{2} + \frac{t}{2}\right)}{wt^2} \tag{110}$$

The maximum stress will occur when the bending stresses found by equations 108 and 109 will both be compression. Hence the total maximum compressive stress is

$$S_{mc} = S_c + S_b + S'_b \tag{111}$$

The maximum stress for the crankarms should not exceed 8000 lb per sq in. for 0.45 per cent carbon steel and 12,000 lb for chrome-nickel steel shafts.

Example. Determine the total stress in one crankarm of a multi-cylinder 12 × 21 in. gas engine for the position of maximum tangential pressure, 15,400 lb when θ (Fig. 128) is 35 degrees, length of connecting rod $52\frac{5}{8}$ in., width of crankarm $6\frac{1}{4}$ in., thickness $4\frac{1}{4}$ in., and diameter of crankshaft $5\frac{1}{2}$ in. Length of each crankshaft bearing 10 in.

Solution.

$$\beta = 41° - 34'$$

$$F_r = \frac{F_t}{\tan 41° - 34'} = \frac{15{,}400}{0.8869} = 17{,}360 \text{ lb}$$

For one crankarm $$F_r = \frac{17{,}360}{2} = 8680 \text{ lb}$$

and $$F_t = \frac{15{,}400}{2} = 7700 \text{ lb}$$

$$S_c = \frac{8680}{6.25 \times 4.25} = 327 \text{ lb per sq in.}$$

$$S_b = \frac{6 \times 7700 \times 7.75}{4.25 \times (6.25)^2} = 2160 \text{ lb per sq in.}$$

Assume $x = d$, $$S'_b = \frac{6 \times 8680 \times \left(\frac{5.5}{2} + \frac{4.25}{2}\right)}{6.25 \times (4.25)^2}$$

$$= 2250 \text{ lb per sq in.}$$

thus $$S_{mc} = 327 + 2160 + 2250 = 4737 \text{ lb per sq in.}$$

This result is less than 8000 lb per sq in. allowable stress; hence the crankarm dimensions are ample.

87. Balancing of Reciprocating Parts. The reciprocating parts and revolving weights of an engine set up forces and moments that must be balanced if vibrations are to be minimized. The forces within the engine cylinder are resisted by the engine framing which is held in tension by the upward thrust of the cylinder head under combustion pressure and the downward thrust of the piston against the crank. The forces of acceleration and retardation, and the centrifugal forces, are called free forces because they are not self-contained within the engine mechanism and framing but are resisted only by the mass of the engine and its foundation unless other free forces are set in opposition to them.

The piston, piston pin, and the upper part of the connecting rod which is assumed as taking the motion of the piston, may be balanced approximately, at the dead centers, by a rotating weight attached to the crankshaft. At the centers the radially directed centrifugal force of the counterweight opposes the force required to stop the reciprocating parts and start them in the opposite direction. The angularity of the connecting rod produces other unbalanced forces and moments. The unbalanced forces produced by the angularity of the rod are not very large, but, as they occur twice during each revolution of the crankshaft, they are high in frequency.

The revolving forces and moments are produced by the rotating parts, which include the unbalanced portions of the crankarms, crankpins, the lower part of the connecting rod which is considered as taking the motion of the crankpin, and counterweights, if used. Fitting counterweights on the crankshaft is the usual method of improving the revolving force and moment conditions; counterweights are also used to improve unbalanced reciprocating forces, which may be ineffective or harmful. Near its half-stroke position the piston momentarily has the same velocity as the crankpin, and being neither accelerated nor retarded the piston exerts no free force on the engine frame. At this instant any counterweighting in excess of that required to balance the rotating parts exerts its full effect at right angles to the cylinder axis and becomes a free force. That is, anything added in opposition to the reciprocating weight is effective only in the direction of reciprocation and acts partially or entirely as a free force at all other positions tending to produce a radial disturbance with a component in a direction normal to the cylinder axis. At the 90-degree positions of the crank the free force caused by too much counterweighting is equal to the reduction of the free force at the dead center positions.

There are certain general principles that should be remembered in connection with balancing. The magnitude of the unbalanced forces is directly proportional to the weights and to the crank radius and varies as the square of the speed; this relation emphasizes the necessity of giving special consideration to balancing high-speed engines. Automobile and high-speed Diesel engines particularly require not only a number of cylinders and an arrangement of cranks that inherently give good balance, but a crankshaft completely balanced within itself; it must be balanced dynamically as well as statically. The connecting rods and pistons for each cylinder of the engine must weigh the same within close limits to avoid unbalance between the cylinders.

To cover thoroughly the mathematical and graphical methods used to attack the many problems connected with the balancing of the numerous types and possible arrangements of internal-combustion engines would require much more space than is available in this book. In general, there is a distinctive balancing problem to every machine which has a moving part. The subject of balancing is well covered by W. E. Dalby in his book "The Balancing of Engines." The reader will also find considerable material on this subject in the books "The

Gasoline Automobile" by P. M. Heldt and "Vibration Problems in Engineering" by S. Timoshenko.

88. The Flywheel. The flywheel is used to keep the speed fluctuations of the engine within desired limits. The power delivered by the piston fluctuates considerably during each revolution of the crankshaft, whereas the power delivered by the crankshaft should be continuously uniform. The flywheel acts as a rotating energy reservoir, absorbing the excess energy delivered to the crankpin during the first part of the engine stroke and releasing this energy during the latter part of the stroke, when the delivery of power has been reduced.

At the beginning of each stroke the reciprocating engine parts must be accelerated from zero velocity to the velocity of the crankpin in less than one-half the stroke. The force required to do this must be obtained from the piston or flywheel. The effect of the inertia of the reciprocating parts is to decrease the pressure on the piston pin at the beginning of the power stroke and to increase the pressure at the end of the stroke when the pressure in the cylinder has decreased because of the reduced gas pressure, thus tending to maintain a more uniform turning effort on the crankpin. The reciprocating parts whose inertia must be considered in a single-acting engine consist of the piston, piston pin, and one-third to one-half of the connecting rod. The inertia effect of the piston rings may be considered as counteracted by friction. The approximate equation for calculating the inertia effect at any crank angle is

$$p_i = \frac{12W_t v^2}{AgR}\left(\cos\theta + \frac{R}{L}\cos 2\theta\right) \tag{112}$$

where p_i = inertia effect, pounds per square inch of piston area; see Fig. 129.

W_t = total weight of reciprocating parts, pounds; piston and one-third to one-half weight of connecting rod; also piston rod and crosshead, if used.

v = crankpin velocity, feet per second.

A = piston area, square inches.

g = 32.17 ft per sec.^{2}

R = radius of crank, inches.

L = length of connecting rod, center to center, inches.

θ = crank angle, see Fig. 128.

Values of $\left(\cos\theta + \frac{R}{L}\cos 2\theta\right)$ are given in Table XIX for a number of "connecting rod to crank" ratios at the crank angles listed.

TABLE XIX

INERTIA FACTORS* = $(\cos\theta + \frac{R}{L}\cos 2\theta)$

Crank Angles, Degrees Forward	Connecting Rod Length ÷ Crank Radius; L/R						Crank Angles, Degrees Return
	3.5	4	4.5	5	5.5	6	
0	1.286	1.250	1.222	1.200	1.182	1.167	360
15	1.213	1.182	1.158	1.139	1.124	1.110	345
30	1.009	0.991	0.977	0.966	0.957	0.949	330
45	0.707	0.707	0.707	0.707	0.707	0.707	315
60	0.357	0.375	0.389	0.400	0.409	0.417	300
75	0.011	0.032	0.067	0.086	0.102	0.115	285
90	−0.286	−0.250	−0.222	−0.200	−0.182	−0.167	270
105	−0.506	−0.476	−0.449	−0.430	−0.415	−0.401	255
120	−0.643	−0.625	−0.611	−0.600	−0.591	−0.583	240
135	−0.707	−0.707	−0.707	−0.707	−0.707	−0.707	225
150	−0.723	−0.741	−0.755	−0.766	−0.775	−0.783	210
165	−0.719	−0.749	−0.774	−0.793	−0.809	−0.822	195
180	−0.714	−0.750	−0.778	−0.800	−0.818	−0.833	180

* Signs refer to forward stroke (toward crankshaft); use opposite sign for return stroke. Inertia forces as determined from this table must be subtracted from the cylinder pressures.

Graphical methods may be used to obtain the net turning effect of the crankshaft for each cycle, from which the quantity of energy to be alternately absorbed and released by the flywheel can be determined. In Fig. 129 the pressures from indicator cards, for an 8 in. × 10 in. single-acting four-cycle Diesel engine of two cylinders and making 850 rpm, have been plotted in sequence for the two cylinders as indicated. Positive pressures are above and negative pressures below the atmospheric line. The ordinates, from equation 112, of the inertia curve for the angles given in Table XIX as positive were plotted below the zero pressure line since they are retarding pressures, and for those given as negative the inertia pressures were plotted above the zero pressure line.

The graphical addition of the cylinder pressure and inertia pressure curves for the several cylinders gives the ordinates for the net effort curve, Fig. 129, which represents the net pressure delivered to the piston pin at the various positions of the piston. In the single-acting four-cycle engine of two cylinders the two pistons move up and down together, hence the net pressure on the piston pin at 20 per cent of the piston stroke for the expansion stroke of No. 1 cylinder (suction stroke for No. 2), can be determined as follows; refer to Fig. 129.

Cylinder pressure, No. 1 cylinder	=	+290 lb per sq in.
Cylinder pressure, No. 2 cylinder	=	− 3 lb per sq in.
Inertia pressure, No. 1 cylinder	=	− 70 lb per sq in.
Inertia pressure, No. 2 cylinder	=	− 70 lb per sq in.
Net pressure, No. 1 power stroke	=	+147 lb per sq in.

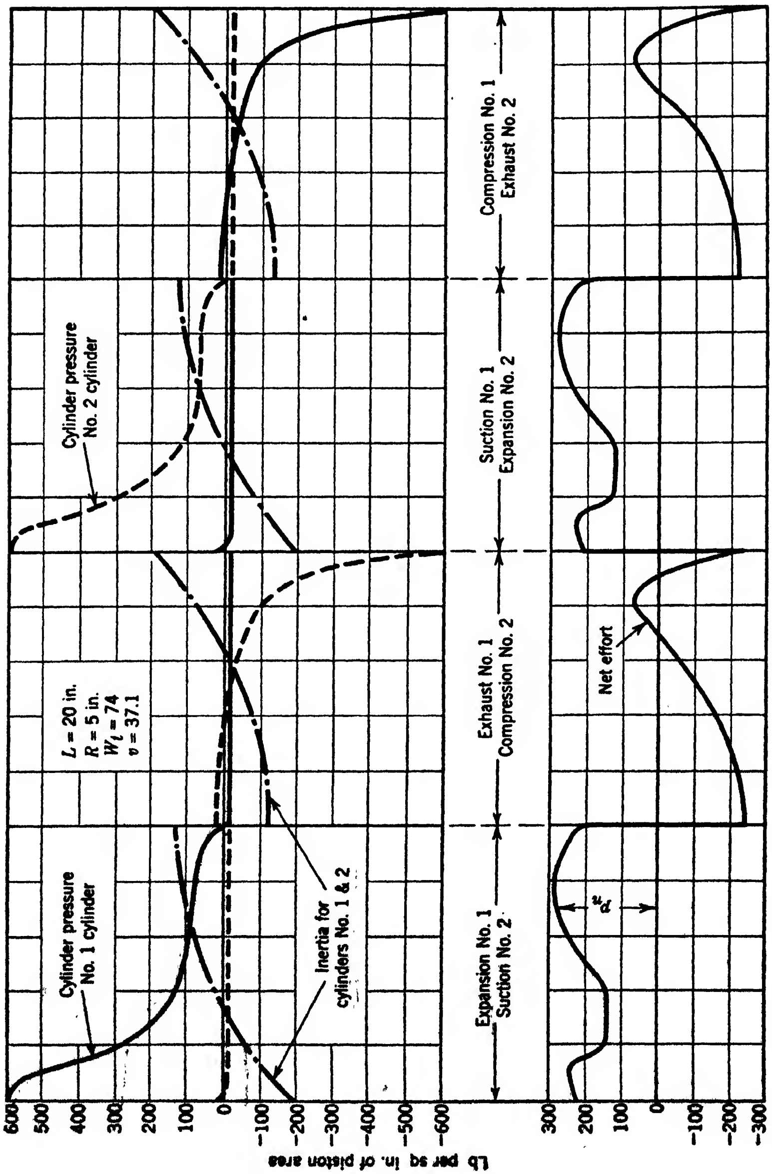


FIG. 129. Cylinder pressure (from indicator diagram), inertia, and net effort curves for four-stroke cycle, single-acting, two-cylinder Diesel engine.

Similar analyses may be made for other multi-cylinder two- and four-cycle engines in the many possible crankshaft arrangements. The net effort pressure curve for any multi-cylinder engine can be developed by superimposing an equivalent number of single-cylinder curves with impulses spaced at the firing intervals. The algebraic sum of the ordinates of the superimposed curves results in the net effort curve for each multi-cylinder arrangement; in general, two-cycle engines and multiple cylinders will give a more uniform net effort curve than four-cycle engines or few cylinders.

In Fig. 128, force F_n shows the direction of the net effort force in the line of piston travel; the turning force F_t acting tangent to the crankpin circle at the crankpin is

$$F_t = F_n \left\{ \sin\theta \left[1 + \frac{\cos\theta}{\left(\frac{L^2}{R^2} - \sin^2\theta\right)^{1/2}} \right] \right\} \tag{113}$$

where F_t = net turning effort on the crankshaft, pounds.

F_n = net effort on the piston, pounds; $A \times p_n$. The latter is obtained as explained above; see also Fig. 129.

Values of the part in braces are given in Table XX for a number of "connecting rod to crank" ratios at the various crank angles.

TABLE XX

TANGENTIAL FACTORS* $= \sin\theta \left[1 + \frac{\cos\theta}{\left(\frac{L^2}{R^2} - \sin^2\theta\right)^{1/2}} \right]$

Crank Angles, Degrees Forward	Connecting Rod Length ÷ Crank Radius; L/R						Crank Angles, Degrees Return
	3.5	4	4.5	5	5.5	6	
0	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	360
15	0.3305	0.3215	0.3145	0.3089	0.3054	0.3005	345
30	0.6249	0.6091	0.5968	0.5870	0.5791	0.5724	330
45	0.8524	0.8341	0.8195	0.8081	0.7988	0.7910	315
60	0.9925	0.9769	0.9641	0.9540	0.9458	0.9390	300
75	1.0402	1.0304	1.0228	1.0169	1.0121	1.0082	285
90	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	270
105	0.8915	0.9016	0.9091	0.9150	0.9198	0.9237	255
120	0.7383	0.7552	0.7680	0.7781	0.7863	0.7931	240
135	0.5614	0.5801	0.5946	0.6061	0.6155	0.6232	225
150	0.3751	0.3909	0.4032	0.4130	0.4209	0.4276	210
165	0.1871	0.1962	0.2032	0.2088	0.2123	0.2171	195
180	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	180

* Forward stroke is toward crankshaft. Piston-pin velocity = crankpin velocity *times* table value.

The rotative effort diagram, Fig. 130, is obtained by plotting the calculated tangential forces, F_t in equation 113, against crank travel. The values of p_n taken from the net effort curve, Fig. 129, are multiplied by the piston area, and the tangential factor, Table XX, for the respective crank angles to get F_t. The area of the rotative effort diagram divided by the length gives the mean torque T_m on the crankpin, as indicated in Fig. 130. If the net turning effort on the crankpin were constant, the flywheel would be unnecessary. The areas, Fig. 130, above or below the mean torque line indicate the necessity for a flywheel. If it is assumed that the power delivered by the crankshaft is constant, the areas above the mean torque represent an excess of energy developed which must be absorbed by the flywheel, and the areas below the line represent a deficiency of energy which must be released by the flywheel.

Usually, the maximum single area between the curve and the mean torque line represents the excess energy on which the weight of the flywheel is based. However, if after the largest positive area the next positive area is larger than the negative area between them, all within 360 degrees of crank travel, the maximum excess energy is represented by the sum of the two positive areas minus the negative area between them. The maximum excess energy in Fig. 130 is that represented by the area ΔE. This excess energy may be found as follows; refer to Fig. 130.

$$\Delta E = 2\pi R A_t \times \frac{s}{12l} \tag{114}$$

where ΔE = excess energy, foot-pounds per revolution.
R = radius of crank, inches.
A_t = excess area of rotative effort diagram, square inches.
s = scale of ordinates of rotative effort diagram, pounds per inch.
l = length of rotative effort diagram for one revolution, inches.

Example. Find the excess energy to be absorbed per revolution and per minute by the flywheel for the two-cylinder Diesel engine mentioned previously in this article and the data given in Figs. 129 and 130.

Solution.

$$\Delta E = 2\pi \times 5 \times 1.74 \times \frac{5300}{12 \times 3.25}$$

$$\Delta E = 7420 \text{ ft-lb per revolution}$$

$$\Delta E = 6{,}310{,}000 \text{ ft-lb per minute}$$

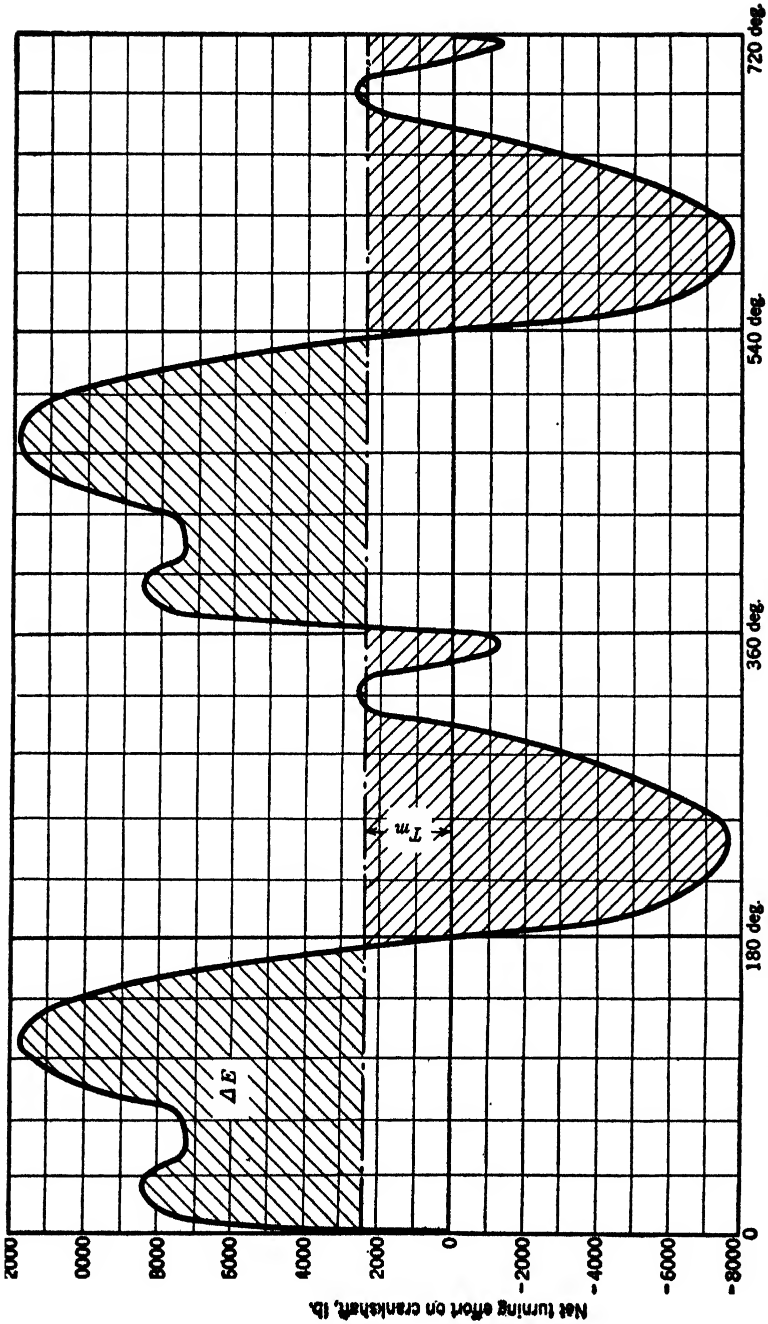


FIG. 130. Rotative effort diagram for four-stroke cycle, single-acting, two-cylinder Diesel engine.

The change of kinetic energy of the rim due to the maximum fluctuation of velocity from v_2 to v_1 ft per sec, or vice versa, is equivalent to the maximum fluctuation of energy, thus

$$\Delta E = \tfrac{1}{2}M(v_2^2 - v_1^2) = \frac{W}{2g}(v_2^2 - v_1^2) \tag{115}$$

where W = weight of flywheel rim, pounds.
v_2 = maximum velocity of rim, feet per second.
v_1 = minimum velocity of rim, feet per second.

Using the velocity in terms of the polar radius of gyration and the revolutions per minute of the flywheel, the change of kinetic energy is

$$\Delta E = \frac{4\pi^2\rho^2 W}{2g \times 3600}(N_2^2 - N_1^2)$$

$$\Delta E = \frac{\pi^2\rho^2 W}{1800g}(N_2 + N_1)(N_2 - N_1) \tag{116}$$

where W = weight of flywheel, pounds.
ρ = polar radius of gyration of flywheel rim, feet; $0.354\sqrt{d_2^2 + d_1^2}$, in which d_2 is the outside diameter and d_1 the inside diameter of the flywheel rim.
N_2 = maximum speed of flywheel revolutions per minute.
N_1 = minimum speed of flywheel, revolutions per minute.

The fluctuation in speed of the flywheel may be designated by a coefficient

$$K = \frac{v_2 - v_1}{v_m} = \frac{N_2 - N_1}{N} \tag{117}$$

where K = coefficient of speed fluctuation; depends upon type of engine and kind of service, 1/20 to 1/40 for engines not requiring close regulation, 1/50 to 1/100 for close regulation, and 1/100 to 1/200 for very close regulation, as for electrical drives.
v_m = mean velocity of flywheel rim, feet per second.
N = mean speed of flywheel, revolutions per minute.

Remembering that $(N_2 + N_1) = 2N$, the fluctuation of energy in the flywheel, from equations 116 and 117, is

$$\Delta E = \frac{\pi^2\rho^2 N^2 W K}{900g}$$

therefore

$$W = \frac{900\Delta E g}{\pi^2\rho^2 N^2 K} \tag{118}$$

From equations 115 and 117, the weight of the flywheel may also be expressed, in terms of the mean velocity of the rim, as

$$W = \frac{g\Delta E}{Kv_m^2} \tag{119}$$

For stationary engines the flywheels are usually made of cast iron; for high speeds they are sometimes made of cast steel, or occasionally a cast-iron rim with steel or cast-steel arms and hub; for high-speed multi-cylinder engines, a solid-web flywheel is generally used. The flywheel effect of the arms and hub is usually neglected because about 95 per cent of the kinetic energy is produced by the material in the rim; neglecting this effect is on the side of safety. The radius of gyration may be determined by assuming a safe maximum rim velocity and knowing the speed at which the engine is to operate. An old rule for the maximum velocity of cast-iron flywheel rims—" a mile a minute "—is frequently followed today, although up to 6000 ft per min may be allowed. For automotive and high-speed Diesel engines using carefully designed steel flywheels, rim speeds of 10,000 to 20,000 ft per min are being used.

The graphical method for determining ΔE requires considerable time; hence the designer frequently uses the following approximate analytical method. The average energy delivered each revolution by the engine shaft is

$$E_m = \frac{33{,}000 \times ihp}{N} \tag{120}$$

and the excess energy absorbed and returned by the flywheel per revolution is

$$\Delta E = K' \times E_m = K' \times \frac{33{,}000 \times ihp}{N} \tag{121}$$

where K' = coefficient of excess energy; see Table XXI for average values

ihp = indicated horsepower of engine.

Combining equation 121 with equation 119

$$W = \frac{K'}{K} \times \frac{33{,}000 \times ihp \times g}{Nv_m^2} \tag{122}$$

Approximate values of the coefficient of excess energy for flywheel design are given in Table XXI; it may be seen that the coefficient varies with the type, number of cylinders, and crank arrangement for the engines listed. It should be noted that the coefficients for six- and eight-cylinder engines are very low; hence flywheels are not necessary.

TABLE XXI

COEFFICIENTS* OF EXCESS ENERGY FOR FLYWHEELS

Number of Cylinders	Type of Internal-combustion Engine	Coefficient K'	
		Two-cycle Engines	Four-cycle Engines
1	Single-acting	1.00	2.40
2	Single-acting, cranks at 180 degrees	0.25	1.50
2	Single-acting, cranks at 360 degrees	....	1.00
3	Single-acting, vertical, 120 degrees	0.15	0.70
4	Single-acting, vertical, 90 and 180 degrees	0.09	0.18
6 or 8	Single-acting, vertical	0.02	0.08
2	Double-acting, twin or tandem	0.20	0.22
4	Double-acting, twin-tandem	0.08	0.09

* For Diesel engines, increase values of K' about 25 per cent.

Example. Determine the weight of the flywheel for the two-cylinder (75 ihp) Diesel engine mentioned previously in this article; assume a cast steel rim with an allowable rim velocity of 8000 ft per min and an allowable speed variation of 5 rpm from the mean speed. Use equations 118, 119, and 122.

Solution. $K = \dfrac{5 \times 2}{850} = \dfrac{1}{85}$, coefficient of speed fluctuation

$K' = 1.0 \times 1.25 = 1.25$, Table XXI, coefficient of excess energy

$D_2 = \dfrac{8000}{\pi \times 850} = 2.99$ ft outside diameter of flywheel

$\rho = 0.354\sqrt{D_2^2 + D_1^2}$, rim thickness is usually small as compared to flywheel diameter, hence it may be assumed that $D_1 = D_2$; $\rho = 0.5D_2 = 1.495$ ft

$\Delta E = 7420$ ft-lb, from previous example

$v_m = \dfrac{8000}{60} = 133.3$ ft per sec

(*a*) Using equation 118 and the excess energy as determined from Fig. 130

$$W = \frac{900 g \Delta E}{\pi^2 \rho^2 N^2 K} = \frac{900 \times 32.17 \times 7420 \times 85}{9.9 \times 2.23 \times (850)^2 \times 1} = 1145 \text{ lb}$$

(*b*) Using equation 119 and the excess energy as determined from Fig. 130

$$W = \frac{g \Delta E}{K v_m^2} = \frac{32.17 \times 7420 \times 85}{1 \times (133.3)^2} = 1140 \text{ lb}$$

(*c*) Using equation 122, approximate analytical method, with coefficients of excess energy and allowable speed fluctuation

$$W = \frac{K'}{K} \times \frac{33{,}000 \times \text{ihp} \times g}{Nv^2_m} = \frac{1.25 \times 85 \times 33{,}000 \times 75 \times 32.17}{1 \times 850 \times (133.3)^2} = 560 \text{ lb}$$

NOTE: For this engine, the analytical method (part *c*) did not check the more accurate graphical method (parts *a* and *b*) for flywheel weight calculations.

The dimensions of the rim of a flywheel may be calculated from the equation

$$W = 2\pi\rho \times 12 \times a \times b \times w \tag{123}$$

hence

$$a = \frac{W}{24\pi\rho b w} \tag{124}$$

where a = width of flywheel rim, inches.
b = thickness of flywheel rim, inches; $b = 0.5a$ to $1.5a$.
W = weight of flywheel, pounds.
ρ = polar radius of gyration, feet.
w = unit weight of the material; 0.26 lb per cu in. for cast iron, 0.28 lb per cu in. for steel.

The corrected outside diameter of the rim in inches is

$$D_2 = (2\rho \times 12) + b \tag{125}$$

The diameter of the flywheel hub may be made from two to two and one-half times the shaft diameter. The length of the hub should be equal to about two shaft diameters.

The arms of a flywheel should be proportioned to transmit the maximum torque of the engine. In general, the fact that the arms do not extend to the center of the shaft is neglected. If the crankshaft diameter has not been enlarged to provide additional stiffness, then for elliptical arm sections, assuming the same factor of safety in shaft and arms,

$$\frac{\pi d^3}{16} S_t = n \frac{\pi e^2 f}{32} S_a \tag{126}$$

where d = diameter of shaft, inches.
n = number of arms in flywheel, usually 6.
e = width of flywheel arms at hub, inches.
f = thickness of flywheel arms at hub, inches; usually $f = 0.5e$.
S_t = allowable stress (torsion) of shaft; about 4000 lb per sq in. for steel.
S_a = allowable stress (bending) of arms; about 4000 lb per sq in. for cast iron and 8000 lb per sq in. for steel.

Substituting above values in equation 126, and simplifying,

$$\text{for six arms, using cast iron, } e = 0.94d \tag{127}$$

$$\text{for six arms, using steel, } e = \tfrac{2}{3}d \tag{128}$$

The load reaction at the hub of the flywheel produces a bending moment on the arm at the rim equal to that at the hub; this consideration would make the arm dimensions the same at the rim and hub. In practice, the arm section at the rim is generally made two-thirds to three-fourths the dimensions at the hub.

Example. Find the rim, arm, and hub dimensions for the 75-hp two-cylinder Diesel engine mentioned previously in this article. Assume thickness of rim to be 75 per cent of the width and use six elliptical arms. The diameter of the shaft is 6 in.

Solution.

Rim: $a = \dfrac{W}{24\pi\rho bw}$

$$a = \frac{1140}{24\pi \times 1.495 \times 0.75a \times 0.28}$$

$a = \sqrt{48} = 6.93$ in.; use 7 in.

$b = 0.75 \times 7 = 5\frac{1}{4}$ in.

Use 7 in. width $\times$ $5\frac{1}{4}$ in. depth flywheel rim.

$D_2 = (2 \times 1.495 \times 12) + 5.25 = 41\frac{1}{4}$ in., corrected outside diameter of flywheel rim

Hub: $D_h = 2 \times 6$

$D_h = 12$ in., outside diameter of flywheel hub

$L_h = 2 \times 6$

$L_h = 12$ in., length of flywheel hub

Arms: $e = \frac{2}{3} \times 6$

$e = 4$ in., width of elliptical arms

$f = 0.5 \times 4$

$f = 2$ in., thickness of elliptical arms

Flywheels are commonly made in one of the following ways: cast in one piece; cast in halves with planed joints; assembled from separate hubs, arms, and rim segments; web and rim unit fastened by bolts to a separately steel-forged hub; and disk-shaped, with or without rim. Small flywheels (less than 8 ft. in diameter) are frequently cast in one piece; sometimes the hub is split to relieve shrinkage strains. After being bored it is clamped to the shaft by hub bolts. Casting the flywheel in halves is good construction and is used for large wheels. It is better to have the split at the arms and not between the arms. Such wheels are less likely to be broken in transportation, and no doubt are stronger when in use. The assembled flywheel made from separate parts gives a minimum of internal stress; it also permits ease of sh'pment.

Some marine engines use flywheels made with a web-rim unit bolted to a separate steel hub, resembling automobile flywheels. The disk-shaped flywheels of automobile and other high-speed engines are generally machined all over; the outside diameter is made about three times the stroke of the engine. The clutch, which is usually built integral with the flywheel, and the heavy crankshaft used on modern high-speed engines add materially to the effective flywheel capacity. In connection with the flywheel capacity of the automobile engine the inertia effect of the moving car itself must be taken into account. The flywheel for high-speed engines generally serves as the driven gear of the starter drive and as one member of the friction clutch; these two auxiliary functions largely determine its shape.

89. Engine Frame and Crankcase. The requirements of engine frames differ considerably, depending upon the type of engine and class of service. In large heavy-duty engines, there must be sufficient mass to provide stability and absorb vibration. The chief consideration in the design of a frame and crankcase for a multi-cylinder engine is to provide sufficient depth to resist the oscillation moments produced by the explosion and inertia forces. This bending moment is not severe in so far as it affects the structural strength of the frame, but it may be of sufficient magnitude to produce vibration at certain critical speeds. In order to prevent vibration it is necessary to design the crankcase (crankshaft supports) to be rigid enough to resist bending; consequently from a structural point of view, it will have a large factor of safety. Since rigidity is a function of size and shape, any material which casts well and possesses reasonable physical qualities is suitable. To maintain rigidity against bending and torsion it is desirable that the crankcase shall act as an integral part of the engine frame and cylinder block; the use of through-bolt construction, Figs. 81, 131, and 132, aids in providing this rigidity.

The type-A frame, so called because its transverse section is shaped like the letter A, is one of the earliest forms of engine frames and is still used for some low-speed oil and gas engines of medium size. Each cylinder rests on its own legs, and is not directly connected to the other cylinders. The legs of the cylinder are cast in one piece with the cylinder jacket into which the liner is fitted. This construction reduces machining and fitting operations.

The type-B or box frame is illustrated in Figs. 131 and 132. ~~Sometimes~~ the box or girder construction of the frame and crankcase is depended upon to transmit the stresses from the cylinder to the engine base. A better construction is to provide steel rods, Figs. 131 and 132, for this purpose; in this design the frame becomes practically a spacer between the cylinder and the foundation. However, the casting must be properly ribbed, as shown, because the stresses produced by tightening down the through bolts may produce serious distortion. If the cylinders are cast separately, they are fastened either by a flange with studs or by passing the tie bolts through each corner of a deep flange of hollow section cast as a part of the cylinder jacket. In large engines the frame is divided into two or more castings, sometimes a separate casting under each

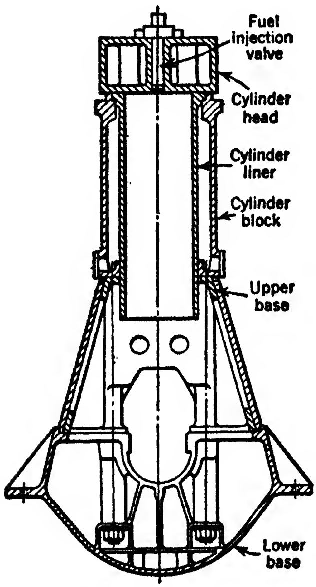


FIG. 131. Crankcase and main frame of two-piece box girder type, for multi-cylinder Diesel engine.

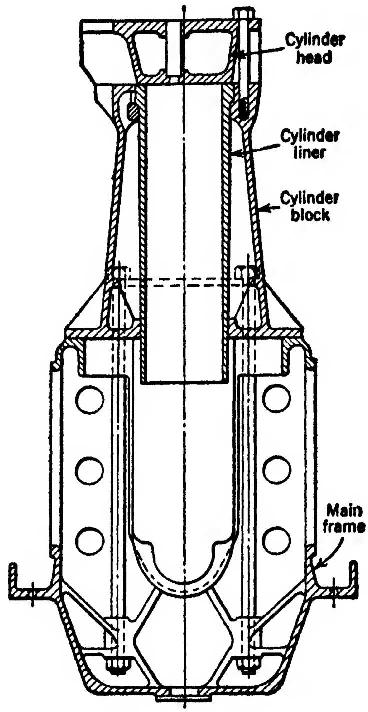


FIG. 132. Crankcase and main frame of one-piece box girder type, for multi-cylinder Diesel engine.

cylinder, see Fig. 133, the separate castings being bolted or welded together and forming a continuous box frame of great strength and rigidity.

The lower base or bed plate in Fig. 134 is divided into separate compartments by heavily ribbed bridges which carry the main bearings. The seats for these bearings are machined at one setting, assuring accuracy and perfect shaft alignment. The bearing shells are of the two-part type, made of cast steel and lined with babbitt which is spun in place. Figure 133 shows the lower base with the crankshaft in place and some of the columns which form the box framing and support for the cylinder block. This through-bolt construction assures rigidity and relieves the major castings of tensile stresses.

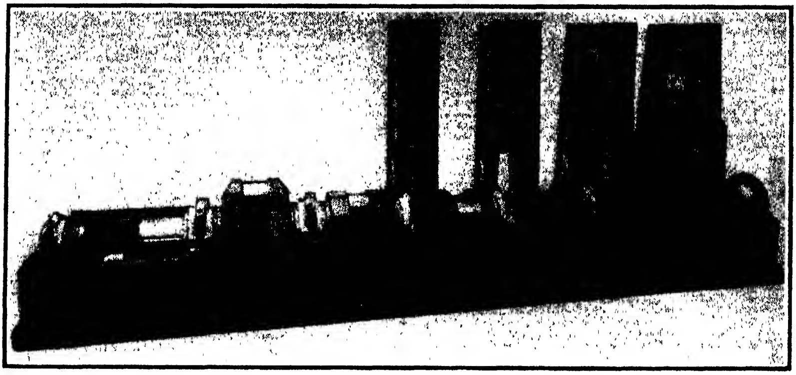


FIG. 133. Lower base (bedplate) with crankshaft and several of the "A" frames in place.

In general, it is impossible to analyze with any degree of accuracy the effect of the various combinations of stresses present in many engine

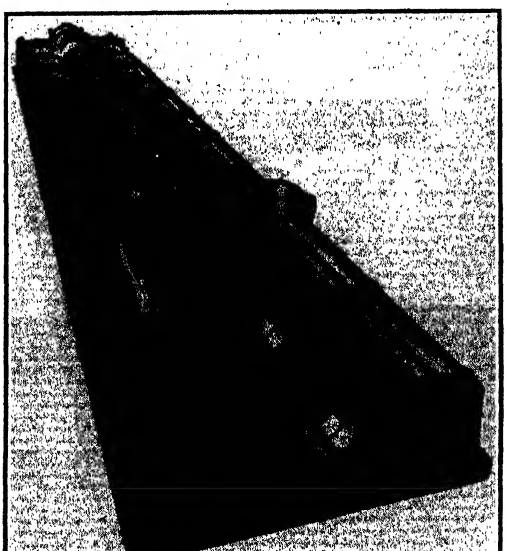


FIG. 134. Lower base (bedplate) of an eight-cylinder engine. The third bearing has the lower shell and bearing cap removed.

frames and crankcases. For this reason but little analysis is usually attempted and the design is based on experience; the main problem in designing the frame and crankcase unit is to make it rigid enough by proper ribbing and fastening. The size of the through bolts required for each cylinder in a design like Fig. 132 is usually determined as follows.

Example. Determine the size of the through bolts, two per cylinder, for a 17×25-in. Diesel engine with a maximum cylinder pressure of 600 lb per sq in. Assume a maximum working stress of 12,000 lb per sq in., and basing the design on applied load only.[3]

Solution. $P = \frac{\pi D^2}{4} \times p = \frac{\pi \times 289}{4} \times 600 = 136{,}180$ lb

$a = \frac{P}{S} = \frac{136{,}180}{12{,}000} = 11.35$ sq in. total area at root of threads for two through-bolts per cylinder

$d = \sqrt{\frac{5.675 \times 4}{\pi}} = 2.69$ in. root diameter, each bolt

$d' = 3$ in. outside diameter; see table of screw thread dimensions in M. E. Handbook

PROBLEMS

1. Find the wall thickness of a 12×17-in. gas-engine cylinder (cast iron) with a maximum internal pressure of 280 lb per sq in. (*a*) Use thin-shell equation. (*b*) Use thick-cylinder equation. (*c*) Estimate the jacket wall thickness. (*d*) Approximate the width of the water space. (*e*) Determine the thickness of semi-steel cylinder head.

2. A $5\frac{3}{4} \times 5\frac{3}{4}$-in. airplane engine employs a forged steel cylinder-barrel and has a maximum cylinder pressure of 450 lb per sq in. Calculate the thickness by using: (*a*) thin-shell equation; (*b*) thick-cylinder equation; (*c*) using the flat-plate equation, find that the thickness of the aluminum-alloy cylinder head.

3. (*a*) Approximate the wall thickness of a cylinder liner (cast steel) for a Diesel-engine of 18-in. diameter. Use a maximum cylinder pressure of 500 lb per sq in. (*b*) Determine the thickness of a cast-steel flat-plate head for this engine.

4. (*a*) Determine the outside diameter of the cylinder and its water jacket for an automobile engine with a cylinder diameter of $3\frac{1}{4}$ in. and a maximum cylinder pressure of 380 lb per sq in. (*b*) Assuming the cast-iron cylinder head as a flat plate, find its thickness.

5. Determine the dimensions and make full-size drawing of an inlet valve, as indicated in Fig. 122, for a $3\frac{1}{4} \times 4\frac{1}{4}$-in. automobile engine with a mean gas velocity through the valve of 180 ft per sec at 1600 rpm. Assume a volumetric efficiency of 85 per cent and the lift one-sixth of the valve diameter.

6. Calculate the mean velocity through a 2-in. inlet valve port for a $3\frac{7}{16} \times 4\frac{5}{16}$-in. automobile engine running at 3800 rpm. Neglect area of valve stem.

7. Find the ratio of the valve lift to valve diameter when the area of the valve opening equals the area of the port, neglecting area of the valve stem. Use a 45-degree seat. *Ans.* 0.3 to 1.

[3] For more detailed information on bolt design, see "Design of Machine Members," A. Vallance, Chapter VII, First Edition, 1938, McGraw-Hill Book Co., Inc.

8. The inlet port diameter of a four-cycle engine is one-half the cylinder diameter; the piston displacement is 500 cu in. per stroke. Calculate and tabulate the mean gas velocities through the inlet port for stroke-bore ratios of $\frac{1}{2}$, $\frac{3}{4}$, 1, $1\frac{1}{4}$, $1\frac{1}{2}$, and 2 for an engine speed of 900 rpm. Plot a curve showing mean gas velocities (ordinate) and stroke-bore ratios. For good charging efficiency, is a large or small stroke-bore ratio preferable?

9. Same as problem 8, using a valve to cylinder-diameter ratio of 0.60, volumetric (charging) efficiency of 0.85, and a speed of 1600 rpm for the various stroke-bore ratios.

10. Find the velocity of the air through the inlet valve, at a crank angle of 90 degrees for a 10×16-in. four-cycle oil engine running at 250 rpm. The valve diameter is $3\frac{3}{4}$ in., valve seat is at 45 degrees, and the valve lift is 1 in. The ratio of connecting-rod length to crank is $4\frac{1}{2}$ to 1; assume full piston displacement.

Ans. 8720 ft per min.

11. Determine the minimum valve area and the velocity, at a crank angle of 90 degrees, through the inlet valve of a six-cylinder $3\frac{1}{2} \times 4\frac{1}{4}$-in. automobile engine developing 18 hp per cylinder at 3400 rpm. The valve diameter is $1\frac{1}{2}$ in., valve seat width $\frac{1}{8}$ in., valve seat angle 45 degrees, and valve lift $\frac{1}{4}$ in. Assume 75 per cent cylinder charging efficiency. *Ans.* 29,750 ft per min.

12. Determine graphically and plot a valve lift diagram (instantaneous valve lift on a crank-angle base) for a tangential cam with a $\frac{3}{4}$-in. diameter roller follower, when the cam has a base circle diameter of $1\frac{1}{2}$ in., a nose radius of $\frac{1}{2}$ in., and a cam angle, from opening to closing, of 135 degrees. Use a valve lift of $\frac{7}{16}$ in. (maximum) at dead center. Assume no clearance between cam and valve gear.

13. The temperatures at the center and edge of a 4-in. cast-iron piston head are 820 F and 400 F, respectively. The thickness of the head is $\frac{3}{8}$ in. (*a*) Find the rate of heat flow to the piston head in Btu per square inch per hour. (*b*) What per cent of the heat supplied to the engine was absorbed by the piston head if the power per cylinder is 20 bhp, gasoline at 20,000 Btu per lb, and fuel supplied 0.60 lb per bhp-hr? Why is the result to part *b* less than 5 per cent, as given with equation 77?

14. The maximum cylinder pressure in problem 13 is 280 lb per sq in., check the piston-head design for strength by determining the stress with equation 80. If it exceeds the allowable stress, state two alternate designs that could be used; which would be preferable? Why?

15. For the piston of problem 13, estimate the thickness of the walls, length of piston, number of piston rings, and thickness of concentric rings.

16. Determine the dimensions and make full-size drawing of an aluminum-alloy piston for a $6\frac{1}{2} \times 8\frac{3}{4}$-in. Diesel engine with a maximum cylinder pressure of 500 lb per sq in. Use equation 97 for piston-pin size.

17. Determine the dimensions and make drawing of a solid, round section, carbon steel connecting rod for a 12×18-in. Diesel engine with a 5 to 1 ratio of connecting-rod length to crank. Maximum pressure, 800 lb per sq in. and speed, 300 rpm.

18. Design a solid, rectangular section, aluminum-alloy connecting rod for a $3 \times 3\frac{3}{4}$-in. automobile engine with a 4 to 1 ratio of L to R. Maximum cylinder pressure, 420 lb per sq in. Make depth of section twice the width. Use a steel bearing-cap at the crankpin end and an engine speed of 4200 rpm.

19. Design a nickel steel connecting rod with an I-section for the engine in problem 18. Make width 0.6 times the depth, the flange and web thickness 0.2 times the depth.

20. Determine the dimensions and make drawing, see Fig. 127*b*, of a single-throw chrome-nickel steel crankshaft for a $5\frac{3}{4} \times 5\frac{3}{4}$-in. radial aircraft engine. Maximum cylinder pressure, 500 lb per sq in.; use $z = 1.5\,D$ in equation 99. Check length of bearing for deflection, equation 100.

21. Find the dimensions and make drawing of one section of a nine-bearing chrome-nickel steel crankshaft for an eight-cylinder $3\frac{1}{4} \times 4\frac{1}{4}$-in. automobile engine. Maximum cylinder pressure, 450 lb per sq in.; use $z = 1.35\,D$ in equation 99.

22. (*a*) Find the dimensions indicated in Fig. 127*b* for a medium-carbon steel seven-bearing crankshaft for a six-cylinder four-cycle $15 \times 18\frac{1}{2}$-in. Diesel engine. Maximum cylinder pressure, 600 lb per sq in.; use $z = 1.5\,D$ in equation 99. (*b*) What would be the diameter of the crankshaft if it were made of chrome-nickel steel?

23. A 12×20-in. four-cycle, single-acting gas engine, single cylinder, 45 bhp (62 ihp), and center crank, has a connecting rod to crank ratio of 5 to 1 and runs at 240 rpm. The weight of the reciprocating parts is 3 lb per sq in. of piston area. Find the inertia effect in pounds per square inch of piston area at each end of the stroke; also determine the crank angle for zero inertia forces.

24. (*a*) Plot the expansion curve of an indicator card taken from the gas engine of problem 23, using the following pressure data: 0 stroke, 150 lb per sq in.; 2.6 per cent, 320 lb; 10 per cent, 250 lb; 20 per cent, 186 lb; 30 per cent, 145 lb; 40 per cent, 118 lb; 50 per cent, 97 lb; 60 per cent, 85 lb; 70 per cent, 70 lb; 80 per cent, 62 lb; 90 per cent, 54 lb; 100 per cent stroke, 27 lb per sq in. Use 5-in. base line for stroke and 100 lb per sq in. pressure per in. for ordinate scale. (*b*) On the same sheet calculate and plot the inertia effect of the reciprocating parts during the expansion stroke. (*c*) Combine curves *a* and *b* to plot the net effort curve; see Fig. 129. Use 110 lb per sq in. compression pressure and $n = 1.30$ for plotting compression curve.

25. (*a*) Calculate and tabulate the net tangential (turning) effort from the results obtained in (*c*) of problem 24 for various positions of the crank. Plot the rotative effort diagram, Fig. 130, using a vertical scale of 4000 lb per in. and a horizontal length of 5 in. (*b*) From the net rotative effort diagram, find the mean torque and the maximum excess energy ΔE. (*c*) Calculate the weight of one of two (alike) cast-iron flywheels. Use a coefficient of speed fluctuation of $\frac{1}{20}$. (*d*) Also find the approximate weight of one of the two flywheels by the analytical method; equation 122. Use 4200 ft per min. flywheel rim velocity.

26. Determine the dimensions and make drawing of one of the cast-iron flywheels from (*c*) of problem 25, showing rim, hub, and arms. Make the rim thickness one and one-half times the width and do not make the outside diameter more than 6 ft because of structural limitations.

27. Calculate the weight of a cast-steel flywheel for a 10×14-in. four-cycle oil engine which operates at 300 rpm. The torque diagram shows a maximum excess energy of 34,600 ft-lb and the coefficient of speed fluctuation is $\frac{1}{50}$. Use a rim velocity of 8000 ft per min.

28. Determine the major dimensions of the flywheel in problem 27. Use a square rim-section.

29. A vertical 100-bhp four-cycle Diesel engine of three cylinders, operating at 600 rpm, has a cast-iron flywheel. The outside diameter is 54 in.; the rim is 4 in. wide and 6 in. thick. Find the coefficient of speed fluctuation. Assume a mechanical efficiency of 80 per cent.

30. Determine the probable variation of speed for the engine of problem 29.

CHAPTER XIII

ENGINE DETAILS. ACCESSORIES AND AUXILIARIES

Many of the major engine parts have been described on previous pages of this book with the general description of the various types of engines. Parts that have already been explained as to uses, material, and design include cylinders, valves, pistons, connecting rods, crankshafts, frames, governors, carburetors, fuel-injection nozzles, and fuel pumps.

The internal-combustion engine constitutes the hub about which is grouped a variable quantity of devices and structures to aid its performance. These parts may be called accessories and auxiliaries; in general they serve to control (improve) operating conditions and insure continuity of service. All small devices which are directly connected (integral) with an engine to facilitate its operation and maintenance are called accessories. Accessories usually provided include: starting devices, ignition systems, governors, lubricating oil systems, cylinder-cooling methods, pyrometers, and air-purification methods. Buildings, engine foundations, water-cooling circulating systems, fuel-oil supply systems, intake air silencer-filters, exhaust silencers, and waste-heat recovery devices are frequently called auxiliaries because they are larger indirect parts of the complete engine installation.

90. Starting Devices. An internal-combustion engine must do the work of drawing in and compressing its charge before power is developed in the cylinder; hence some special device is required to start it. This involves the use of power from an external source.

If the engine is small and can be disconnected from its load, it may be started by turning the flywheel by hand for a few revolutions, until an explosive mixture is available and ignited. When an engine is started in this manner, the ignition should be retarded to occur when the piston is at or beyond the dead center; otherwise the high pressure of the explosion, acting on the piston before the crank has reached dead center, will overcome the inertia of the engine and reverse its direction of rotation. Automobile and high-speed Diesel engines are occasionally started by hand cranking, but more generally by an electric motor. The electric starting systems used with small Diesel engines are replicas of those used for automobile engines.

To facilitate starting at low temperatures an electric resistance heating unit may be installed in the intake manifold, glow plugs may be used in the combustion chamber, Figs. 82, 83, and 84, or punk sticks may be inserted in plugs with breech lock threads. The electric heating apparatus should preferably be installed outside the combustion chamber; an electric preheater requires from 500 to 2000 watts for several minutes prior to starting. Electric starting motors are of the series type; they should be capable of supplying a rolling torque at the crankshaft equal to twice the normal full-load torque of the engine when cranking at 200 rpm. The starting motor has a pinion on its shaft which can be engaged and disengaged with gear teeth cut in the flywheel rim; the ratio of motor pinion to flywheel ring gear is usually about 10 to 1. The electric motor is supplied with current from a storage battery. The battery is recharged by a generator which is always in gear with the engine.

A steam engine is started by turning steam under pressure into the cylinder. An internal-combustion engine cannot be started in this way, unless compressed air is used. Large engines are usually started with compressed air, in the same way that steam is used in a steam engine. Compressed air from a container, called "the air-starting bottle," is admitted to the cylinder at a pressure of about 250 lb per sq in. through a special air-starting valve in the cylinder head; see Figs. 34, 65, and 87. This valve is operated by a cam on the camshaft, timed to open when the piston is at top center, and remains open 45 to 75 degrees after top center. An air-starting valve may be fitted on all or on only a certain number of the cylinders in a multi-cylinder engine, as required.

Since it is not desirable that the air-starting valves function while the engine is running on fuel, some mechanical means must be provided for engaging and disengaging the operating gear, between the valve and its cam, at will. The air pressure in the air-starting bottle is obtained from the engine air compressor or from a separate auxiliary air compressor. This insures an abundant supply of starting air at all times.

The action of the air-starting system is as follows: The internal-combustion engine being in all respects ready to start, the operating gear between the air-starting cam and valve is thrown into the operating position. Air from the starting bottle is turned into the air-starting line and the air flows to the faces of the starting valves on the respective cylinders. Those cylinders whose pistons are between the top center and 75 degrees on the power stroke will receive air at a pressure of about 250 lb per sq in.; for some engines an air pressure of 100 lb per sq in. is sufficient for starting. This air pressure will cause the piston to move and the engine crankshaft to turn over. As soon as the engine begins to turn over, air is cut off from half the cylinders and fuel is substituted.

When these cylinders begin to fire, the air is cut off from the two cylinders and fuel is supplied to them. The engine is now operating with all cylinders firing and delivering power to the crankshaft.

91. Ignition Systems. As already discussed, the problem of increasing the heat energy of a mixture of fuel and air in the cylinder of an internal-combustion engine demands that after the mixture has been compressed it must be ignited. The fuel will then combine with the oxygen supplied with the air and impart the increased pressure, resulting from this heat, to the piston. The ignition should be so timed as to occur at the proper point of the cycle so far as the fuel mixture is concerned, and at the proper point of the stroke of the piston so far as the engine is concerned. With the Otto cycle this ignition should be so timed that the combustion will be complete, or nearly so, when the power stroke begins. In Diesel engines the ignition of the fuel is brought about by the high temperature of the air charge in the working cylinder at the end of the compression stroke.

The time of ignition varies with operating conditions of the engine and cannot be set by any fixed rule. In general, the ignition system is so arranged that the point of ignition may be varied between certain limits. The conditions which generally affect the time of ignition are: the compression pressure, the kind of fuel used, the quality of the mixture, and the speed of the engine.

Since a high compression pressure increases the rate of combustion, the higher the compression pressure the later may be the ignition for a given fuel. As some fuels burn more rapidly than others, these will then require a later ignition than the slower-burning fuels. When an engine is running with a lean mixture, combustion will be slower, and the ignition should be correspondingly earlier. A mixture which is too rich would require an early ignition for the same reason. A perfect fuel-air mixture has the most rapid combustion and requires the latest ignition.

The higher the speed of an engine, the earlier should be the ignition. This is explained by the fact that it takes an appreciable time for combustion to occur after the fuel is ignited. At high speeds, the ignition must be advanced in order to have the maximum pressure occur at the beginning of the power stroke. Some engine manufacturers make arrangements for advancing or retarding the point of ignition automatically by a centrifugal governor, as the engine speeds up or slows down. As an engine slows down, owing to the increase of load, the period of time during which combustion may take place is lengthened; and if the ignition is advanced for high speed, it occurs too early for the reduced speed and the maximum pressure is developed before the piston reaches dead center. When this trouble occurs, a sharp knocking in the cylinder is generally heard, as described in Art. 25.

In early forms of the Otto engine the ignition of the compressed charge was effected by carrying a flame to the mixture within, through a narrow port in a slide valve, from a gas jet that was kept burning outside the cylinder. A later and more successful form was tube ignition, which consisted of a short closed tube (metal or porcelain), extending into the end of the cylinder and maintained at a bright red heat by a flame directed on its outside surface. A portion of the explosive mixture was allowed to enter the tube from the cylinder at the time when it should be fired. In most cases the ignition tube was used in conjunction with a "timing valve" which determined the instant at which explosion should occur by being opened to allow a portion of the compressed explosive mixture to enter the ignition tube. In some gas engines the "timing valve" was omitted and the ignition tube was always in free communication with the cylinder, the instant of firing being determined only by the compression of some of the explosive mixture into the tube. This type of ignition was largely used for gas engines, but it is now entirely displaced by some form of electric ignition. For this purpose a spark plug is provided, with an insulated terminal from which a spark or succession of sparks is discharged at the desired instant by the use of a battery and induction coil or a high-tension magneto. An alternative, now little used, is to produce a low-tension spark by the mechanical breaking of a contact within the cylinder, in a circuit having a large amount of self-induction.

The types of electric ignition may be classified as low- and high-tension, or make-and-break and jump-spark. Make-and-break ignition is generally used when low-tension current is used at the igniter, and the jump-spark type when high-tension current is used at the igniter. The source of electric current is generally a storage battery or magneto.

With the low-tension ignition system low-voltage current is used at the igniter; the average tension is 6 to 50 volts, and the resistance is such that 2 to 5 amperes flow when the circuit is closed. The igniter is located in the cylinder head and is a part of the electrical circuit, as indicated in Fig. 135. The igniter consists of a cast-iron body fitted with two steel spindles *A* and *B*. *B* is a fixed spindle, separated from the body of the igniter by insulating washers *W* at each end. It is connected by a flexible cable to the positive side of a low-tension battery or to a low-tension magneto.

Spindle *A*, Fig. 135, has a hammer head at the end inside the cylinder and is free to rotate. Leakage of gas is prevented by the shoulder formed on the spindle and this forms a tight joint in the corresponding recess in the igniter. Normally, hammer head *A* rests on the fixed spindle *B* and the electrical circuit is completed through the body of the engine. When ignition is desired, the lever *E* is tripped and carries with it the head *A*, abruptly breaking the contact and causing an electric spark to form as *A* moves away from *B*. This electric spark ignites the charge in the cylinder, if the charge is at the proper condition of mixture and compression. The advantages of the make-and-break system are hot spark insuring good combustion and little trouble with insulation; the disadvantages are moving mechanism required in the cylinder and the possibility that the points of contact may become fouled or wear away.

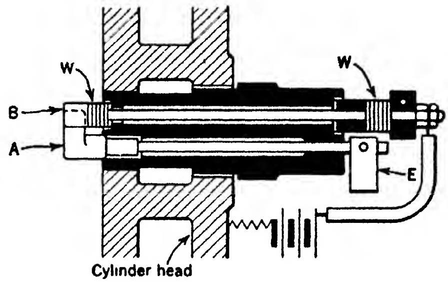


FIG. 135. Hammer-break igniter plug.

All automobile engines and many other engines operating on the Otto cycle employ the high-tension (jump-spark) ignition system, as shown diagrammatically by Fig. 136. An electric current of low voltage, generally 6 volts, is used with this type of ignition and is supplied by a storage battery *B*. The generator *G*, driven by the engine, is used as an auxiliary source of current when the engine is in operation, the generator potential (voltage) depending on the engine speed. When this potential is less than that of the battery, the cut-out *F* automatically cuts the generator out of the circuit. The ammeter *A* indicates the amount and direction of flow of the current to or from the battery.

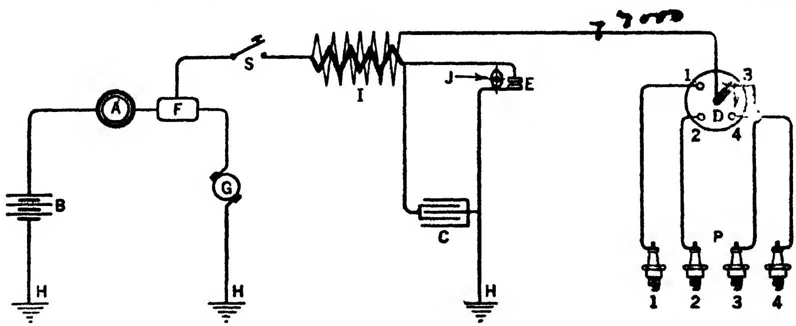


FIG. 136. Schematic diagram of a high-tension (jump-spark) ignition system as applied to a four-cylinder automobile engine.

The ignition switch is shown at *S*. The current flows through the primary winding of the induction coil *I* to the breaker points *E*. A condenser *C* is connected across the terminals of *E*; this prevents sparking at *E* and speeds up the electro-magnetic change in the coil. The primary winding of the coil consists of a comparatively few turns of coarse wire around an iron core. Around this is wrapped the secondary winding, consisting of a large number of turns of very fine wire. The breaking of the primary circuit at E by cam J causes a change in the magnetic field and this induces a momentary current of very high potential (about 20,000 volts) in the secondary winding. This high-tension current passes to distributor D and thence to the individual spark plugs P which are screwed into the cylinder, as shown in Figs. 39, 43, 49, and 62.

Most airplane engines, Fig. 48, and some other engines, Fig. 62, operating on the Otto cycle have magnetos as sources of electric current for ignition purposes; magnetos are used primarily because of their simplicity, accessibility, and dependability. A diagram of the electric and magnetic circuits of the Scintilla magneto is shown in Fig. 137. The rotating magnet 1 has four poles, two N and two S poles joined as

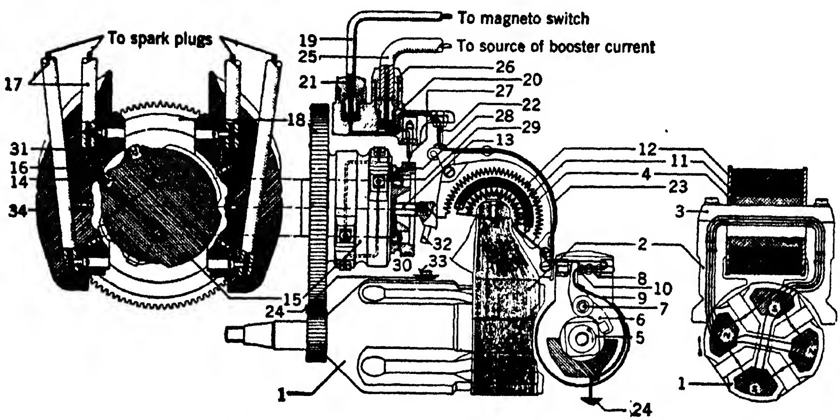


FIG. 137. Diagram of the electric and magnetic circuits in a Scintilla magneto.

shown on the right (end view) of Fig. 137. The rotating magnet 1 revolves between the laminated pole pieces 2 producing an alternating field in the core of the coil 3. When the current reaches its maximum the breaker cam 5 causes the breaker lever 6 to turn on its pivot 7, thus opening the platinum contacts 8 and 10. The cam 5 is mounted on the rear end of the shaft of rotating magnet 1, its position being fixed in relation to the magnetic field.

The short contact point 10 is connected to the ground at 24, whereas the long contact point 8 maintains permanent contact with the primary winding 4. Therefore, when contacts 8 and 10 are opened, the primary current is suddenly interrupted. The condenser 11 is connected in parallel with contact points 8 and 10, thus preventing excessive sparking when the contacts are separated. The interruption of the primary current induces a high tension current in the secondary winding 12 which is composed of a large number of turns of fine wire. One end of the secondary winding is connected to the ground and the other end terminates in the carbon brush holder 13, which transmits the high tension current to the spark plugs through the distributor 15 as shown on the left (end view) of Fig. 137. A safety gap is provided between electrodes 32 and 33, which will protect the secondary winding against excessive voltage in the case of an open circuit between the distributor and the spark plugs. To stop the engine the ignition is cut off by bypassing the primary current around the breaker points 8 and 10 to the ground.

The advantages of the jump-spark ignition system are: no moving parts inside the cylinder, the spark gap being a permanent gap; easy adjustment of the time of ignition by varying the time of opening the breaker points. The disadvantages are: high insulation requirements; likelihood of spark plugs becoming fouled with oil or dirt; varying of intensity of spark with the compression pressure in the cylinder. In the usual high-tension magneto, the intensity of the spark is at a maximum only at the point of maximum flux, and is therefore reduced when the time of ignition is either advanced or retarded beyond this point. In the coil and battery system, on the other hand, the intensity remains the same irrespective of the time of ignition. This is a substantial advantage for the latter system of ignition.

92. Governing. The governing of internal-combustion engines does not differ materially from the governing of steam engines. The purpose of engine governors is to maintain approximately constant speed of rotation and to obtain maximum thermal efficiency at all loads.

Automobile and marine engines, also some high-speed Diesel engines, are usually hand-governed. There are three general types of internal-combustion engine governors: hit-and-miss governors, quantity governors, and quality governors.

With the *hit-and-miss* type of governor there is a working stroke for every cycle under conditions of maximum load. At lighter loads, when the speed increases, the governor mechanism acts to prevent admission of a charge of fuel occasionally and there is no explosion causing the engine to "miss." When a "miss" occurs not only does the inlet valve remain closed, but the exhaust valve is generally held open, so that during the strokes corresponding to the ordinary cycle the piston pumps exhaust gas into and out of the exhaust pipe with practically no loss of energy. The loss of a power stroke decreases the speed of the engine; the governor mechanism opens the inlet valve, an explosion or a "hit" occurs, and the engine receives a power impulse. With this method of governing, the engine either operates under conditions of maximum efficiency, or does not fire at all. Hit-and-miss governing gives better economy at light loads than the other methods, but does not give close speed regulation. Considerable intervals of time may intervene between power strokes of the engine and thus a heavy flywheel is required to avoid considerable variation in the speed. Hit-and-miss may be used for engines which do not require close speed regulation and with the smaller sizes, less than 40 brake horsepower.

Quantity governing may be accomplished by varying the amount of mixture entering the cylinder, whereas the proportions of the mixture and the number of cycles are constant. This result may be obtained in one of two ways: cutting off the charge before the piston reaches the end of the suction stroke; or throttling, except at full load, the charge during the suction stroke to reduce the amount of mixture entering the cylinder. The compression varies with the charge taken into the cylinder. Reducing the compression lowers the efficiency; hence quantity governing is not as efficient for a large variation of load as the hit-and-miss type. A quantity type of governor is shown in Fig. 34. The automobile engine is hand-governed by a quantity (throttle) control of the charge entering the cylinders, the proportion of the gasoline to air remaining the same for a given carburetor adjustment.

The *quality* governor varies the proportion of fuel to air, but the number of cycles and the quantity of charge per cycle remain the same. At full load a rich mixture is used, and at low loads a lean mixture. Mechanically, this method is quite simple but low thermal efficiencies result at light loads. Difficulty may be encountered in trying to obtain satisfactory ignition with very rich or very lean mixtures. Such mixtures burn very slowly and combustion continues in some cases throughout the entire expansion stroke.

Diesel engines use quality governing. The same amount of air is generally drawn into the cylinder during each suction stroke, with the governor controlling the amount of fuel injected per cycle. On some high-speed Diesel engines, the amount of air is varied by a vacuum control of the inlet air. The amount of fuel forced into the cylinder may be governed by varying the length of the stroke of the fuel pump or by by-passing the fuel so that more or less is injected into the cylinder during each power stroke. These methods of regulating the delivery from Diesel fuel pumps are explained in Art. 80 and are illustrated in Figs. 116 to 120, inclusive.

A number of combinations of the foregoing governors have been used. Thus quality governing may be used for the high and medium loads, and the hit-and-miss governor for the light loads. Close regulation may be obtained by employing quality governing at high loads and changing to quantity governing at low loads. These combinations are desirable, but they tend to complicate the valve gear and require accurate adjustments.

93. Lubrication. Lubrication is a matter of great importance in the operation of internal-combustion engines, because the rubbing surfaces are generally dry, the temperature, pressure, and speed are frequently high. Friction is a loss of power or energy occasioned by the rubbing of one body or surface over another. It is the resistance to the relative motion of surfaces in contact, and depends upon the nature and roughness of the surfaces. Friction is the conversion of useful energy into useless heat, accompanied by wear, and it can never be entirely eliminated in any machine. By making use of suitable lubricants properly applied, friction can be reduced to a minimum.

Lubrication may be defined as the introduction of a smooth fluid or semi-fluid substance, such as oil or grease, between two moving surfaces to prevent their coming into direct metallic contact. All bearing surfaces, however smooth to the naked eye or touch, are microscopically rough; and unless some medium is introduced which will cover and fill up these depressions, the surfaces will interlock and give rise to friction, heat, and rapid wear. The fundamental theory of lubrication requires a continuous film of oil between the rubbing surfaces of a properly lubricated bearing. When the moving parts are thus separated by a film of oil, the friction then takes place within the oil itself, and between its particles and the surfaces in contact with it. In this way the oil film prevents actual metallic contact, and eliminates probable abrasion and seizing of the surfaces.

All rubbing surfaces in an engine should be lubricated. The most important of these surfaces are the cylinder walls, main bearings, crankpins, piston pins, and camshaft bearings, but there are also numerous other parts to be lubricated. In an internal-combustion engine the oil film, in addition to reducing friction, must form a seal between the piston rings and cylinder wall, to prevent the leakage of the gases under pressure past the piston. For cylinder lubrication mineral oils of high flash point and nearly constant viscosity over a reasonable temperature range are most suitable. In all lubrication the selection of oils with the correct viscosity (light, medium, heavy, etc.) is very important. Other desired physical characteristics of a lubricating oil are: body sufficient to prevent metallic contact under maximum pressure and maximum temperature; fluidity at minimum temperatures; capacity to resist high temperatures without decomposition; high fire test; freedom from oxidation; freedom from corrosive action on metals; and a minimum of carbon residue when burned.

The coefficient of friction of a bearing with good lubrication, moderate pressures, and high speeds is practically independent of the materials composing the rubbing surfaces, but is proportional to the viscosity of the oil, to the rubbing speed, and to the area. The loads on the principal bearing surfaces are variable, going through a cycle of changes every two revolutions of a four-cycle engine, and varying from a maximum to a low minimum value. The direction of the force changes in these bearings, so that the portion of the bearing which at one instant is supporting maximum pressure may later be relieved of all pressure. This intermittent application of the load is favorable to good lubrication and permits the use of maximum pressures greatly in excess of what would be possible with continuous loading. The oil film which is squeezed out by the application of the maximum pressure is replaced during the reduction or reversal of the pressure.

The permissible pressure on a bearing depends on the viscosity and therefore on the temperature of the lubricating oil used. The temperature tends to rise and should be kept down by oil cooling. Temperatures of 160 F and higher are common. The friction at the bearings of an engine results in heat which must be taken away as rapidly as it is generated if the bearings are not to rise in temperature. Much of this heat is conducted through the metal to cooler parts of the engine but some heat is carried away by the oil. For this purpose a large flow of oil is desirable. The lubricating oil which gives minimum temperature rise of the bearings is the best to use, other factors being equal. An oil of lower viscosity would cause greater friction because it would squeeze out and allow a closer contact of metallic parts and increase metallic friction and wear. An oil of higher viscosity results in increased fluid friction of the oil. With a complete film of oil, the oil flows like a pack of playing cards sliding over each other, the outer layers adhering to the surfaces and not sliding with reference to them.

Lubricating systems divide naturally into two parts. One portion, by various devices, supplies the oil to the parts to be lubricated; the other tends to maintain the quality of the lubricant by the use of, singly or in combination, filters, coolers, settling tanks, reclaimers of various designs, separators, and centrifuges. Regardless of the combination of these devices employed for any engine, or engines, if the grouping is to serve successfully, it must be capable of removing the bulk of the heat and impurities from the lubricant at the same rate as that at which contamination is taking place. The rate of contamination, obviously a variable, is influenced by several definite factors. These are the quality of the lubricant, the type of combustion, the suitability of the engine fuel, and the mechanical condition of the engine.

In the *splash* system of lubrication, oil is poured into the crankcase of an internal-combustion engine, the oil level being maintained nearly constant by occasional supplies from an outside source. When the engine is in operation, the lower ends of the connecting rods strike the oil and splash it in all directions, thus filling oil cups which feed the main crankshaft bearings. Some of the oil is fed to the crankpin bearings through holes drilled in the lower ends of the connecting rods. Part of the oil is splashed onto the walls of the cylinder and lubricates the cylinder, piston, and piston rings. Another part is splashed into the hollow piston, where oil collects under the piston heads, passes through grooves in the upper end of the connecting rods, and lubricates the piston (wrist) pins. The camshaft parts catch the oil in pockets and feed it, by means of holes bored through them, to the bearings. The distribution gears and pushrods are lubricated by the oil which splashes over them.

Semi-splash lubrication is similar to the above, except that a circulating pump is used to force the oil from the crankcase to a channel or trough extending the full length of the upper part of the crankcase. From this trough the oil overflows into separate compartments under the connecting rods, thus maintaining a constant level in each, the surplus oil returning to the crankcase. The dippers on the connecting rods strike the oil as the engine turns and splash it in all directions, as explained under the splash system.

With high speeds the pressure on the bearings, particularly during the latter part of the working stroke, becomes great, because of the inertia of the reciprocating parts. In order to withstand this pressure, either larger bearing areas or *forced lubrication* of the bearings must be provided. In this system of lubrication the oil is forced from the crankcase of the engine by a submerged circulating pump, usually of the rotary-gear type, see Figs. 42 and 43, under a pressure of 20 to 40 lb per sq in., through oil pipes to the crankshaft bearings and the camshaft bearings. From the crankshaft bearings the oil is forced through holes bored in the crankshaft to the crankpin bearings. From the latter bearings the oil is further forced, through hollow-drilled connecting rods, to the piston pins. Oil escaping from the piston pins and crankpins lubricates the cylinders, pistons, and piston rings. Some engines employing forced-feed lubrication have provision for additional oil feed, under pressure to each cylinder, the feed being controlled by the speed of the engine. The bearings of most modern engines are lubricated by forced-feed systems.

94. Cooling of Engines. Water-cooling Systems. Of the heat generated by the combustion of the fuel in the cylinder of an internal-combustion engine, about 25 to 30 per cent is delivered as useful work at the crankshaft; the remainder of the heat must be conducted away either by the exhaust gases or by means of the cooling system. The exhaust gases carry away, some of which may be recovered by waste-heat boilers (Art. 96), about 30 per cent of the heat generated; and about 25 per cent is abstracted from the cylinder jackets, pistons, and lubricating oil. If some means were not provided for conducting this heat away from the cylinder, the high temperatures of the parts would burn the lubricating oil, and the piston and rings would expand to such an extent that they would seize the cylinder walls, and the engine would become inoperative.

The amount of heat that will flow per unit of surface from an engine cylinder depends on the design and material of the cylinder wall, cylinder head and piston, the thickness of the material, the condition (primarily cleanliness) of the surfaces on the inside and outside, the velocity and amount of the coolant, and the difference in temperature. The two systems of engine cooling generally used are *air cooling* and *water cooling.*

Small engines may be cooled successfully by increasing the area of the outside surface of the cylinder, as shown in Fig. 52. The cylinder has cooling fins arranged so that they are perpendicular to the axis of the cylinder. A current of air flowing over the cylinder wall will materially aid the cooling effect. The current of air may be obtained either from a fan driven by the engine, or by the movement of the engine itself, as in aircraft, automobile, and motorcycle engines. For further information on air-cooled engines, see Arts. 40 and 41.

The sectional views of water-cooled engines illustrated in this book show that the cylinder barrel and cylinder head have double walls, and in every case provision has been made for cooling by the continuous circulation of water through the space between the two walls. This space is called the water jacket. Single-acting engines are generally cooled only around the cylinder wall or liner, the cylinder head, and the exhaust valve. In some engines the exhaust manifold is also water-cooled; see Figs. 34, 65, and 74. When the diameter of the cylinder exceeds 20 in., the piston must also be cooled; some manufacturers employ piston cooling for diameters of 12 in. and more. Water may be used for piston cooling, see Figs. 69 and 77, as explained in Art. 54; cooling with oil, Figs. 70, 73, and 74, is preferable to cooling with water as explained in Art. 53. In double-acting engines, the piston and the piston rod must be liquid-cooled, because there is no opportunity for radiation as with single-acting engines. The heat absorbed by the lubricating oil and the heat carried away by the oil used for cooling some pistons are frequently transferred to the cooling water through the walls of an oil cooler; see Fig. 82.

In automobile and other portable types of internal-combustion engines, only a limited supply of cooling water can be carried; it therefore follows that means must also be provided for cooling this water after it leaves the cylinder. This is done in the so-called radiator. A small pump is used to force the cooling water through the water jacket and radiator. In one type of radiator, a large number of tubes are set vertically between a top and a bottom header, the tubes being provided with innumerable thin metal fins which increase the cooling surface. Another radiator design uses corrugated plates spaced to effect a honeycombed appearance. The suction created by the engine fan placed behind the radiator, also the forward movement of the vehicle in many cases, is sufficient to maintain the necessary flow of cold air across the radiator surfaces to effect the desired cooling.

The water-circulating system best adapted for any given stationary engine depends upon the quality and quantity of the available cooling water. Both of these factors vary greatly in different localities; consequently, a circulating system which is satisfactory for one plant may be entirely unsuitable and inadequate for another. Obviously, if an ample source of soft water is available at low cost, the circulating system can be made very simple by letting the discharge go to waste. Water that is entirely free from scale-forming matter is seldom available, but in some districts the quality of the water is reasonably good, so that the rate of scale formation is slow and the character of the scale is such that it can be easily removed. This system may be used under these conditions without undue hazard of overheating, provided the scale is not permitted to accumulate to an appreciable thickness before removal and provided the system maintains the rate of water flow so that the discharge temperature does not exceed 120 F. To produce this result requires about 20 to 30 gal per hp-hr at full engine load; for large engines, above 16 in. diameter, a somewhat smaller amount is sufficient. The quantity of cooling water required varies with the load for any engine; but it is desirable that the temperature difference between inlet and outlet of the cylinder jacket water does not exceed 20 F, and that the maximum outlet temperature does not exceed 130 F, in order to minimize scale formation.

Most internal-combustion engine installations require either open (direct) or closed (indirect) systems for cooling of the circulating water. An *open cooling system* is one in which the water is forced through the engine, then cooled by evaporation with a cooling tower or spray pond, and finally recirculated. The open system requires no heat exchangers, smaller evaporative equipment, and less pumping. A certain amount of the water (5 to 10 per cent) is always lost by evaporation and windage; this loss must be made up periodically. The make-up water (if not treated) carries scale-forming impurities into the system, making it necessary to drain frequently and refill the system. A *closed cooling system* continuously employs soft water and is completely sealed. The heat may be removed from the cooling water by transfer to air through extended radiation surface, as in a radiator system, or by transfer through heat exchanger tubes to raw water which is in turn cooled by evaporation and constantly recirculated.

Figure 138 is a schematic diagram of a closed water-cooling system that uses an atmospheric deck type of cooling tower and a shell-and-tube type of heat exchanger. This system may be designed to pump the soft water through the engine cylinders, directly into the heat exchanger, and back to the pump; the whole system would be under pressure. To provide for the surge effect of the cooling water and to permit escape of entrapped air, a vented surge tank is generally installed at the high point of the system; see Fig. 138. Where the available water is hard to any degree the savings in repairs, maintenance, and insurance costs more than balance the higher cost of the closed cooling system. The additional protection of a closed system is recognized by insurance

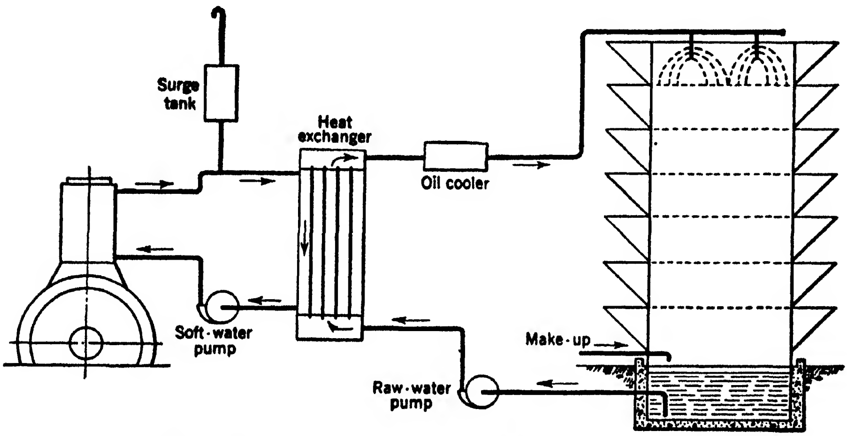
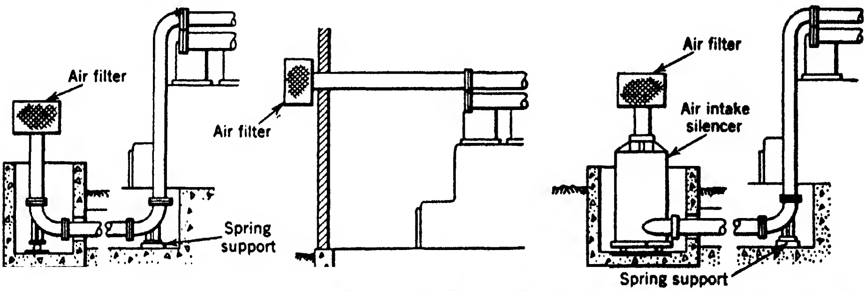


FIG. 138. Schematic diagram of a closed water-cooling system.

companies in that premiums are about 40 per cent less than for engines employing the open system of cooling. Since closed systems deposit little or no scale, the cylinder temperatures can be considerably higher, thus effecting not only higher engine efficiency but smaller cooling equipment. The size of the heat exchanger (or radiator) and the size of the evaporative cooling equipment vary inversely as the temperature difference between the jacket cooling water and the fluid receiving the heat.

95. Air Filters and Silencers. As population concentrates, the consideration of dust, noise, and odors with their various manifestations become of increasing importance as they affect machines, workers, and nearby population. Internal-combustion engines handle large volumes of air; even when this air contains a small percentage of dirt, a volume detrimental to engine operation is taken into the cylinders. Operation of engines with and without air filters has demonstrated the economic importance of supplying clean air to them. Noise is subtle in its consequences and induces a nervous fatigue that is cumulative, but its effect is not readily manifest to the subject. If noise were repeated in an identical amount as to frequency, volume, and tone, its effect over a period of time would induce a fatigue similar to that obtained by bending a piece of metal back and forth. Municipal regulations affecting crowded and congested population areas require the suppression or minimizing of noises, smoke, and odors; hence the development and applications of methods to reduce these offenses are receiving increased attention by the designer, builder, and user.

Scientific research has shown that in the average community there are about 4 grains of dust (45 per cent silica matter, 45 per cent smoke-



From "Standard Practices," page 94; Diesel Engine Manufacturers' Association.

FIG. 139. Three methods for installing air filters.

carbon, and 10 per cent miscellaneous) per 1000 cu ft of air. The silica dust in the air is of a very abrasive nature, and when combined with carbon and lubricating oil it forms a grinding paste. In an internal-combustion engine, the objection to dirty air is contamination of lubricating oil and wear of pistons, cylinders, and valves; also increased carbon deposits. It has been estimated that 90 per cent of the so-called carbon deposits in internal-combustion engines are dust saturated with oil and baked into a hard mass. A 200-hp engine takes in approximately 1250 cu ft of air per min, operating 8 hr per day and 30 days per month. It can readily be computed that about 10.3 lb of dirt enter the engine cylinders per month. To prevent this large amount of silica- and carbon-bearing dust from damaging the engine, it is essential to remove this dust from the air before it enters the air intake. If dust-free air entered the engine, it is likely that a saving of 75 per cent could be effected on lubricating oil, valve grinding, reboring of cylinders, replacement of valves and liners, etc. Air filters are used to supply the engine with the cleanest air possible under the circumstances; air filters may be of the *dry* or *wet* type. Three methods for installing air filters are shown in Fig. 139. One method illustrated shows a filter used in conjunction with an intake silencer; installations requiring the use of both are becoming general practice.

The characteristics desired in the ideal air filter are: efficiency in separating dust from air, little restriction, infrequent need for servicing, simplicity in design and construction, small size, freedom from troubles, ability to muffle intake noises, ability to prevent ignition of fuel from backfires in Otto-cycle engines, neatness, sturdiness, durability, and low cost.

Several air filters of the dry type are available. They are made in various forms according to their capacity and type of installation. These filters usually consist of an assembly of pockets formed of wire or wire mesh over which is placed a special felt or filter cloth, through which the air passes. The cloth may be cleaned by passing air through the filters in an opposite direction to the normal air flow or by using a vacuum cleaner on the outside. In some dry-type air filters of large size the filter cloth is supported in the form of bags, from which dust is removed mechanically. If the material is loosely woven or packed, plain dry filters may stop practically 100 per cent of the dust at first, but may later release that previously caught and allow it to pass on into the engine. If, on the other hand, the material is felted or closely woven, it may continue indefinitely to stop practically all the dust; but, as more and more dust is caught, the restriction may increase to such an extent that loss of engine power will result. Dry filters made of sponge rubber about $\frac{3}{4}$ in. thick have been used as air cleaners; the efficiency is very high and the restriction not excessive. They may be serviced by washing and rinsing in water.

The most popular type of wet filter operates on the adhesive-impingement principle. Dust-laden air is drawn through a series of viscous-coated metal baffles, called the filter media, which break the air current up into fine streams, causing the dust to be impinged against the many surfaces of the baffles. The filter media in this type, Fig. 140, consists of expanded metal or continuous strands of galvanized crimped wire which is packed into a compartment or cell in such a manner as to give a graduated variation in density, thereby causing uniform distribution of dust throughout the entire depth of the filter and permitting it to hold a large quantity of dust without materially increasing the resistance to air flow. These units may be combined in assemblies, as shown by Fig. 141, to provide the desired capacity. If the filters are washed with gasoline, dried, and re-oiled at proper intervals, they will maintain efficiencies well above 90 per cent and show continued low restriction to air flow.

Some of the small-size air filters are of the centrifugal (mechanical separation) type that remove the dirt by whirling the air around the

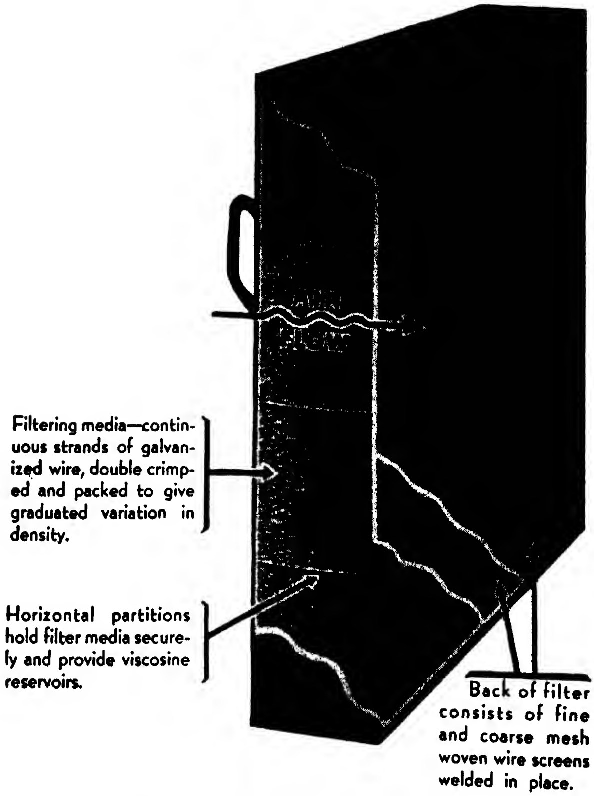


Fig. 140. Section through cell of an American air filter.

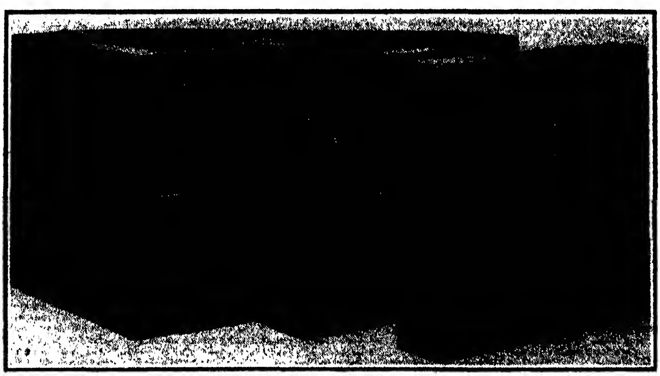


Fig. 141. Outside installation of air-filter assembly; note louvres for protection from weather.

inside wall of a circular intake chamber. In general, dry-centrifugal type filters do not afford adequate protection; at low air speeds this type loses practically all of its efficiency. Combinations of the adhesive-impingement and centrifugal principles are also used. In these the air is given a primary cleaning by centrifugal action, the dirt being collected in an oil pool. After being subjected to this action, the air then passes through an adhesive-impingement filter for the final cleaning.

The pressure of the gases in the cylinder of an internal-combustion engine is still high enough when the exhaust valve opens to cause them to escape with a loud explosive sound. Some provision is generally made for deadening or silencing the sound of the exhaust. Similarly to the exhaust we must also consider the intake which on many engines makes a very offensive noise. The devices for accomplishing the gradual lowering of the intake and exhaust pressures are known as silencers or mufflers. There are many types of silencers, among them the tank or concrete pit, baffle plate, expansion, ejector, and absorption or sound-wave filter.

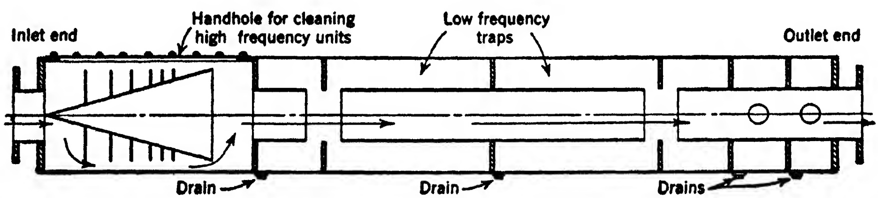


FIG. 142. Maxim silencer suitable for intake and exhaust.

The simplest silencing device is a tank or pit placed in the intake or exhaust connections of the engine. The enlargement of volume, Fig. 139, causes a corresponding fall in pressure, the entering air or exhaust gas velocity is reduced, and the noise muffled or eliminated. In the baffle type, Fig. 142, the sound wave is suppressed by passing the air or gases around various forms of baffling. The design shown in Fig. 142 when used as an intake silencer should give very nearly perfect silencing and when used on the exhaust it should eliminate about 90 per cent of the exhaust noise. In the expansion type, water supplied to a chamber cools the gases and thereby reduces the pressure. The absorption type of silencer employs a perforated tube surrounded by an outer casing, the space between being filled with an absorbent material. A successful silencer should eliminate practically all the intake or exhaust noise and also impose a minimum of resistance to the flow of intake air or exhaust gases.

For some installations the resistance of a waste-heat (exhaust) boiler, as explained in Art. 96, provides sufficient silencing on the exhaust side of the engine. With the windows closed and ventilation obtained through ventilators, the mechanical noises can be retained within the building and eliminated by the use of efficient sound-proofing materials on the walls and ceiling. The engine foundation should be insulated from the subsoil by placing cork slabs on the bottom and sides of the excavation before pouring the concrete. The engine (and generator) should be mounted on the foundation with a vibration-absorbing (resilient) material between the base of the engine and the top of the foundation. Short flexible metallic connectors are used on intake and exhaust piping as a means of preventing transmission of all stresses and vibration from and to the engine.

96. Waste-heat Recovery. The best designed internal-combustion engine can use only one-third of the heat in the fuel, the remaining two-thirds being about equally divided between the jacket water and the exhaust gases. The heat carried away in the water from the cylinder jackets is at a relatively low temperature, seldom exceeding 185 F. The exhaust gases are at a considerably higher temperature, especially in four-cycle engines, and therefore the recovery of heat from the exhaust presents greater possibilities. In some cases hot jacket water may be usefully employed as feedwater to the heat-recovery device used for the exhaust gases. Heat from the circulating water and exhaust gases may be employed for heating the engine fuel before it enters the cylinder, and for outside heating purposes, as in a hot-water (radiator) heating system or for process work requiring hot water.

The temperature of the gases, when first issuing through the exhaust valve, is about 1000 F. This temperature does not exist beyond the valve, since the pressure drop is accompanied by a temperature drop. The exhaust temperature will range from 400 to 700 F; the usual temperature of gases from Diesel engines of the four-cycle type is from 600 to 650 F and in engines of the two-cycle type the exhaust gases seldom exceed 500 F. By using ample radiating surface, a large part of the waste heat may be used for heating the plant. Assume a Diesel engine capacity of 400 horsepower and a fuel consumption of 0.45 lb per hp-hr; the hourly rate of fuel consumption would be $400 \times 0.45 = 180$ lb. If the oil contains 18,600 Btu per lb and the cooling water absorbs one-half of the cylinder-jacket and exhaust-gas heat losses, the cooling water would deliver to the heating system $180 \times 18{,}600 \times \frac{1}{2} \times 0.60 = 1{,}004{,}400$ Btu, or the equivalent of about 900 lb of steam per hr. An evaporation of 6 lb of water per lb of coal is typical of heating-plant practice; thus the heat recovery would be equivalent to 150 lb of coal per hr, or a saving of 3600 lb of coal per day.

The value of waste-heat recovery from the exhaust gases of internal-combustion engines depends, primarily, upon certain basic considerations. The maximum temperatures available are too low to have any useful radiant effect; therefore means must be devised to obtain the most effective transfer of heat by connection from the hot gases to the heating surface. If the recovered heat is to be used only for preheating the fuel or for heating air, a simple homemade heat exchanger, or one of the common commercial units, will function satisfactorily. If, however, steam or hot water at high temperature is to be generated a waste-heat boiler is more satisfactory. Either a heater or boiler, for this service,

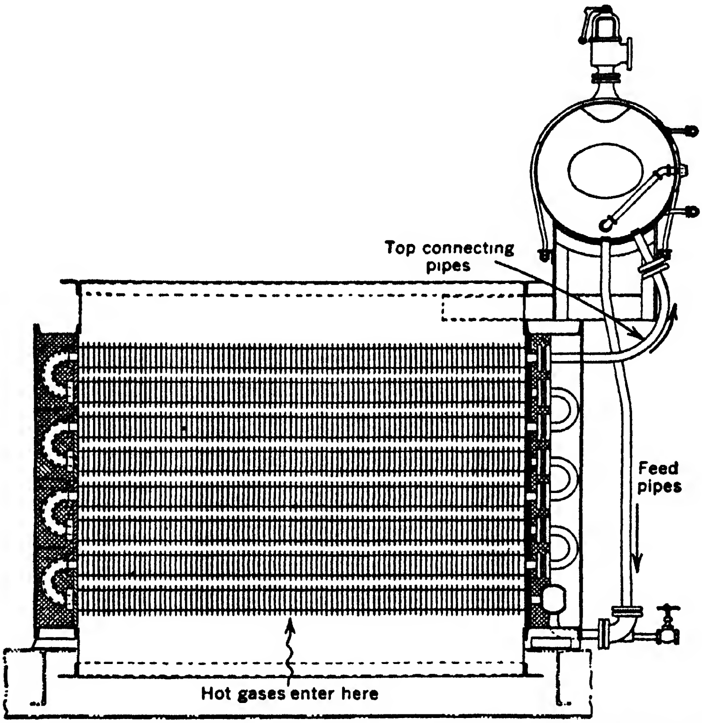


FIG. 143. Foster Wheeler waste-heat muffler boiler; design used for 960 to 5000 sq ft of heating surface.

must be able to withstand sudden changes in temperature, must be able to operate continuously, intermittently, or dry (when heat recovery is not needed) with a minimum of attention, and at the same time continue its other useful function of silencing. Smallness of size and weight are also important factors, especially for marine and railroad service. The waste-heat boiler should not materially increase the back pressure of the engine, and provision should be made for simple and effective removal of soot and scale from the boiler heating surfaces.

Figure 143 shows a cross section through a large waste-heat muffler boiler; this type is installed in many marine installations and in the municipal Diesel-electric plant at Vernon, California. Two 100-lb pressure waste-heat boilers per engine recover 62 per cent of the exhaust heat. Cast-iron rings over the 2-in. horizontal tubes increase the heat-absorbing area of the tubes six times; total heating surface, 1872 sq ft per boiler. A boiler of this type can be operated as a muffler only (without water) for exhaust temperatures below 750 F. The water volume is $36\frac{1}{2}$ cu ft per boiler, generating 2240 lb of steam per hr at 50 lb absolute and at $87\frac{1}{2}$ per cent of engine rating, with a gas temperature leaving the boiler of 360 F. The generated steam is used to heat the fuel and lubricating oil before centrifuging, and supplies evaporators for making distilled water to be used in the double-circuit enclosed type of engine-cooling system. The Foster Wheeler Corporation also manufactures smaller waste-heat boilers which employ vertical heating elements; boilers of this type contain 72 to 963 sq ft of heating surface.

The hot gases enter the waste-heat boiler at the bottom, as indicated in Fig. 143, and pass over the cast-iron rings on the tubes; the water is inside the tubes. This construction is compact and accessible; it permits placing the elements close together to provide the high gas velocities necessary for rapid heat transfer with the comparatively low temperature gases available. All joints are accessible from outside the setting; hence leaks may readily be detected. The heating elements are completely encased in an air-tight metal casing and insulated to prevent objectional heat radiation from the setting. The boiler drum and connecting pipes are also well insulated against unnecessary heat losses.

Success in constructing a steam boiler to operate with horizontal heating tubes has been obtained through the control of the area in the downcomer feed pipes, Fig. 143, from the bottom of the boiler drum. Restriction orifices are placed in that line adjacent to the elbow at the bottom of the line, and these have been so proportioned as to supply ample feedwater, while presenting resistance to a reverse flow in the downcomer pipes. Consequently, the flow setup because of the generation of steam is in the direction of least resistance and continuously upward through the heating surface to the steam drum through the top connecting pipes. During actual operation of these boilers, no steam or water hammer has developed and no circulation difficulties have been experienced.

The heat recovery from the exhaust gases of an internal-combustion engine can be estimated readily by the equation

$$h = bhp \times w_g \times (t_1 - t_2) \times c_p \qquad (129)$$

where h = heat recovered, btu per hour.
bhp = brake horsepower of engine.

w_g = weight of exhaust gases, pounds per brake horsepower per hour; about 15 for four-cycle Diesel engines, 22 for two-cycle Diesel engines, 10 for natural-gas engines.

t_1 = temperature of exhaust gases entering waste-heat boiler, degrees F.

t_2 = temperature of exhaust gases leaving waste-heat boiler, degrees F.

c_p = specific heat of exhaust gases; use 0.25.

More heat can be recovered in the form of hot water than in the form of steam, because it is obvious that when generating steam the temperature of the waste gases leaving the boiler must be considerably above boiler temperature. The difference is usually not less than 50 F, so that in a boiler generating steam at 100 lb per sq in., the final temperature of the exhaust gases would be about 400 F. If, on the other hand, hot water were produced at 200 F, the gases could easily be cooled to 300 F; this, in a two-cycle engine, would represent an increase of 100 per cent in the amount of heat recovered by the hot water as compared to that recoverable by steam.

The theoretical limit of recovery of a waste-heat boiler is the heat in the gases between the temperatures of the gases leaving the engine and the temperature of saturated steam at the desired operating pressure. It is impractical, however, to build a boiler which even closely approximates this condition; but it is economically feasible to design a boiler with an efficiency of 60 per cent, if it is permissible to build up a draft loss of 3 to 4 in. of water through the boiler. In small units, where the waste heat is derived from either a natural-gas or Diesel engine, the steam generated varies from 0.75 to 1.25 lb per bhp per hr. For large units, approximate figures on possible steam production can be obtained by taking 60 per cent of the total heat in the gases between the temperature leaving the engine and the saturated steam temperature at the desired pressure. Dividing this figure by the heat required to convert a pound of feedwater into steam will give the approximate steam production.

97. Control of Engine Vibrations. Engine Foundations. The need of controlling machine vibrations is generally recognized today. Modern building construction complicates the vibration problem; steel and concrete are excellent vibration-conductors and present-day designs of light weight facilitate the transmission of machine vibrations to all parts of the structure. Vibrations may also be carried through the soil to near-by structures. Machine designers attempt to balance moving parts to reduce vibration to a minimum, but even for the most modern machinery rigid bolting to foundations is inadequate for the purpose of preventing vibration transmission. The accepted remedy is to interpose a resilient medium between the engine and the foundation in such a manner that it will absorb the vibrations and prevent their spreading to adjoining structures.

Vibrations, when considered from the standpoint of control, may be classified into two major groups. One deals with vibrations resulting from direct contact of the vibrating body with a solid structure and progressing through the continuity of solid materials. These are called *solid-borne* or *impact* vibrations. As this discussion is concerned mainly with the control of engine vibrations by isolation, only impact vibrations will be considered.

An engine foundation system if it is not to transmit objectionable vibrations must comply with the following requirements: The engine must operate without excessive oscillation. The vibratory forces transmitted to the sub-structure must be reduced to such an extent that they will cause no disturbances. The natural frequency of the sub-structure must be sufficiently removed from the forced frequency of the operating engine. To control these disturbances, usually the best and most economical method is to block their progress at the source. This is done by isolating the machine by means of a material that is physically suitable for the purpose and that follows the mathematical laws governing vibration control. The engineer should select the most desirable material and then apply it in such a manner that it will also function most effectively as a vibration absorber.

There are many resilient media available—natural cork, compressed cork, rubber, gelatinous imitations of rubber, steel-spring devices, etc. The selection of the absorbing medium and its proper adaptation and installation should be based on scientific principles in order to attain effective isolation. The absorbing medium must have resiliency and a low modulus of elasticity; its resistance to sound conductivity must be markedly different from that of the vibrating engine. The isolating material must be resilient, by which is meant its ability to rebound quickly to its original state when the pressure causing a deformation is removed. The function of resiliency is to absorb vibrations by converting the energy of motion (kinetic energy) into internal strain (latent energy). The arrangement of an organic isolating material will also be greatly influenced by its ability to retain its shape under compression. Natural cork can be compressed to a fraction of its original volume and otherwise maintain its shape without the slightest deformation. It has a zero value of Poisson's ratio; see Table XXII. Rubber and rubber-like materials change their shape rather than volume and therefore if not permitted to expand laterally act as a rigid mass under compression; hence their use as isolators is restricted.

TABLE XXII

Physical Properties of Materials Considered in Control of Engine Vibrations

Material	Modulus of Elasticity, lb per sq in.	Poisson's Ratio	Sound Velocity, v, ft per sec	Density, d, lb per cu in.	Resistance to Sound Conductivity, $v \times d$
Steel	30,000,000	0.28 to 0.31	17,200	0.283	4860
Cast iron	10,000,000	0.28	12,400	0.260	3230
Concrete	....	....	16,500	0.072	1190
Wood (pine)	1,000,000	....	11,000	0.0145	159
Cork	700	0.00	1,600	0.0086	13.9
Rubber	....	0.50	200	0.0442	8.84

The isolating medium must have a low modulus of elasticity which indicates that its elongation or compression is comparatively great for a given loading. Table XXII gives values for different materials considered in the control of engine vibrations, with cork having the lowest modulus of the materials listed. Steel coil springs occupy the unusual position that, in spite of the high modulus of elasticity of the material from which they are made, they act admirably as isolators because of their shape, form, and heat treatment. A fundamental law states that to retard the progress of vibratory waves it is necessary to interpose between the source of the disturbance and the structure on which it is mounted a material which has a mechanical resistance to sound conductivity varying greatly from its two adjoining materials. The mechanical resistance to sound conductivity, Table XXII, is the product of the velocity at which sound progresses through a material and its density. The engine is made of cast iron and steel, whereas the foundation is composed of concrete and steel, all having a high resistance to sound conductivity, as shown in Table XXII. It is obvious therefore that for use as an interposed sound-absorbing medium, a material must be chosen from those listed near the bottom of the table, as wood, cork, or rubber.

The use of concrete for foundations of stationary engines is almost universal practice. A massive foundation[1] may be used to eliminate

[1] For foundation details refer to: "Kent's Mechanical Engineers' Handbook," Vol. II, Section 12, 11th edition, 1936, John Wiley and Sons, Inc.

"Vibration Problems in Engineering," Timoshenko, page 49, 1928, D. Van Nostrand Co.

"Foundations, Abutments, and Footings," Hool and Kinne, pages 293–298, 1923, McGraw-Hill Book Co.

vibration but it may place an excessive strain on the engine by restricting its freedom of operation. Cork mats are being used with success as an isolating material in engine foundations, as shown in Fig. 144. The mats are laid on the base and sides of the foundation pit and the joints sealed to prevent a seepage of concrete. The foundation area should be sufficient to produce a uniform loading not to exceed 4000 lb per sq ft on the cork of a thickness (1 to 3 in.) selected for its requirements. An improperly designed foundation may impair the functions of an otherwise well-planned isolation arrangement. Weight of foundation, cor-

FIG. 144. Installing of Korfund base isolator (cork mats) in a large engine foundation.

rectly distributed in length and width, must be carefully calculated to provide, in conjunction with the natural frequency of the vibration device, such amplitude of motion in the cushioned mass as is not excessive for satisfactory and safe operation. This calculation is essentially the work of the experienced designer backed by actual installations and tests to check his calculations. This type of installation must include the use of flexible connections, see Fig. 145, for all piping and conduits between the engine and other parts of the structure.

Advanced types of steel spring suspensions for absorbing engine vibrations mark the highest development of isolation engineering. The efficiency of the steel coil spring is due to its excellent elastic properties and its unusual adaptability to all isolation needs. In these respects it has a decided advantage over any other isolating medium. The scope of its application is almost limitless since the proportions (size and number

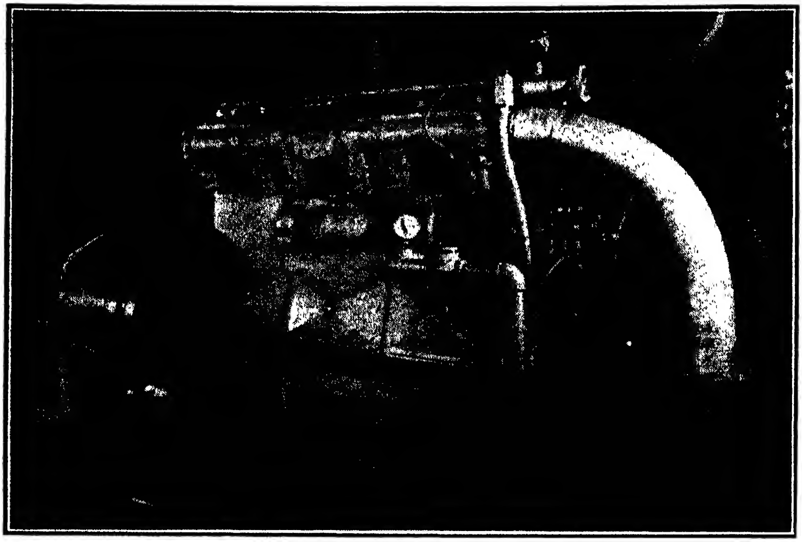


FIG. 145. High-speed Diesel engine installation with flexible piping connections between the engine and other parts of the structure. This engine-generator unit is supported on twelve Korfund Vibro-dampers, the latter resting on a cork-isolated foundation.

of springs) may obviously be varied over a wide range. One type of this device, illustrated in Fig. 146, consists principally of four units: a strong heavy base-housing and base plate drilled for anchoring the damper to

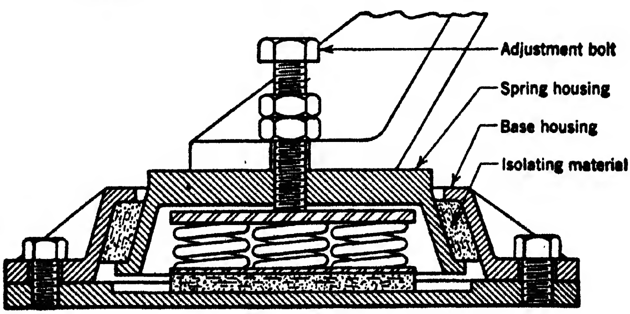


FIG. 146. Cross-section of Korfund Vibro-damper.

the foundation; a spring-housing resiliently separated from the base-housing and entirely suspended on springs; a number of cold-rolled oil-tempered supporting springs for each damper, depending on its size;

and an adjustment bolt which maintains the correct height of the damper and regulates the tension on the springs. Under the proper loading the spring housing drops several thousandths of an inch until it is entirely suspended on the supporting springs. Vibrations travel through the spring housing and the spring adjustment plate to the springs where they are absorbed. The springs in the Korfund Vibro-damper are combined so as to cover loading ranges from 50 to 15,000 pounds.

CHAPTER XIV

TESTING AND PERFORMANCE

Internal-combustion engines, compared to other power-generating and power-using machines, include a much greater variety in form, size, speed, weight, fuels, and operating conditions, with more or less special features for particular applications. Sizes of internal-combustion engines vary from 1 to 25,000 hp; speeds range from less than 100 to 4000 rpm. The number of cylinders employed may be from 1 to 12, or more; horizontal or vertical cylinder arrangement; some single-acting and others double-acting; some two-cycle and others four-cycle, with every variety of attached and separate scavenging pump for two-cycle engines. Fuels range from the lightest distillates to the heavy residue fuel oils, organic and mineral, in the liquid-fuel class; and in the gaseous form include every combustible gas commercially available.

Purpose and variety of load or other service conditions for each engine cover wide ranges. Stationary engines operate at all loads normally at constant speed under governor control, for any period of time and any degree of intermittent service. Marine engines operate normally at full load and at the constant speed fixed by the propeller, but must be maneuvered ahead and astern at any speed under hand control. They may operate for any period of time from short intermittent runs to the continuous full load of a month or more required of seagoing motorships. Aircraft engines must meet about the same service conditions as marine engines, except that a shorter length of single run is required; but they must operate in the greatest known ranges of atmospheric pressure and of temperature. Other engines of the high-speed group—those for automobiles, motor trucks, tractors, railroad locomotives, and general industrial purposes—are essentially variable in speed as normally used, and equally variable in power output.

98. Tests and Test Procedure. The purposes of a test, for determining the performance characteristics of an internal-combustion engine, may vary somewhat with the type of service for which the engine is to be used; but in most cases the major results desired are power output, thermal efficiency, mechanical efficiency, and cost of operation. These results may also be expressed in other terms, such as pounds of fuel per horsepower per hour, Btu supplied per horsepower per hour, indicated horsepower, brake horsepower, friction horsepower, etc. In any case some of the essential test measurements are: higher heating value of fuel, rate of fuel consumption, engine speed, power developed (ihp), and power output (bhp). The usual type of testing consists of making a number of tests at the same speed but with different loads; such tests are repeated at other speeds over the entire speed range when a variable-speed engine is being tested.

In testing an engine, efficiency and economy are the usual results desired; but the purpose of the test may be entirely different, such as a comparative study of engine fuels, of air-fuel ratios, or the effect of some design characteristic of the engine.

Important precautions that should be observed when testing an internal-combustion engine, as stated in the "Test Code for Internal-Combustion Engines" of the American Society of Mechanical Engineers (A.S.M.E.), are: Before any test readings are taken, it is necessary that the engine be ~~operated at a steady~~ load long enough for all conditions to become constant; this applies also to successive runs under the same or other loads. Spring-type indicators must not be used for determining the mean effective pressure in the cylinder (and the indicated horsepower) when the engine speed exceeds 400 rpm. Indicators must not be used on engines governed by the hit-and-miss principle. Brakes or absorption dynamometers, when used, must be suitable for the engine to be tested, in capacity, in speed-torque relations, and in ability to maintain an adjustment. The higher heating value of the fuel shall be used, and determined by a reputable laboratory. All instruments, meters, etc., must be carefully calibrated, preferably before and after the test.

For testing high-speed engines (above 400 rpm), the direct-connected cradled electric dynamometer is undoubtedly the best. With a source of direct current available, the dynamometer can be used to start the engine and can also be used to determine the friction horsepower by motoring. When the engine under test is direct-connected to an electric generator, it is usually necessary to make a test of the unit as a whole. If it is not practicable to make an electrical test or analysis for the calibration of the generator, the engine tests are reported on the basis of the electric horsepower or kilowatt output. Large engines built for marine or other purposes for which no generator is required are generally loaded by means of a hydraulic (also called fluid-friction) dynamometer. These dynamometers have a wide range of power absorption for medium to low speeds and are built to absorb upward of 5000 horsepower.

For the rather slow-speed type of stationary engines, spring-type indicators similar to the common type of steam engine indicator may be used. They are, however, usually made more rugged to withstand the higher stresses imposed by the higher and more rapidly increasing pressures found in internal-combustion engines. For small engines of less than 100 hp and operating at speeds below 500 rpm, prony or rope brakes may be used. These brakes may be cheaply and easily constructed, and therefore are favored for temporary installations.

Standard methods for conducting tests on various types of power machinery, also directions for computing and presenting the results, have been prepared and published by the several engineering societies and manufacturers' associations. It is obviously impossible in any single test code to prescribe items in detail that would be equally applicable to all conditions; thus some exclusions are generally necessary. The A.S.M.E. Test Code for Internal-Combustion Engines, Appendix Tables XXVII and XXVIII, is applicable for practically all types of engines. Items may be omitted if not required for the purpose of the test, or new items may be added for other data and results desired.

The Society of Automotive Engineers has prepared Standard Test Forms for Gasoline Engines and for Diesel Engines, Appendix Tables XXIX, XXX, XXXI and XXXII. Both sets of forms consist of four sheets, as follows: rules and directions, specification sheet, log sheet, and curve sheets. The S.A.E. forms provide spaces for very complete information on construction details and performance of an engine, its component parts, cooling, carburetion, ignition, lubrication, accessories, etc.

99. Horsepower. If indicated-horsepower determinations have been specified in the agreement before test and when the indicator can be used, and when in addition after using it the diagrams are found to be acceptable as to accuracy for determining the mean indicated pressure, then the indicated horsepower (item 53, Table XXVII) shall be calculated by the following equation

$$ihp = \frac{p_m LAN'}{33{,}000} \tag{130}$$

where p_m = mean effective pressure, pounds per square inch.
L = length of stroke, feet.
A = cross-sectional area of cylinder, square inches.
N' = number of *explosions* per minute.

When an explosion occurs during every cycle: for a single-acting four-stroke engine, the number of explosions per minute will equal one-half the revolutions per minute of the crankshaft; for a single-acting two-stroke engine the number of explosions per minute will equal the revolutions per minute of the crankshaft; and for a double-acting two-stroke cycle engine the number of explosions per minute will be twice the revolutions per minute. The total indicated horsepower of a multi-cylinder engine is the sum of the horsepowers developed in the individual cylinders.

The engine manufacturer invariably works on a brake horsepower basis, on account of the difficulties encountered in trying to measure accurately the indicated horsepower. The accuracy of the indicator diagram may be effected by the high speed, high temperature, high pressure, and small clearance volume of the engine. Special indicators, such as optical[1] and diaphragm[2] types, are generally used for engine speeds exceeding 400 rpm, for research work, and for greater accuracy at lower speeds. With internal-combustion engines an indicator is generally more valuable for checking the working cycle in its various details than for determining the horsepower. The usefulness of indicator diagrams for setting the ignition or fuel injection, valve timing, determining the volumetric efficiency, etc., has frequently been referred to throughout this book.

The brake horsepower is usually determined by some type of power-absorption device, such as a prony brake, hydraulic dynamometer, fan dynamometer, electric generator, or electric dynamometer; of these the cradled electric dynamometer is most satisfactory for test purposes as it serves either as a motor to run the engine or as a generator for absorbing power. The shaft of the engine under test is coupled to the shaft of the dynamometer armature. By electromagnetic interaction, between the armature and the field magnets, the torque on the shaft is transmitted to an external floating field or casing which includes the stationary poles. The floating field is mounted on anti-friction bearings, and the torque exerted is weighed directly by scales. The adjustment of either the torque or speed is made by a dual control of the strength of the current in both the armature and field windings. By means of this equipment, an engine may be tested at any speed or torque.

Regardless of which type of power-absorbing device is used, the brake horsepower (item 54, Table XXVII, and item 14, Table XXVIII) is determined as follows:

$$bhp = \frac{2\pi LWN}{33{,}000} \tag{131}$$

where L = length of brake arm, feet.
W = net weight on the brake, pounds.
N = number of *revolutions* per minute.

The nominal brake horsepower rating, of a four-cycle engine used for

[1] "Indicators," A Symposium of Papers on Indicators, L. Pendred, Proc. Inst. Mech. Engrs., January, 1923.

[2] Report 109, National Advisory Committee for Aeronautics, Washington, D. C.

automobile taxation by some states, is given by the S.A.E. equation

$$hp = \frac{d^2n}{2.5} \tag{132}$$

The use of this equation and its limitations are discussed in the latter part of Art. 39.

The brake mean effective pressure (item 55, Table XXVII, and item 24, Table XXVIII) is the mep which, if acting on the piston, would develop power equivalent to the brake horsepower; it is calculated from the equation

$$bmep = \frac{bhp \times 33{,}000}{LAN'} \tag{133}$$

The brake mep affords an accurate means for comparison of engine performance, and is used as such in preference to the indicated mep. This is due to the inherent inability of the spring-type indicator to give an accurate diagram. Because of the variable factors that influence combustion, such as the amount of burned gases left in the cylinder, the air-fuel ratio, etc., the successive cycles of an internal-combustion engine may vary considerably. This variation is eliminated, however, in the mep based on the brake horsepower.

The torque (item 25, Table XXVIII), which is frequently used as a measure of the output of an engine, is the turning moment or pounds of force exerted at a radius of 1 ft. If T_q represents the torque in pound-feet, then

$$T_q = \frac{bhp \times 33{,}000}{2\pi N} \tag{134}$$

The difference between the indicated and brake horsepower is known as the mechanical loss or friction horsepower and includes the negative loop (pumping loss) of the indicator diagram. When an engine is tested with a dynamometer, the friction horsepower is determined by motoring the dynamometer at the same speed used in determining the brake horsepower immediately after the brake horsepower test. During the friction test the fuel supply and the cooling water supply should be shut off; all other settings and accessories should remain the same as during the brake horsepower test. The friction run should be just long enough to obtain accurate results; the friction horsepower is calculated by substituting measured quantities in equation 131. This method of determining the friction horsepower is not very accurate, but it is practical and convenient. The friction determined by motoring is not the same as the friction existing while the engine is delivering power; the load on the bearings and side thrust of the piston are reduced. Also the piston will carry sufficient lubricating oil to cover entirely the cylinder walls in a few revolutions, whereas most of the lubricating oil is burned from the cylinder walls exposed to the hot expanding gases when the engine is developing power. At a given speed, the indicated horsepower is approximately equal to the sum of the brake horsepower and the friction horsepower,

$$ihp = bhp + fhp \tag{135}$$

100. Decrease in Power of Engines at Altitude. The power developed by an internal-combustion engine is the product of the heat of combustion of the fuel and the thermal efficiency of the engine. The thermal efficiency is not affected by the density of the air and, in general, depends only on the compression ratio. The heat of combustion is determined by the weight of the fuel which can be burned; this depends on the weight of air admitted and consequently on the density of the air. Hence, if other conditions were constant the power developed by an engine would vary directly as the density of the air supplied to the engine.

Internal-combustion engine tests made under various conditions of barometric pressure and temperature should be corrected to standard conditions of 29.92 in. Hg and 60 F, thus

$$bhp_s = bhp_a \times \frac{p_s}{p_a} \times \left(\frac{T_a}{T_s}\right)^{1/2} \tag{136}$$

where bhp_s = brake horsepower, corrected to standard conditions.
bhp_a = brake horsepower at altitude (or test conditions).
p_a = barometric pressure at altitude, in. Hg.
p_s = standard barometric pressure, 29.92 in. Hg.
T_a = absolute temperature at altitude, degrees F.
T_s = standard absolute temperature, 520 F abs.

The horsepower determinations may be corrected for exhaust back pressure variations, assuming 1 in. Hg back pressure as standard. An increase in back pressure of 1 lb per sq in. decreases the power output of an engine about $2\frac{1}{2}$ per cent.

The pressure of the suction stroke is governed by the pressure of the surrounding atmosphere; the temperature, however, is as much affected by the cylinder temperature as by the atmospheric temperature. Therefore, the power does not fall quite as rapidly as the pressure but more rapidly than the density. The rate at which pressure and temperature fall with altitude is shown in Table XXIII. Experiments have shown

TABLE XXIII

EFFECT OF ALTITUDE UPON ATMOSPHERIC CONDITIONS*

Altitude in Feet above Sea Level	Atmos. Pressure, Inches of Mercury	Mean Temperature, degrees F	Atmos. Density, lb per cu ft
0	29.92	59.00	0.0765
500	29.38	57.22	0.0754
1,000	28.86	55.43	0.0743
5,000	24.89	41.17	0.0659
10,000	20.58	23.34	0.0565
20,000	13.74	−12.32	0.0408
30,000	8.88	−48.00	0.0286
35,000	6.93	−67.00	0.0234
40,000	5.54	−67.00	0.0187
50,000	3.44	−67.00	0.0116
60,000	2.13	−67.00	0.0072

* From Report 218, National Advisory Committee for Aeronautics.

that the brake horsepower of an engine decreases about 3 per cent for every 1000 ft of altitude; see Report 295, National Advisory Committee for Aeronautics. Tests have shown that the bhp_a at altitude can be found by the equation

$$bhp_a = ihp_s \times \frac{p_a}{p_s} \times \left(\frac{T_s}{T_a}\right)^{1/2} - fhp_s \tag{137}$$

where ihp_s is at sea level, fhp_s is the friction horsepower at sea level and may be considered constant for all altitudes. p and T are the absolute pressure and temperature of the air at sea level or altitude, as indicated by the subscripts. If e_m is the mechanical efficiency of the engine at sea level (standard conditions)

$$ihp_s = \frac{bhp_s}{e_m} \tag{138}$$

but

$$ihp_s = bhp_s + fhp_s$$

therefore

$$\frac{bhp_s}{e_m} = bhp_s + fhp_s \tag{139}$$

and

$$fhp_s = bhp_s\left(\frac{1 - e_m}{e_m}\right) \tag{140}$$

from (137),

$$bhp_a = \frac{bhp_s}{e_m} \times \frac{p_a}{p_s} \times \left(\frac{T_s}{T_a}\right)^{1/2} - bhp_s\left(\frac{1 - e_m}{e_m}\right) \tag{141}$$

hence

$$bhp_a = \frac{bhp_s}{e_m}\left[\frac{p_a}{p_s} \times \left(\frac{T_s}{T_a}\right)^{1/2} + e_m - 1\right] \tag{142}$$

The decrease in power with altitude in any given engine resolves itself into finding a relation between the temperature and pressure inside the engine at the end of the suction stroke, and the temperature and pressure of the surrounding air. Experiments have shown that the pressure in the engine when compression begins is directly related to the outside pressure. When the atmospheric pressure is halved, the pressure at the beginning of compression is also halved; thus pressure variations have their full effect. This, however, is not generally the case for temperature variations. The temperature of the charge at the beginning of compression for ground level conditions averages about 215 F. The air on its way to the engine cylinder will generally lose heat in the carburetor; hence a considerable amount of heating must take place in the cylinder. These heat exchanges will tend to conceal, more or less, the effect of outside temperature variations, provided these are within moderate limits. The exact relationship for various engines operated under different conditions of carburetion will vary. In general only about one-third to one-fourth of the outside temperature variation is effective in producing changes of temperature in the cylinder.

For atmospheric air, a decrease of pressure during ascent is associated with a decrease in temperature; see Table XXIII. If this decrease in temperature is not fully effective inside the engine cylinder, the power will decrease slightly faster than the air density; most aircraft engines exhibit this characteristic. Measurements under flying conditions are difficult, and the results may be affected by changes in carburetion and mechanical efficiency; for this reason altitude test rooms are used to test engines at ground level under conditions as to pressure and temperature corresponding to various altitudes. A large number of engine tests in an altitude chamber and the results of flight tests have shown that the performance of aircraft engines at different altitudes is expressed closely by equation 142.

The decrease in power, with higher altitudes, limits the performance of an airplane. Attempts have been made to design an engine which up to a certain altitude will develop constant power. Two general methods for increasing engine power at high altitude are possible. First, build an engine so large that it will give the desired power when operating with wide-open throttle at the high altitude at which the airplane is intended to fly, and can be operated at partial throttle at all lower altitudes. Second, build an engine which gives the desired power at ground level and add some device for supplying the cylinder with air at a pressure greater than the barometric pressure when desired; this method is known as supercharging. At low altitudes the oversize engine is kept throttled down; as the height increases the throttle is opened and constancy of power is thus obtained until the throttle is wide open. For altitudes up to 20,000 ft, the over-dimensioned engine is generally considered the simpler design. In the supercharged engine, air is compressed by a blower or other device and is delivered to the carburetor at a pressure in excess of the surrounding atmospheric pressure. The blower may be either geared to the airplane engine, or driven by the exhaust through a small turbine, or it may be driven by an entirely separate power unit.

It might appear as if the over-dimensioned engine would necessarily be heavy. This is not strictly true; although of larger bore and stroke than the normal engine, it is called upon to develop only the same power. Thus it can be built relatively light.

Suppose it were desired to fly an airplane at 20,000 ft developing 500 hp. This can be accomplished either by installing an engine which would develop 1000 hp with wide-open throttle at ground level, or by installing a 500-hp engine provided with a supercharging device which would be able to maintain that horsepower at all altitudes, up to 20,000 ft. An estimate made of the increase in weight which would result from doubling the power of an engine by doubling the piston area per cylinder, while keeping the stroke constant, indicates this increase of weight would be about 40 per cent. If the power were doubled by doubling the number of cylinders the weight would be nearly doubled. It should be noted that if the engine does not develop more than 500 hp at any elevation, the radiator, water pump, and general cooling system will not be larger than for a 500-hp engine. If the supercharged engine were used the engine weight would be increased by the addition of the supercharging apparatus and the engine would become more complicated. In the oversized engine, the greater weight is offset not only by greater simplicity (as compared with a supercharged engine) but also by greater economy. The oversized engine should be provided with an automatically controlled throttle valve which responds to changes in atmospheric pressure. This control should be so adjusted as to give constant horsepower at all altitudes up to that at which the throttle is wide open; the power could not be maintained beyond that point. With an engine so operated it would be possible to use a higher ratio of compression than with an engine which had a wide-open throttle at the ground level.

Performance curves for the Wasp nine-cylinder air-cooled radial engine are given in Figs. 147 and 148. The full throttle horsepower curve in Fig. 147 shows the brake horsepower developed at various speeds with full-open throttle; however the maximum horsepower available for the take-off is considerably less. To develop the power indicated by the full

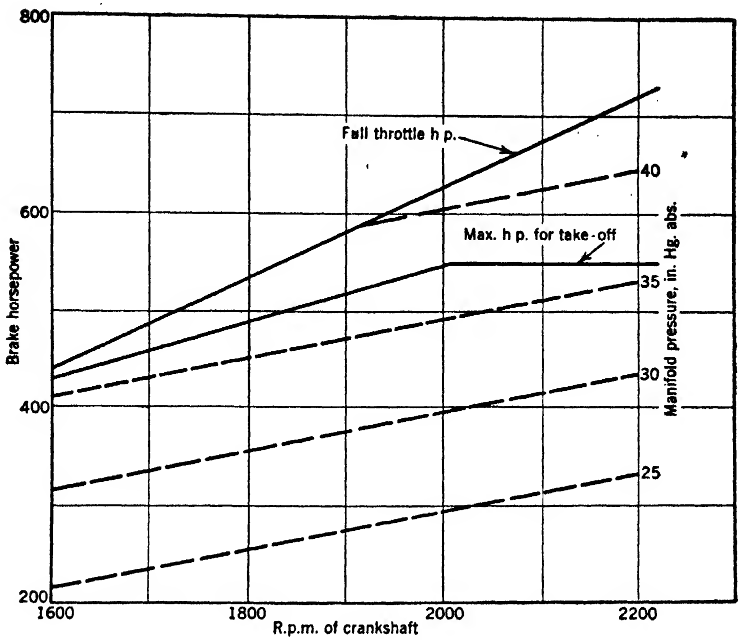


FIG. 147. Horsepower and intake manifold pressure for various speeds of Wasp radial airplane engine at *sea level.*

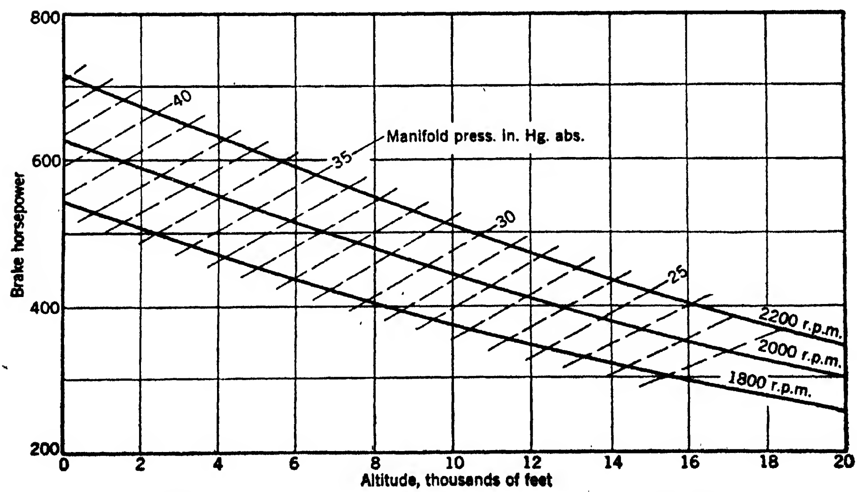


FIG. 148. Horsepower and intake manifold pressure at several speeds of Wasp radial airplane engine for *various altitudes.*

line curves of Fig. 147, the supercharger produced a corresponding intake manifold pressure according to the dotted lines. As an airplane ascends, the atmospheric pressure and temperature decrease, and consequently there will be a decrease in the power developed by the engine. The supercharger compensates, in part, for the decrease in pressure; under test conditions of full-open throttle and constant engine speed, Fig. 148, it is to be observed that the manifold pressure and the engine brake horsepower decrease with increase in altitude. The amount and rate of decrease depend upon the engine and supercharger design.

101. Horsepower Correction for Humidity. The presence of any given volume of water vapor in the cylinder, by lessening the oxygen present, reduces the quantity of fuel that can be burned efficiently per cycle and correspondingly decreases the power output. Tests[3] have shown that humidity should be taken into consideration in engine calculations because of its relatively important effect on maximum power, spark-advance requirements, carburetor metering characteristics, and radiator performance, and its lesser but measurable effect on detonation. Under extreme conditions, failure to allow for the effect of atmospheric pressure and humidity may introduce errors of nearly 10 per cent in each case.

For small changes in pressure, both atmospheric air and water vapor can be assumed to behave in accordance with the laws of perfect gases, including Dalton's law of partial pressures. Observed barometric readings thus represent the sum of the partial pressures of dry air and water vapor. The water-vapor pressure may be determined by substituting the dry- and wet-bulb thermometer (psychrometer) readings in the following equation; see also Fig. 149,

$$p_{wv} = p_1 - 0.000367p_a(t - t_1)\left[1 + \frac{t_1 - 32}{1571}\right] \tag{143}$$

where p_{wv} = existing water-vapor pressure, in. Hg.

p_1 = saturation pressure at temperature t_1, in. Hg as obtained from steam tables.

p_a = barometric (total) pressure, in. Hg.

t = dry-bulb temperature, degrees F.

t_1 = wet-bulb temperature, degrees F.

The characteristic gas equation, $PV = MRT$, shows that the partial pressure of a given constituent defines its weight in a given volume of mixture. Using gas constants of 53.35 for air and 85.8 for water vapor,

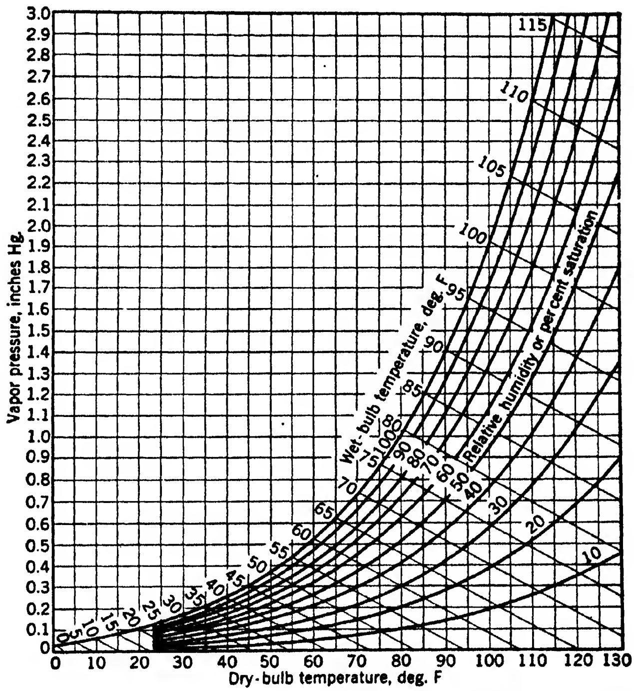
[3] "Atmospheric Humidity and Engine Performance," Arthur W. Gardiner, page 267, S.A.E. Trans., Vol. 24 (1929).

"Horsepower Correction for Atmospheric Humidity," Donald B. Brooks, page 273, S.A.E. Trans., Vol. 24 (1929).

$$M_d = 1.326(p_a - p_{wv})\left(\frac{V}{T}\right) \tag{144}$$

$$M_{wv} = 1.326(0.6217p_{wv})\left(\frac{V}{T}\right) \tag{145}$$

$$M_a = 1.326(p_a - 0.378p_{wv})\left(\frac{V}{T}\right) \tag{146}$$



Courtesy of *S.A.E. Transactions.*

FIG. 149. Humidity chart giving water-vapor pressure directly from psychrometer readings.

where M_d = weight of dry air in a given volume V of atmospheric air, pounds.

M_{wv} = weight of water vapor in a given volume V of atmospheric air, pounds.

M_a = weight of atmospheric air (mixture) in a given volume V, pounds.

V = volume of atmospheric air (mixture), cubic feet.

T = dry-bulb temperature of atmospheric air (mixture), degrees F abs.

As an engine under given conditions can be assumed to induct a constant volume of air-vapor charge, and as, for small changes in operating conditions, indicated torque can be assumed to vary proportionally with the weight of dry air, it follows that torque varies proportionally with the dry-air pressure $(p_a - p_{wv})$. This fact forms the basis for obtaining rational power correction factors; to correct the horsepower to a dry-air condition, the correction factor for humidity is $[p_a \div (p_a - p_{wv})]$. Instead of correcting for humidity independently, a combined correction can be employed, in which the pressure of dry air $(p_a - p_{wv})$ is substituted for the barometric pressure p_a, in the power correction factor for pressure. Thus, the correction factor for humidity and pressure is $[29.92 \div (p_a - p_{wv})]$, and equation 136 becomes

$$bhp'_s = bhp_a \times \frac{29.92}{(p_a - p_{wv})} \times \left(\frac{T_a}{T_s}\right)^{1/2} \qquad (147)$$

where bhp'_s = brake horsepower, corrected to a dry-air basis and standard atmospheric pressure of 29.92 in. Hg.

Example. An internal-combustion engine delivers 95 bhp when the barometer reads 28 in. Hg. The engine is supplied with an air-fuel ratio of 14 to 1, by weight. The air has a relative humidity of 60 per cent at 80 F. (*a*) What is the dry-air pressure? (*b*) What is the air-fuel ratio on a dry-air basis? (*c*) How much power would the engine deliver at standard conditions of dry air?

Solution. (*a*) Refer to Fig. 149, from dry-bulb temperature of 80 F go vertically to 60 per cent humidity curve; at intersection go horizontally to right margin and read 0.62 in. Hg vapor pressure. This value can be calculated with equation 143.

$$p_d = 28 - 0.62 = 27.38 \text{ in. Hg, dry-air pressure}$$

(*b*) From (144) $M_d = \dfrac{1.326\,(28 - 0.62) \times 1}{540} = 0.0672$ lb dry air per cu ft of mixture

$$M_{da} = \frac{PV}{RT} = \frac{28 \times 0.491 \times 144 \times 1}{53.35 \times 540} = 0.0687 \text{ lb dry air per cu ft of dry air}$$

$$\text{Air-fuel ratio} = \frac{0.0672}{0.0687} \times 14 = 13.7 \text{ to 1 on a dry basis}$$

(*c*) $\text{bhp}'_s = \dfrac{95 \times 29.92}{(28 - 0.62)} \times \left(\dfrac{540}{520}\right)^{1/2} = 106$ bhp at standard conditions

102. Efficiencies. The *volumetric efficiency* is the ratio of the charge which enters or is forced into the cylinder during the suction stroke, at standard conditions, to the piston displacement per stroke. Some of the factors which decrease the volumetric efficiency are: long and circuitous inlet passages, silencer and air-filter restrictions, insufficient inlet (and exhaust) valve area, insufficient valve lift, incorrect valve timing, premature preheating of the charge, inertia effect of high charge velocities at high engine speeds, and back pressure of exhaust gases. The volume of the air entering the cylinder may be measured with a meter of the orifice or venturi type. The volumetric efficiency of an engine increases with the speed, reaches a maximum, and then decreases. Constant volumetric efficiency may be maintained by supercharging of the engine cylinders, as explained in Arts. 19 and 100.

Example. A $12 \times 18\frac{1}{4}$-in. single-acting four-cycle cylinder of a Diesel engine was supplied with 458 lb of air per hour when operating at 200 rpm; atmospheric pressure was 29.44 in. Hg and the temperature was 64 F. Find the volumetric efficiency of the cylinder.

Solution. Air supplied per suction stroke $= \dfrac{458 \times 2}{60 \times 200} = 0.0763$ lb

At standard conditions, $$V_s = \frac{MRT}{P} = \frac{0.0763 \times 53.35 \times 520}{29.92 \times 0.491 \times 144} = 1.002 \text{ cu ft}$$

Piston displacement per stroke, $$V_d = \frac{\pi}{4} \times \frac{(12)^2}{144} \times \frac{18.25}{12} = 1.194 \text{ cu ft}$$

Volumetric efficiency, $$e_v = \frac{1.002}{1.194} \times 100 = 84 \text{ per cent}$$

The *mechanical efficiency* of an engine is the ratio of the brake horsepower output to the indicated horsepower developed.

$$e_m = \frac{bhp}{ihp} \tag{148}$$

The mechanical efficiency increases with the increase of load, as the friction losses of an engine are independent (nearly) of the load for a given speed; obviously, the higher the mechanical efficiency the lower must be the losses. The losses include the following items: suction and exhaust pumping losses in four-stroke cycle engines, scavenging in two-stroke cycle engines; friction of pistons, bearings, gears, and valve mechanisms; ventilating losses of fan and flywheel; power absorbed by accessories and auxiliaries. The type and size of engine, speed, cooling conditions, lubrication methods, and the accuracy of manufacturing and assembly of parts are factors which will affect the mechanical efficiency of an engine.

The *indicated thermal efficiency* (item 64, Table XXVII) shows what fraction, or percentage, of the heat supplied was transformed into indicated (developed) horsepower in the engine cylinders.

$$e_i = \frac{2545 \times ihp}{W_f \times H_f} \tag{149}$$

where ihp = indicated horsepower of engine.
W_f = fuel supplied per hour; in cubic feet for gaseous fuels, in pounds for liquid fuels.
H_f = higher heating value of fuel supplied; Btu per cubic foot or pound, as required.

The *brake thermal efficiency* (item 65, Table XXVII and item 23, Table XXVIII) shows what fraction, or percentage, of the heat supplied was actually delivered by the engine.

$$e_b = \frac{2545 \times bhp}{W_f \times H_f} \tag{150}$$

where bhp = brake horsepower of engine. Other symbols as for equation 149.

103. Heat Losses and Heat Balance. The total amount of heat supplied to an internal-combustion engine in the fuel may be accounted for as follows: heat converted into work (brake horsepower); heat used in overcoming friction; heat given to the water jacket; heat carried away by the exhaust gases; and heat lost by radiation, leakage, etc. There are wide variations in the relative proportions of these items, depending upon the type, size, and operating conditions of the engine under consideration. For engines operating on the four-stroke Otto cycle, the distribution of heat averages: heat converted into work about 25 per cent, lost to friction 5 per cent, given to the cooling water 27 per cent, carried away in the exhaust gases 35 per cent, and radiation and other losses 8 per cent. For typical Diesel engines, the distribution of heat may be: converted into work about 30 per cent, lost to friction 9 per cent, given to the cooling water 25 per cent, carried away in the exhaust gases 28 per cent, and radiation and other losses 8 per cent.

The heat carried away by the exhaust gases is generally the largest loss, resulting from incomplete expansion, incomplete combustion, and sensible heat because of the high temperature (400 to 800 F) of the exhaust. This unavoidable loss can be reduced by continuing the expansion down to a lower pressure, but mechanical difficulties are encountered. By suitable means a large percentage of the exhaust loss and water-jacket loss may be recovered and utilized, as described in Art. 96.

The high temperature attained during explosion results in a very rapid extraction of heat by the cylinder walls. The higher the temperature, the more rapid is this loss through the cylinder walls to the water jacket. Cooling is necessary as a protection for the cylinder walls, to facilitate lubrication and to prevent pre-ignition. The large percentage of heat carried away by the cooling water does not indicate poor utilization of heat inside the engine cylinder. Much of the heat is transferred to the cooling water from the cylinder gases after combustion is completed, also while the gases pass through the cylinder head of a four-cycle engine or through the exhaust ports of a two-cycle engine. Practically all the piston friction work is also included in the cooling-water

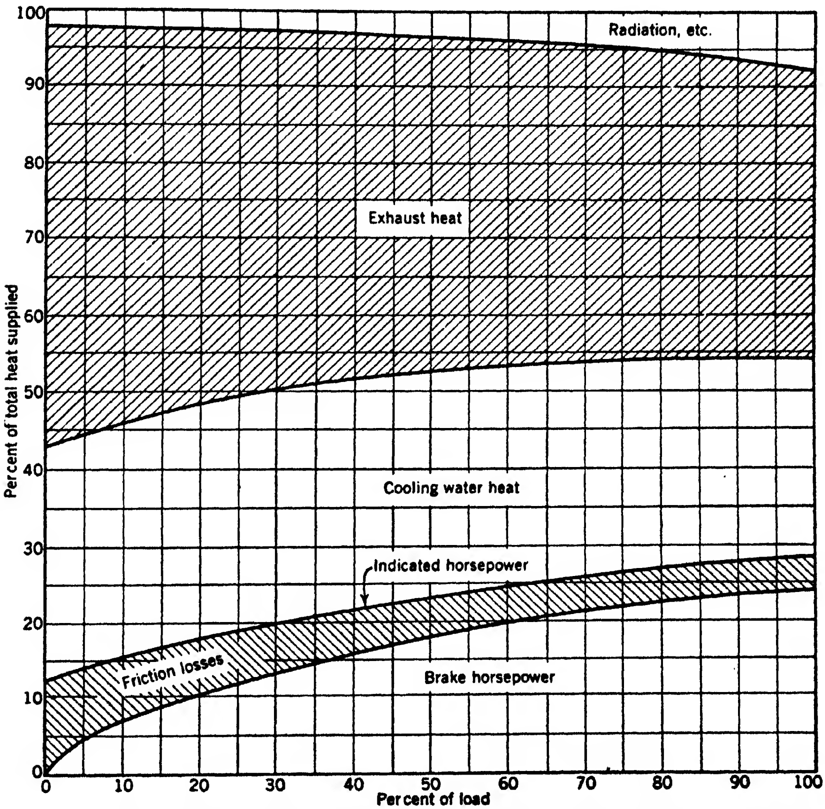


FIG. 150. Typical heat balance curves for small four-stroke cycle gas engine.

heat. If the amount of cooling water were decreased, the cooling water would receive less heat, but the temperature of the exhaust gases would begin to rise and more heat would be carried away in the exhaust.

The heat supplied to an internal-combustion engine in the fuel may be accounted for in a tabulated heat balance, and may be graphically represented as in Figs. 150, 151, and 152. These figures are typical for an Otto-cycle engine, a small four-cycle Diesel engine, and a large two-cycle Diesel engine, respectively, for loads from full load down to no load. The vertical distance between two adjacent curves represents the corresponding percentage in heat used for any given load. The indicated horsepower may be subdivided into brake horsepower and friction losses; the friction losses increase as the load decreases. The percentages of heat carried away by the exhaust gases and cooling water vary very little as the load changes. In Fig. 152 the recoverable heat in the exhaust gases is indicated as averaging about 40 per cent of the total heat in the exhaust gases; the heat absorbed by the cooling water has been

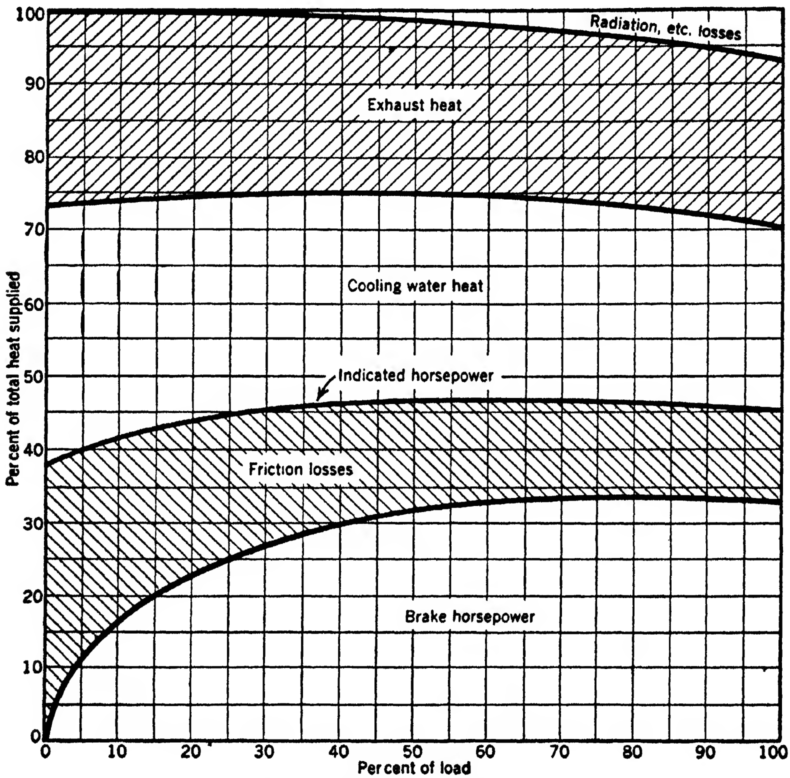


Fig. 151. Typical heat balance curves for small Diesel engine.

subdivided into that removed from the exhaust manifold, lubricating oil, and water jacket.

104. Fuel Consumption. Performance Curves. The fuel consumption, measured in cubic feet of gas per brake horsepower per hour or pounds of oil per brake horsepower per hour (W_f in equations 149 and 150), varies considerably with the different types of internal-combustion engines. The fuel used by engines of a given type decreases with increase of cylinder diameter; the decrease may be considerable as the size progresses. The single-cylinder engine is invariably the most economical for a given horsepower. However, practical difficulties limit the size to which cylinders may be built without complicating the design, so that multi-cylinder engines are used for large sizes and higher speeds.

The comparative results of tests on a four-cycle gas engine and a two-cycle gas engine (with and without gas injection), of identical dimensions and operating at the same speed, are shown in Figs. 31 and 32. The heat balances obtained from the tests are shown in Table XII, and a dis-

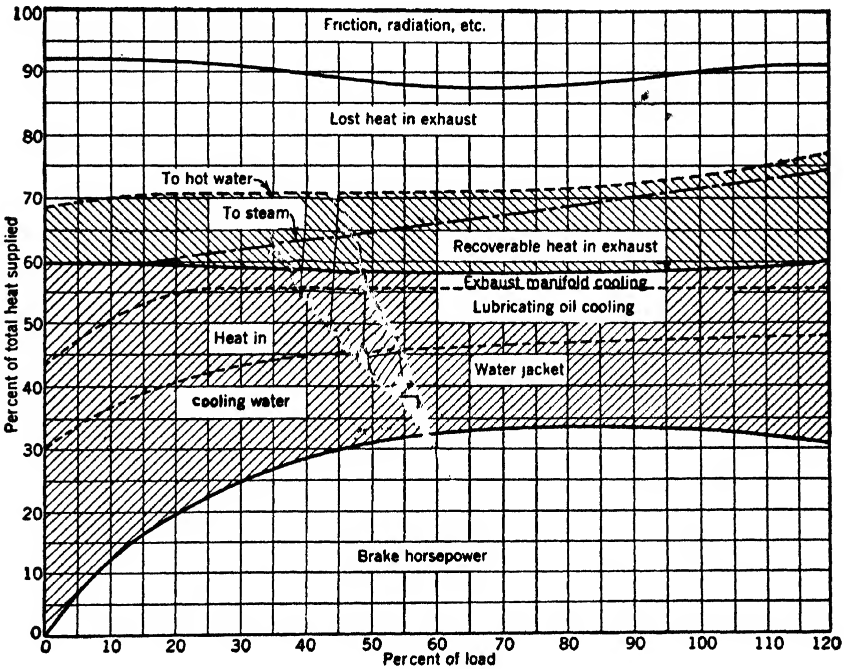


FIG. 152. Heat balance curves for 16 × 20-in., 300 rpm. 175 bhp per cylinder, two-stroke cycle, mechanical injection Diesel engine.

cussion of the results is given in the latter part of Art. 36. The highest thermal efficiency and the least fuel consumption were obtained at full load for each engine. Large modern gas engines are guaranteed to produce a brake horsepower-hour for each 9000 to 10,000 Btu supplied in the gas to the engine, corresponding to a thermal efficiency of 28.3 to 25.5 per cent.

For engines which operate at widely varying speeds, such as automobile and high-speed Diesel engines, tests at one speed only are not sufficient when complete performance data and results are desired. The performance curves, Figs. 153, 154, and 155, were obtained from dynamometer tests of the engines tested for maximum power at the different speeds, the load applied by the dynamometer being varied as required. This method of testing is similar to operating an automobile with wide-open throttle but changing the grade from level to a steep incline. The speed attained decreases accordingly. These three sets of curves show the relative performance curves of six-cylinder gasoline, spark-ignition, and Diesel engines, all within 4 cu. in. of the same displacement.

In a gasoline engine, where one carburetor supplies fuel to a number of cylinders through a common manifold, it is not possible to obtain

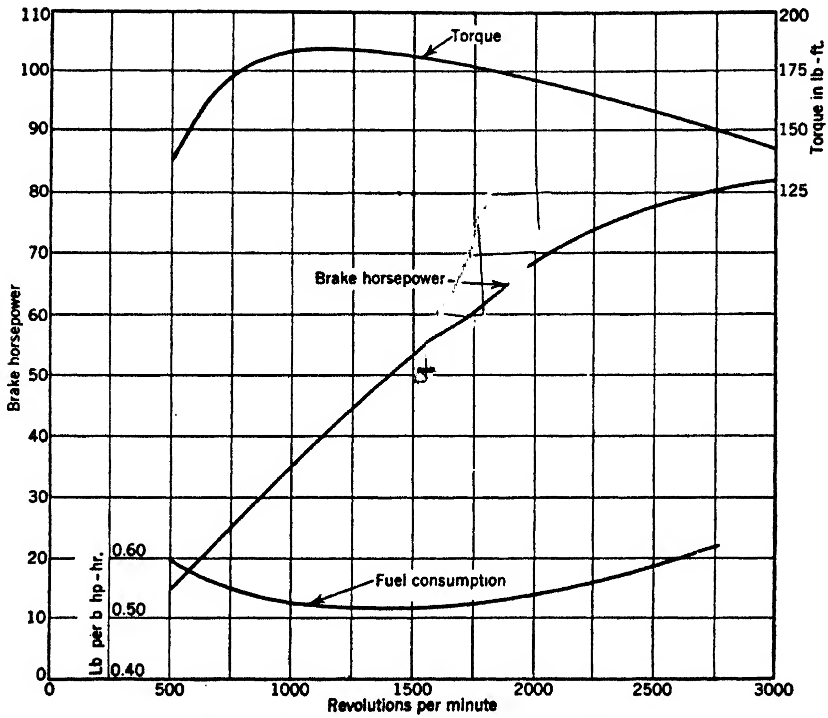


FIG. 153. Typical performance curves of a six-cylinder gasoline engine, 282 cu in. piston displacement, variable torque and speed.

equal distribution to all the cylinders. The only satisfactory way would be to use a separate carburetor for each cylinder or pair of cylinders. This complication would put the gasoline engine at a disadvantage in the matter of simplicity of fuel distribution as compared to the Diesel engine. In a Diesel engine, because of the exact metering and timing of the fuel supply, an equal distribution of the fuel to each cylinder in multi-cylinder engines is obtained and better fuel economy is effected. Except in cases of considerable injection advance, the Diesel engine produces a power stroke of long duration and high average torque over a fairly large speed range. The torque-speed curve is much flatter, Fig. 155, than that of a gasoline engine, Fig. 153; thus the Diesel engine is more flexible over a wider speed range.

Modern automobile engines have a fuel economy of 0.52 to 0.80 lb of gasoline per bhp-hr when operating at or near the rated load and speed, with an air-fuel ratio of 16 to 1 (by weight). The fuel consumption of the high-speed Diesel engine is appreciably less than the gasoline engine, as determined in terms of weight per brake horsepower per hour.

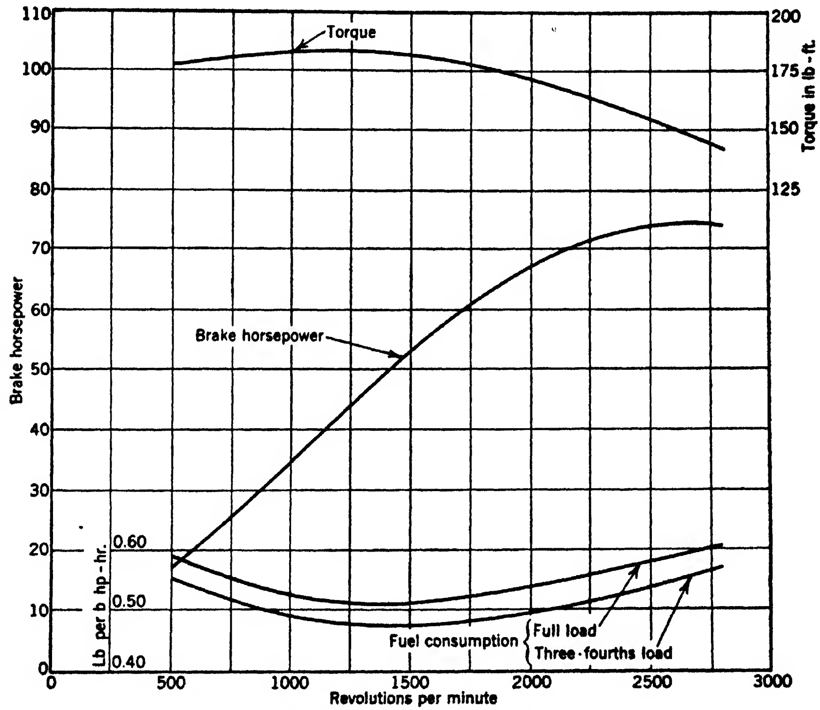


FIG. 154. Typical performance curves of a six-cylinder four-stroke cycle high-speed spark-ignition oil engine, 282 cu in. piston displacement, variable torque and speed.

However, fuels for both gasoline and Diesel engines are purchased on a volume basis, i.e., cost per gallon. The specific gravity of fuel oil is about 0.87, that of gasoline about 0.74; hence a gallon of fuel oil will weigh about 20 per cent more. In effect, this is equivalent to a further reduction in fuel costs for the Diesel engine. The popularity of the present-day gasoline engine, despite its higher fuel consumption, is in large part due to its low weight per horsepower, low cost, ease of starting, clear exhaust, less noise, early development and use as a high-speed engine, simplicity of parts, and ease of manufacture. A complete comparison of the advantages and disadvantages of gasoline engines and high-speed Diesel engines is given in Art. 66.

The spark-ignition oil engine, as explained in Art. 45, uses Diesel fuel oil without employing the high pressures of the Diesel cycle. This design is a compromise between the Diesel and the gasoline engine, utilizing as many of the benefits of high compression as possible without sacrificing the advantages of the low-compression engine. Performance curves show economies of 0.51 to 0.58 lb of fuel per bhp-hr at full load, with better

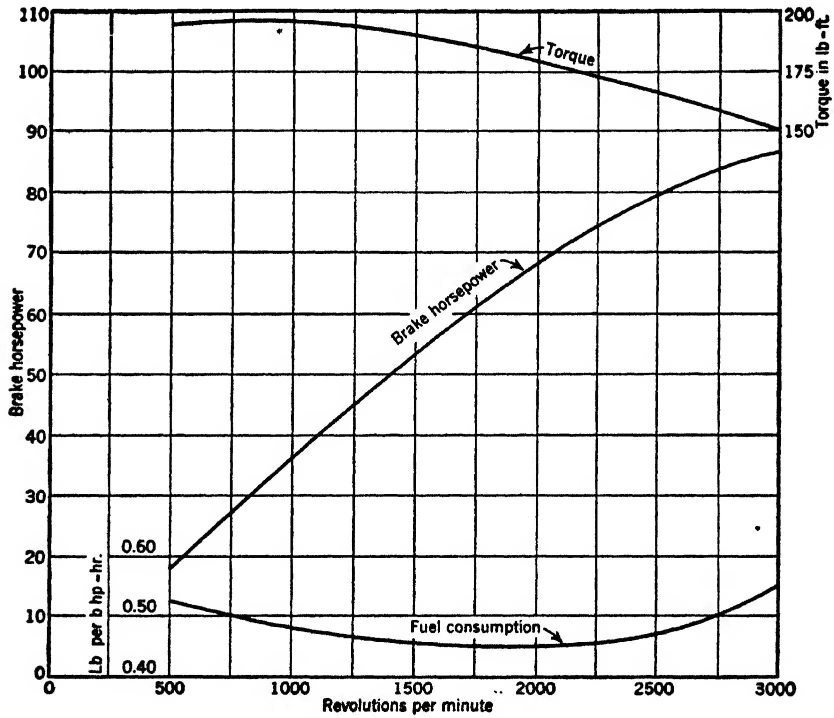


FIG. 155. Typical performance curves of a six-cylinder four-stroke cycle high-speed Diesel engine, 278 cu in. piston displacement, variable torque and speed.

economies at three-fourths load of 0.48 to 0.54 lb; see Fig. 154. This is slightly more than the fuel consumption of the high-speed Diesel engine, and considerably better than the fuel consumption of the automobile engine.

Low-pressure (80 to 150 lb per sq in. compression) oil engines consume approximately 0.55 to 0.80 lb of fuel oil per bhp-hr. The fuel consumption of moderate-pressure (200 to 300 lb per sq in. compression) oil engines is somewhat lower than for low-pressure engines, the performance being about as shown in Fig. 154. The fuel consumption of the

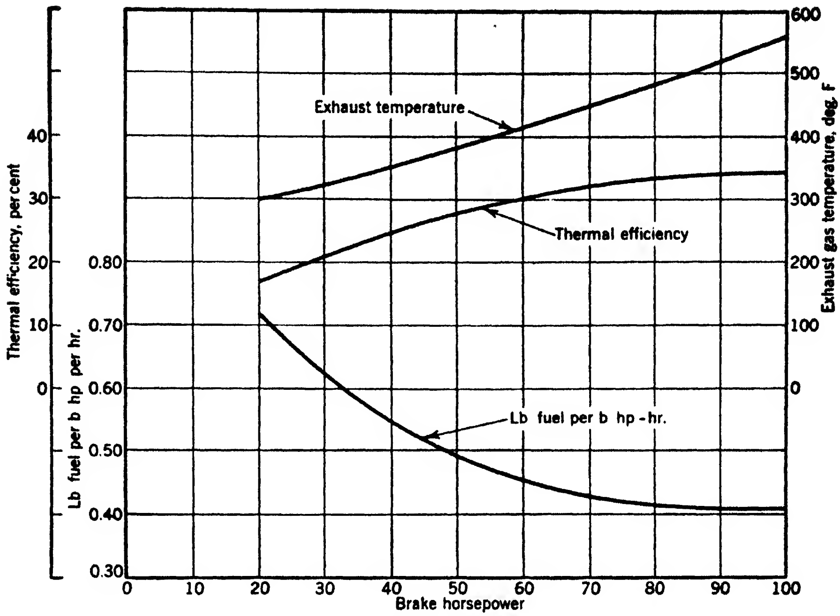


FIG. 156. Performance curves of a 100-bhp medium-speed Diesel engine, 600 rpm, six-cylinders, $6\frac{1}{2} \times 8\frac{1}{2}$-in., four-stroke cycle, fuel oil 18,500 Btu per lb, mechanical injection, and direct-connected to a generator.

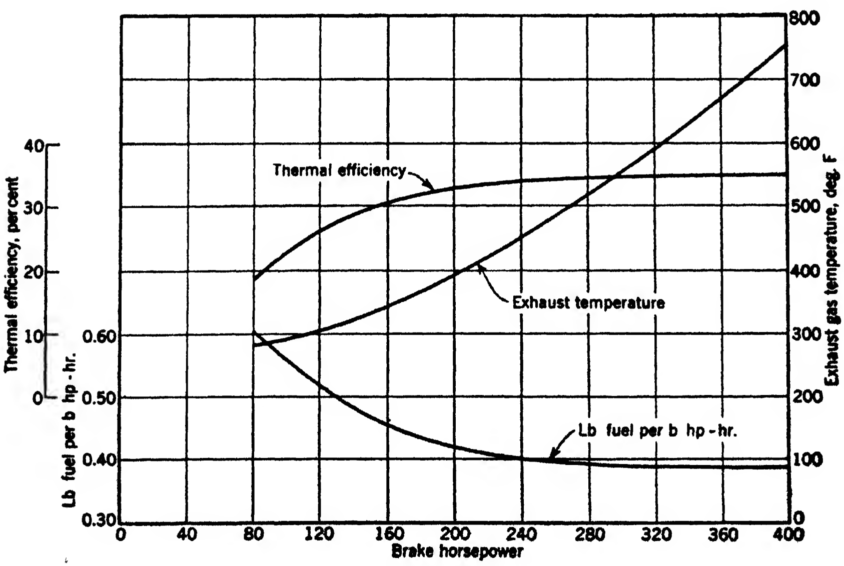


FIG. 157. Performance curves of a 375-bhp Diesel engine, 300 rpm, five-cylinders, $13\frac{3}{4} \times 18$-in., four-stroke cycle, fuel oil 19,000 Btu per lb, mechanical injection, and direct-connected to an alternator.

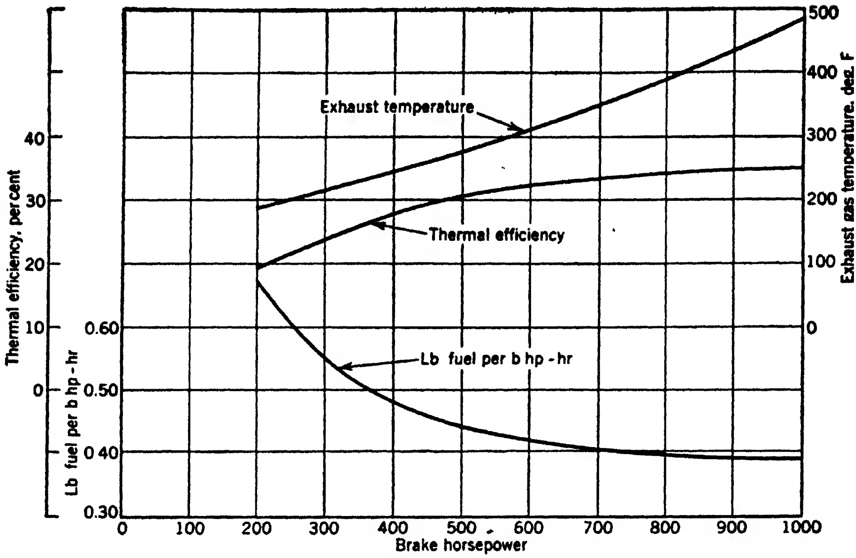


FIG. 158. Performance curves for a 1000-bhp Diesel engine, 300 rpm, six-cylinders 16 × 20-in., two-stroke cycle, fuel oil 19,000 Btu per lb, mechanical injection, pum scavenging, and direct-connected to an alternator.

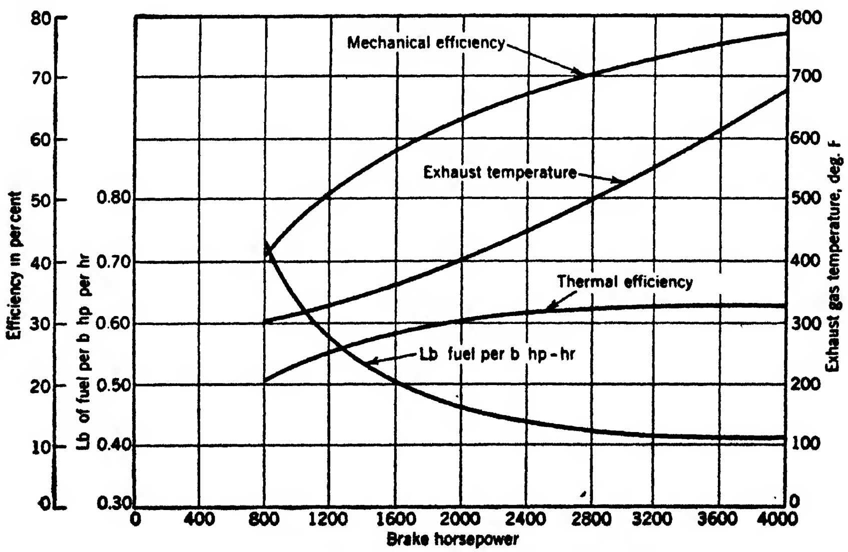


FIG. 159. Performance curves of 4000-bhp Diesel engine, 125 rpm, six-cylinders single-acting, two-stroke cycle, fuel oil 18,740 Btu per lb, air injection, and direct connected to generator.

Diesel (compression-ignition) engine varies from 0.48 to 0.38 lb per bhp-hr from one-half to full load, which corresponds to a thermal efficiency of 28 to 35 per cent. The rapid growth of the Diesel engine industry may in large part be attributed to this economical use of fuel and resultant high efficiencies.

Typical performance curves of Diesel engines, varying in size from 100 to 4000 bhp, for the most general types of design, operating under various loads are shown in Figs. 156, 157, 158, and 159. The temperature of the exhaust gases is higher in the four-cycle than in the two-cycle engine, compare Figs. 157 and 158, owing to the use of scavenging air in the two-cycle engine. A discussion of the relative advantages and disadvantages of the two-stroke cycle and four-stroke cycle Diesel engines is given in Arts. 57 and 58. Some Diesel engine tests have shown fuel economies of less than 0.40 lb per bhp-hr with a resultant thermal efficiency of 35 to 40 per cent. These figures give some idea of the utility of internal-combustion engines and the future possibilities for this type of prime mover, when considered from a standpoint of low fuel consumption and minimum operation costs. Also the same fuel consumption may be obtained in small engines as in large engines, compare Figs. 155 and 156 with Figs. 157, 158, and 159; also the fuel used in pounds per brake horsepower-hour varies but little with the changes in power between one-half and full load.

PROBLEMS

1. A four-cycle, single-acting, 12-in. by 18-in. gas engine with single cylinder was operated at 200 rpm and made 98 explosions per minute. The gross weight on the brake was 260 lb, the tare 40 lb, and the length of the brake arm 60 in. The area of the indicator card was 1.44 sq in. and the length 3 in. The scale of the indicator spring used was 200 lb. Find: (*a*) the indicated horsepower developed; (*b*) the brake horsepower delivered; (*c*) the friction horsepower; (*d*) the engine torque; (*e*) the mechanical efficiency; and (*f*) the thermal efficiency based on brake horsepower, if 400 cu ft of natural gas were supplied per hr, higher heating value of gas being 1050 Btu per cu ft.

2. A four-cycle, single-acting, 12-in. by 21-in. gas engine with single cylinder rated at 40 bhp had a gas consumption of 12,500 Btu per bhp-hr. Find: (*a*) the thermal efficiency; (*b*) the cost of operation per hour at rated load, if the gas had a higher heating value of 500 Btu per cu ft and cost $0.75 per 1000 cu ft; and (*c*) how the cost of gas compares with the cost of operating an electric motor (to deliver 40 bhp) that is 85 per cent efficient, when power costs 3 cents per kw-hr.

3. A four-cycle, single-acting, 13½-in. by 17-in. gas engine with single cylinder rated at 60 bhp required 24 cu ft of gas per bhp-hr at 200 rpm; higher heating value of gas supplied was 460 Btu per cu ft. Find: (*a*) the suction displacement in cubic feet per brake horsepower per minute; (*b*) the mep based on rated brake horsepower; (*c*) the Btu supplied to the cylinder per cubic foot of suction displacement: and (*d*) the thermal efficiency in percentage

4. A gas producer yielded 70 cu ft of gas containing 115 Btu per cu ft from each pound of coal containing 11,000 Btu per lb. The gas engine supplied by this producer used 95 cu ft of gas per bhp-hr. Find: (*a*) the thermal efficiency of the producer; (*b*) the thermal efficiency of the engine based on bhp; and (*c*) the combined thermal efficiency of producer and engine.

5. A $10\frac{1}{2}$-in. by 18-in. four-cycle and single-acting Diesel engine, used 22 lb of fuel oil per hr at 250 rpm. A prony brake with an arm $31\frac{1}{2}$ in. long was attached to the engine; the gross weight on the brake was 330 lb, and the tare was 30 lb. Find the thermal efficiency of the engine based on brake horsepower if the fuel oil had a higher heating value of 19,000 Btu per lb.

6. A two-cycle, single-acting, 29-in. by 44-in. Diesel engine with six cylinders was guaranteed to deliver 3750 bhp at 125 rpm with a fuel consumption of 0.40 lb per bhp-hr; higher heating value of fuel oil was 19,000 Btu per lb. Find: (*a*) the suction displacement in cubic feet per brake horsepower per minute; (*b*) the mep based on brake horsepower; (*c*) the Btu supplied to the cylinder per cubic foot of piston displacement; and (*d*) the thermal efficiency in percentage.

7. A nine-cylinder radial airplane engine, when tested at constant speed and load, required 8 lb of gasoline in 6.58 min. The engine turned 8883 revolutions in this time. The torque due to the brake load, as measured by a cradled electric dynamometer, was 490 lb-ft. What was the fuel consumption in pounds per brake horsepower-hour?

8. A marine Diesel engine that delivered 3330 bhp had a mechanical efficiency of 80.3 per cent. The thermal efficiency based on the brake horsepower was 33.2 per cent, and the heat lost to the water jacket was 22 per cent. How many gallons of cooling water were circulated per minute, if the water entered the jacket at a temperature of 80 F and left at 120 F? Use 62 lb per cu ft of water.

9. The following data were obtained during a test of a four-cycle, single-acting, 12-in. by 18-in. Diesel engine with single cylinder: duration of test, 45 min; fuel oil used, 12 lb; total revolutions, 9090; jacket water, 1122 lb; rise of temperature of jacket water, 50 F; mep in cylinder, 94 lb per sq in.; torque due to brake load, 840 lb-ft; higher heating value of fuel oil, 19,000 Btu per lb. Find: (*a*) the indicated and brake horsepowers, also the mechanical efficiency; (*b*) the oil used, pounds per indicated horsepower-hour and pounds per brake-horsepower-hour; (*c*) the percentage of total heat supplied that was converted into brake horsepower; (*d*) the percentage of total heat supplied that was absorbed by the water jacket; and (*e*) the percentage of heat lost by friction, exhaust gases, radiation, etc.

10. An automobile engine has eight cylinders $3\frac{1}{2}$ in. by $4\frac{1}{4}$ in. and delivers 130 bhp at 3400 rpm. (*a*) Find the SAE horsepower rating at both 1000 ft per min piston speed and at 3000 rpm. (*b*) Find the mep based on brake horsepower. (*c*) If the rear axle gear reduction is 4.22 to 1 and the tires are 29.5 in. in diameter (external), find the miles per hour of travel. Assume 10 per cent slippage of tires on roadway. (*d*) How many miles can be traveled per gallon of gasoline (6.4 lb per gallon) if 40 Btu of gasoline-air mixture are required per cubic foot of suction displacement. Heating value of the gasoline, 20,000 Btu per lb.

11. Determine the cylinder size of a six-cylinder automobile engine to deliver 125 bhp at 3000 rpm when supplied with a gasoline-air mixture having a heating value of 68 Btu per cu ft of suction displacement. Assume a cylinder pressure of 13.5 lb at the beginning of compression and 115 lb per sq in. abs at the end of the compression stroke; the equation of the compression curve is $PV^{1.32} = C$. The stroke-bore ratio is 1.2 and the mechanical efficiency 85 per cent.

12. A four-cycle, single-acting, internal-combustion engine with single cylinder used natural gas of the following analysis, percentages by volume: CH_4, 92.6; C_2H_6, 4.3; CO_2, 0.6; and N_2, 2.5. The engine delivered 80 bhp and operated at 200 rpm; thermal efficiency based on ihp, 0.22; mechanical efficiency, 82 per cent; and volumetric (charging) efficiency, 88 per cent. (*a*) Find the higher heating value per cubic foot of gas. (*b*) Calculate the cubic feet of air supplied per cubic foot of gas, assuming 25 per cent excess air. (*c*) Find the piston displacement in cubic feet per minute. (*d*) Determine the cylinder size (bore and stroke) if the stroke is 1.5 times the diameter.

13. A $3\frac{3}{4}$-in. by $4\frac{3}{4}$-in. automobile engine with six cylinders has a clearance volume of 20 per cent and a brake thermal efficiency of 20 per cent. Find the maximum power which the engine can deliver, running at 2000 rpm, if at the end of each suction stroke the entire cylinder is filled with an explosive mixture of gasoline and air, having a heating value of 57 Btu per cu ft.

14. Gas engine 179 at the Illinois Steel Company is four-cylinder, twin-tandem, double-acting, and direct-connected to a generator. The cylinders are all alike, 48-in. by 60-in., with a 16-in. piston rod passing through both ends of all cylinders. The engine is four-cycle and operates at 83 rpm on blast furnace gas of 92.5 Btu per cu ft. During a test the net output of the generator at the switchboard was 3800 kw. The indicator cards had an average mep of 80 lb per sq in. The engine required 12,400 Btu (at engine) per kw-hr at the switchboard; thermal efficiency of the engine based on ihp was $27\frac{1}{2}$ per cent. Find (*a*) the indicated horsepower of the engine; (*b*) the mechanical and electrical efficiency (combined efficiency); (*c*) the suction displacement in cubic feet per indicated horsepower per minute; (*d*) the Btu per cubic foot of suction displacement based on indicated horsepower; (*e*) the amount which the original volume of gas has been increased and diluted at the end of the suction stroke.

15. On test a four-cycle, single-acting, $20\frac{1}{2}$-in. by $35\frac{1}{2}$-in. Diesel engine with six cylinders gave the following data: 136 rpm, 843 bhp, 32.3 per cent (brake) thermal efficiency, and 18,600 Btu per lb higher heating value of fuel oil. (*a*) Find the pounds of fuel used per brake horsepower-hour. (*b*) Each cylinder has an individual fuel oil pump whose stroke is twice its diameter. The oil weighs 60.4 lb per cu ft. Find the diameter and stroke of each pump. (*c*) Find the cubic feet of suction displacement per brake horsepower per minute. (*d*) Determine the Btu supplied per cubic foot of suction displacement.

16. An internal-combustion engine develops 600 hp at standard conditions of dry air. (*a*) What would it probably develop at 5000 ft above sea level with a temperature of 80 F and a relative humidity of 50 per cent? (*b*) What would be the probable horsepower at 20,000 ft above sea level with a temperature of −15 F? Refer to Table XXIII.

17. An automobile engine develops 100 hp when the barometer is 26.82 in. Hg, temperature is 40 F, and humidity is 80 per cent. How much power would the engine develop at standard conditions and with dry air? Also determine the effect of each factor separately.

18. A Diesel engine delivers 450 bhp when the barometer reads 29.25 in. Hg. The engine is supplied with an air-fuel ratio of 20 to 1, by weight. The air has a relative humidity of 90 per cent at 90 F. (*a*) What is the dry-air pressure? (*b*) What is the air-fuel ratio on a dry-air basis? (*c*) How much power would the engine deliver at standard conditions of dry air?

19. An eight-cylinder $3\frac{1}{4}$-in. by $4\frac{1}{4}$-in. automobile engine developed 120 hp at 3200 rpm. The fuel consumption was 0.75 lb per hp-hr and the air-fuel ratio was 10 to 1 (by weight). Find the volumetric efficiency of the engine.

20. A $4\frac{1}{2}$-in. by 6-in. four-cycle, high-speed Diesel engine with six cylinders used 0.45 lb of fuel oil per bhp-hr when delivering 100 bhp at 2000 rpm. The fuel used contained 0.85 C and 0.15 H_2 by weight. (*a*) Calculate the approximate excess of air supplied per pound of fuel; use a volumetric efficiency of 80 per cent. (*b*) Find the brake mep.

21. A 14-in. by 21-in. four-cycle Diesel engine with four cylinders developed a mep of 100 lb per sq in. on full load at 200 rpm. The fuel-consumption rate was 0.32 lb per ihp-hr with fuel oil of 19,000 Btu per lb. If 20 per cent of the heat supplied was carried away by the jacket water, calculate the probable quantity of cooling water required for the jackets in gallons per hour if the rise in temperature was 100 F.

22. Assume that the exhaust gases from the engine of problem 21 left the cylinders with 25 per cent of the fuel heat and that 40 per cent of this heat was recovered in a waste-heat boiler generating dry steam at 18 lb per sq in. abs. The feedwater for this boiler is supplied by the water leaving the jackets at 170 F. Calculate the quantity of steam generated in pounds per hour.

23. For a 900-bhp Diesel engine, the mechanical efficiency is 85 per cent, the jacket loss is 25 per cent, and the thermal efficiency based on ihp is 30 per cent. The jacket water enters at 90 F and leaves at 155 F; how many pounds of water must be circulated per minute?

24. A 24-in. by 33-in. four-cycle, double-acting, gas engine with six cylinders had a clearance of 18 per cent and operated at 140 rpm. The indicated thermal efficiency was 60 per cent of the ideal efficiency. The gas contained 145 Btu per cu ft and required 1.6 cu ft air per cu ft gas for complete combustion; excess air supplied was 20 per cent. The pressure at the end of the suction stroke was 13.9 lb abs and the temperature 155 F. Find: (*a*) indicated thermal efficiency; (*b*) volumetric efficiency; (*c*) Btu per cubic foot of mixture as it enters the cylinder, and at standard conditions of 60 F and 14.7 lb per sq in.; (*d*) the mep; and (*e*) the total indicated horsepower of the engine. The engine was supplied with 1590 cu ft of gas per min.

25. Determine the diameter of the cylinder and stroke of an eight-cylinder gasoline engine which is to deliver 175 bhp at a piston speed of 1500 ft per min and give a thermal efficiency (based on bhp) of 25 per cent. The stroke-bore ratio is to be 1.2, the volumetric efficiency 90 per cent, and the mechanical efficiency 88 per cent. Assume gasoline to contain 20,000 Btu per lb and to require 15 lb of air for complete combustion; also assume gasoline vapor as 3 times the weight of air and as using 15 per cent excess air. Assume air at end of the suction stroke to be at 200 F and 13 lb per sq in. abs.

CHAPTER XV

ECONOMICS OF GAS AND OIL POWER

The widespread use of electric power among the vast majority of our population has demonstrated that the city or state must take the responsibility of providing some kind of regulatory measures to assure a continuous source of electricity to its citizens at a reasonable price. The question of private or public ownership of electric utilities will not be discussed here, because for a given plant there is little or no difference between the power generation costs regardless of ownership. In this chapter is given a discussion of the application, merits, and economics of internal-combustion engines for the production of electric power for distribution and sale, also for industrial and isolated plants where these simple, compact, and efficient power-generating machines are economically justifiable.

105. Selection of a Prime Mover. The selection of the type of power plant—hydraulic; reciprocating steam engine or steam turbine, with oil- or coal-fired boilers, high or low pressure; as against oil or gas engines—is an economic problem. The growth of the size of the units and the steady increase in total horsepower and number of units in use, supplemented by the authenticated profitable results of operations of many plants, both private and municipal, in this country and abroad, prove that the internal-combustion engine is a good investment when properly selected and utilized. It is not always the most economical source of power, because local conditions such as kinds and prices of available fuel, regularity of supply of fuel, load factors, especially peak load demands, seasonal variations in load, and probable growth of load, are important features to be studied. Other factors to be considered are fuel-storage space, the important question of quality and quantity as well as price of water, and the heating load, if any. In general, questions of personnel for operating the plant, occupancy of adjoining property, as bearing on noise, vibration, smoke, fine ashes and odors, as well as space occupied and cost of the building, have relatively little bearing on the selection of type, because there is, in these respects, little difference between oil-engine and steam plants when the specifications are equally good.

The design of power plants of less than 100,000 horsepower should be based on the assumption that steam turbines and oil engines are equally available and equally reliable and that the choice between the two must be made for each installation on the economics of the case after making an adequate comparison of first cost and operating charges, including capital charges. If the choice is a Diesel engine, the installation should be just as complete and as good as the steam installation would have been if it had been selected. There are good steam plants, and cheap, inefficient installations. There are good Diesel plants; but, unfortunately, there have been many poor, inadequate installations. Comparisons should be made on the basis of equal quality. An incomplete, overrated Diesel plant will not have the reliability of a good steam plant; nor will a poor, inefficient steam plant either have the earning power or give the continuity of service of a first-class Diesel plant.

In designing a power plant, it is considered good engineering to select generating units of such size that the largest unit may be down for repairs during the time of the peak load on the plant without causing curtailment of service. In the June 30, 1934, issue of the *Electrical World,* Mr. J. S. Hartt presents "Service Records of Diesel Plants in Wisconsin." He concludes that only two out of ten municipal Diesel engine plants in Wisconsin had sufficient installed capacity to assure continuity of service, and that if adequate equipment were installed in the other eight plants the additional fixed charges would considerably increase plant output costs.

The insistent demand for larger prime movers has resulted in the growth of the steam turbine to more than 200,000 horsepower per unit, with the practical elimination of the reciprocating steam engine for central-station service, and the growth of the Diesel engine to more than 20,000 horsepower units. Meanwhile, the gas engine for blast-furnace, pipe-line, and similar applications has considerably increased in the number and horsepower of installations; but the Diesel has practically eliminated the gas engine in the central-station field and has kept the same relative position of being about 10 per cent of maximum unit output in relation to steam turbines.

Large steam plants are more economical than small steam plants because steam plants increase rapidly in efficiency with increases in plant size, and the investment per horsepower decreases. On the other hand, the small Diesel engine is almost as efficient as the large Diesel and costs but little more, if any, per unit of capacity. The plant composed of several small Diesel engines may, and should, be more efficient than a plant having one large engine of equal plant capacity because of the inherent flexibility of the small units. Thus the gasoline or the Diesel engine may be the logical choice for small isolated plants. It seems unlikely that the large steam plant will encounter much competition from the large Diesel plant; however, the small and medium condensing steam station (500 to 5000 hp) will encounter increased Diesel competition.

The progress made in the development and applications of large Diesel engines has advanced more rapidly in Europe than in the United States. In this country, coal is cheaper than oil in many localities; and with modern utility steam plants developing a kilowatt-hour on a pound of coal, the fuel cost is so low that Diesel fuel must sell at about three cents per gallon to attain parity. In the oil fields, where oil is cheap, it happens that natural gas is obtainable at so low a price that the fuel item in large plants is almost negligible. Whereas large Diesel engines are employed to carry the peak load of European utilities, in this country most systems have excess steam capacity, and the peak is handled by older, less-efficient plants without additional investment.

106. Costs of Gas and Oil Power. The most expensive power for industrial and small municipal installations will usually be steam because of the inherent low thermal efficiency of low-pressure steam. The overall efficiency of a low-pressure non-condensing engine drive is seldom more than 5 per cent. Purchased electric power, Diesel-engine drive, and gas-engine drive are more competitive. The cost of electric power will vary with the locality, load, and demand; the cost may not exceed $1\frac{1}{2}$ cents per kw-hr, but an annual average of purchased electric power for small industries is more nearly 2 cents per kw-hr. The Diesel engine has a high thermal efficiency; to offset this fact, its fuel costs are frequently high, to which must be added freight, handling charges, and in some states a tax. One gallon of fuel oil contains approximately 140,000 Btu; at 4 cents per gal this is the equivalent of about 30 cents per 1000 cu ft of natural gas, which is not an unusual rate for this clean, desirable fuel.

Because of its high-pressure cycle the Diesel engine is heavier and consequently more costly than the gas engine. Wherever gas is available at 35 cents per 1000 cu ft or less, the gas engine will provide the cheapest power for small plants, unless unusual conditions exist. A recent gas-engine installation consisting of three 260-hp engine-generator units, with exciters and switchboard, cost about $50.00 per horsepower. The fuel consumption (at full load) for good two-cycle and four-cycle gas engines varies from 10 to 12 cu ft per hp-hr, based on 1000 Btu per cu ft gas. Articles 36 and 37, also Table XIII, give additional information on the applications and performance of various types of gas engines. In recent years the gas engine has begun to assume a place of importance as a prime mover in sewage disposal plants, using the gases of sludge digestion for fuel, as explained[1] on page 21 and included in the list of gas engines given in Table XIII.

[1] For more complete information refer to "Operation of a 200-Hp Gas Engine Using Sludge Gas for Fuel," C. F. Tennant, *Sewage Works Journal*, January, 1936.

Diesel engines smaller than 500 hp usually cost less per horsepower installed than do steam plants; for sizes between 500 and 2000 hp the cost is approximately the same. The cost usually ranges from \$60.00 to \$90.00 per horsepower, with an average of \$70.00; this indicates a cost of approximately \$90.00 per kilowatt, plus \$15.00 to \$30.00 per kilowatt for generator, exciter, foundation, switchboard, etc. Installation costs will vary considerably, depending upon the type of building, type of foundation, cooling water system, regulation, and types of accessories and auxiliaries.

Table XXIV covers a yearly summary of the operating costs and fixed charges of four Diesel engines in a municipal electric-light plant. The capacity of each unit is 850 kilowatts (engine size 1250 bhp); one unit had been installed for 7 years, another unit for 6 years, and two units for 2 years when the records for Table XXIV were compiled. The population of the city served was 20,000, and the number of power and light connections was 5522. The net kilowatt-hour output is found by subtracting the power used for plant auxiliaries and lighting from the total gross power produced by the plant. The running plant capacity factor is the ratio of the annual plant output in gross kilowatt-hours and the total annual rated kilowatt-hours of the individual units. The annual plant load factor is the ratio of the annual plant output in gross kilowatt-hours and the total kilowatt-hours if the peak load were produced continuously throughout the year.

The investment in the above Diesel-electric light plant is about \$340,000.00, and the building and distribution outlay approximates \$300,000.00; hence the total municipal light and power investment is \$640,000.00, or about \$32.00 per person in the community. For smaller communities the cost per person varies from \$35.00 to \$50.00. The plant investment includes cost and installation of four engines direct-connected to alternators, V-belted 15-kw exciters, cooling tower, fuel and lubricating oil centrifuges, air cleaners, silencers, heat exchangers, pumps, piping, foundation, switchboard, and wiring complete. The building and distribution investment costs include land, building, trackage, distribution system complete, street lighting, and miscellaneous. Even though the engines were running at 64 per cent of capacity, the fuel consumption per brake horsepower-hour and production cost per kilowatt-hour were low, being 0.53 lb per bhp-hr and 6.4 mills per kw-hr, respectively. In those larger centers where electric power can be purchased for less than 2 cents per kw-hr, it is doubtful whether the Diesel engine could be employed to advantage. However, in communities where electric power rates exceed 2 cents and where the power demand is greater than 100 kw. the community or manufacturer could probably install a Diesel engine to advantage and thereby reduce power costs; see

TABLE XXIV

SUMMARY OF YEARLY DATA AND COSTS FOR MUNICIPAL DIESEL-ELECTRIC LIGHT PLANT

Population of city, 20,000	Engines and auxiliaries installed,	$340,000.00
Power and light connections, 5522	Land, building, distribution, etc.,	300,000.00

General Data

Kilowatts capacity, 4 units at 850 kw each		3,400
Kilowatt-hours produced (gross) per year		8,525,800
Kilowatt-hours plant output (net) per year		8,068,500
Kilowatt-hours sold per year		7,641,890
Peak load for year, kilowatts		2,550
Running plant capacity factor, per cent		64
Annual plant load factor, per cent		38
Fuel oil, gallons per year		815,299
Lubricating oil, gallons per year		4,333
Kw-hr (gross) per gallon of fuel oil, average for 4 engines		10.3
Kw-hr (gross) per gallon of lube oil, average for 4 engines		1,852
Fuel oil cost per gallon, average for year, cents		3.82
Lubricating oil cost per gallon, average for year, cents		50
Fuel oil per kw-hr produced (gross), average for year, lb		0.70
Fuel oil per bhp-hr of 4 engines, average for year, lb		0.53

Total and Unit Cost per Year

Plant Costs, Variable		
Salaries and wages, office and plant	$12,458.20	
Fuel oil	31,144.42	
Lubricating oil	2,166.50	
Plant supplies	769.62	
Maintenance and repairs	3,472.08	
Miscellaneous	1,584.14	
Total variable cost per year		$ 51,594.96
Variable cost per kw-hr output (net)	$0.0064	
Plant Costs, Fixed		
Insurance	$ 1,420.77	
Depreciation, 4½ per cent	15,300.00	
Interest, 5 per cent	17,000.00	
Total fixed charges and overhead per year, plant		33,720.77
Total cost of power per year at switchboard		$ 85,315.73
Total cost per kw-hr plant output (net)	$0.0106	
Distribution Cost, Power Plant to Consumer		
Salaries	$ 2,850.00	
Material	986.41	
Miscellaneous	2,258.53	
Insurance	625.36	
Depreciation, 6 per cent	18,000.00	
Interest, 5 per cent	15,000.00	
Total distribution cost (power plant to consumer)		39,720.30
Total cost of power per year delivered to consumer		$125.036.03
Delivered cost per kw-hr (net) produced	$0.0155	
Delivered cost per kw-hr consumed (sold)	$0.0164	

It should be remembered that production or operating costs are by no means total costs; the latter comprise production costs, interest on investment, depreciation, insurance, taxes, and in some cases distribution charges. In the July 28, 1934, issue of the *Electrical World*, Mr. J. S. Hartt presents the article "What Electricity Costs in Municipal Diesel Plants." He says: "Cost per kw-hr of Diesel-engine generated electricity at the switchboards of ten such municipal utilities in Wisconsin for 1933 ranges from 1.589 to 3.435 cents, with the highest cost not in the plant of least output." If a longer period of time and a larger group of plants had been used, these proportions might be changed materially. For these ten plants the average total cost per kilowatt-hour of plant output was 2.71 cents, with fuel oil at an average delivered price of about 5 cents per gallon.

Recent advances in Diesel power-plant design and greater dependability of the engine and its auxiliaries have changed total plant costs. Instead of installing double the needed capacity in expensive reliable slow-speed engines, one-half for standby service, the plant of today frequently has a limited standby capacity, consisting of an additional small high-speed Diesel engine for carrying the base load as emergency service. Since the Diesel generating unit can be started so quickly and is most economical at or near full load, many modern plants use several small engines instead of one large unit, starting and stopping them successively as required by load conditions. This multiple-unit plan is favored by the fact that there is very little difference in first cost per horsepower for small and large engines; and since smaller engines usually operate at high speeds, the smaller generators will be cheaper. In addition to reduced operating cost and increased flexibility, the multiple-unit plant will have higher availability, because when one engine is incapacitated, a smaller proportion of the plant will be out of service.

Table XXV shows a summary, prepared by an engine manufacturer, of the estimated cost of a 10,850-kw Diesel-electric light plant consisting of 5 main generating units of 2100-kw each; two-cycle, mechanical injection engines, operating at 240 rpm. The total cost of the complete power plant has been subdivided into its various component parts; the proportionate costs per kilowatt for each part is also given. The cost of the various items listed in Tables XXIV and XXV, and the method of calculation may differ considerably for other plants; hence the values given should be considered only as representative figures. With slight modifications for variations in conditions which may be encountered in other installations, these figures may be used in the study of the production and distribution of electric energy for other Diesel plants. Unit figures will be subject to some variation of local conditions affecting the design, construction, and operation of each Diesel engine installation.

TABLE XXV

ESTIMATED COST OF 10,850-KILOWATT DIESEL-ELECTRIC LIGHT PLANT

General Data		
Kilowatts capacity, 5 main generating units at 2100 kw each (240 rpm)		10,500
Kilowatts capacity, 1 auxiliary unit at 350 kw		350
Total station capacity, kilowatts		10,850
Estimated Cost	Total	Per kw
Land and building, complete	$ 88,500.00	$ 8.16
Cranes	10,500.00	0.97
Spur railroad track	3,000.00	0.28
Machinery foundations	20,000.00	1.84
Main generating units, at factory	585,000.00	53.91
Auxiliary generating unit, at factory	21,000.00	1.84
Station piping and ducts	16,500.00	1.52
Water pumps	3,500.00	0.32
Fuel storage, pumps, and centrifuge	10,500.00	0.97
Lubricating oil storage, pumps, and centrifuge	3,000.00	0.27
Switchboard, and station wiring	38,500.00	3.55
Miscellaneous accessories and auxiliaries	13,500.00	1.24
Freight and erection of equipment	37,000.00	3.41
Engineering expense	35,000.00	3.22
Total investment, based on 10,850-kw capacity	$885,500.00	$81.50
Variable Cost, 60 per cent capacity factor, 55,200,000 kw-hr		
Salaries and wages, 1 chief engineer	$ 4,800.00	
3 assistant engineers	9,000.00	
3 operating engineers	6,300.00	
3 helpers	3,600.00	
Fuel oil, 50,000 gallons at 4 cents	200,000.00	
Lubricating oil, 30,700 gallons at 60 cents	18,420.00	
Maintenance and repairs	15,600.00	
Total variable cost per year		$257,720.00
Fixed Charges		
Interest,* 6 per cent	$ 53,130.00	
Sinking fund,† 25-year basis, 1.82 per cent	16,116.10	
Insurance and taxes, 2 per cent	17,710.00	
Total fixed charges per year		86,956.10
Total cost of power per year at switchboard		$344,676.10
Total estimated cost per kw-hr plant output	$0.00624	

* Annual charges should be calculated on the cost value minus accrued depreciation to date.

† Under the sinking fund method of calculating depreciation, the annual depreciation charge increases as a geometric series having unity plus the interest rate as the ratio.

NOTE. If the annuity method of depreciation were used and the interest-on-investment item were omitted, the same results would be obtained.

The cost studies of the various engine arrangements and possibilities for a Diesel-engine generating plant will indicate which type, size, and combination will be most economical. The great number of variable quantities and values involved makes widely different total costs possible, one of which will likely be best for some specific case, but others of which may have considerable merit. In certain fundamental respects the assumptions for the problem are incomplete, such as, for instance, an assumption in regard to the rate of increase of business of the community served. These facts have not been overlooked, but have merely been disregarded in order to relieve the problem of unnecessary complications. For reliable cost data on plants of various sizes, the reader is referred to the latest annual "Report on Oil-engine Power Cost" by the Oil and Gas Power Division, American Society of Mechanical Engineers, 29 West 39th Street, New York City.

107. Economic Justification of Diesel-electric Trains. Of all the advances effected in the modern transportation world, there is none that has caught the public interest and captured the imagination of railroad men more than the recent application of the Diesel engine to light-weight, high-speed, streamlined electric trains. The idea of applying the Diesel engine to rail transportation is not new; Dr. Diesel had visualized this possibility years ago. In his book "Rational Theory and Construction of a Heat Motor," published in 1894, he said: "We consider the new motor specially applicable on railways to replace ordinary steam locomotives not only on account of its great economy of fuel but because there is no boiler. In fact, the day may possibly come when it may completely change our present system of steam locomotives on existing lines of rails."

In 1925 the first sale of a four-cycle Diesel-electric locomotive to a railroad in the United States was made; this engine is still being used in switching service. It weighed too much per horsepower and developed insufficient total power to be suitable for high-speed road service. There are an appreciable number of Diesel-electric switch engines being used, but the total number is small when compared with the number of steam-operated switch engines. The cost per switch locomotive hour, exclusive of switchmen, is about $4.20 for steam as compared to $2.00 for the Diesel locomotive using a 450-hp four-cycle engine. The economies obtained and the experiences with these small switching engines resulted in the subsequent use of 1000-hp, four-cycle, Diesel-electric units, making 1000 rpm.

In 1934 a two-cycle light-weight Diesel engine of a new type was installed in America's first Diesel-powered passenger train; it delivered almost twice the power of a four-cycle engine of the same cylinder size. The engine weighed only 20 lb per hp, the reduced dimensions and higher revolutions per minute making it possible to use electrical equipment of lighter weight. A typical Diesel-engine-powered streamlined train is shown in Fig. 88, and an interior view of the locomotive engine room is shown in Fig. 89. The Diesel engine produces the power; a direct-connected generator supplies electric current to electric motors geared to the locomotive axles. This type of motive power, therefore, has the flexibility of the steam locomotive in being self-contained, capitalizes the high efficiency of the Diesel engine, and has the advantageous operating features of the electric locomotive since the method of power application to the axles is the same. Other advantages of Diesel-electric trains are: longer runs without change of locomotives; less time for refueling and water en route; faster acceleration and deceleration of train; easier maintenance; increased availability and greater continuity of service; smaller operating crew; cleaner, safer, and more economical than steam trains.

Because of increased patronage some of the original three-car (1934) Diesel-electric trains, powered with 660-hp engines, have been replaced by six-car (1937) trains which are powered with 1800-hp engines. On the Chicago-Denver ten-car (1937) trains an 1800-hp unit and a 1200-hp unit are coupled together, thus supplying 3000 hp; the distance of 1034 miles is traveled in 16 hours. These trains have demonstrated that they are entirely suitable for the heavier passenger-train service and that they provide the operating characteristics necessary to meet modern competition. To the passenger and the railroad operating man, the Diesel-electric train has opened a new era in transportation—which is providing a clean, comfortable, fast, dependable, and economical service —whereby the railroads are rapidly regaining their share of the country's passenger traffic.

More and more heavy transfer and freight service is also being handled by Diesel-electric units on several railroads. The locomotives use either two 900-hp engines or a single engine of 1800- to 2400-hp (or more) capacity; the large engines, see Figs. 89 and 90, of at least two manufacturers are now in use with multi-bank cylinders.

Although Diesel-electric trains are being used for switching, passenger, and freight service this does not necessarily imply the passing of the steam locomotive. The builders of conventional steam-generating units are alert, and today more than at any previous time are endeavoring to produce complete portable steam power plants that will make longer runs, give greater availability, and cost less to maintain than existing steam locomotives. The designers of a new flash-type high-pressure and high-temperature steam boiler[2] have successfully adapted it for in-

[2] "Steamotive," Bailey, Smith, and Dickey, *Mechanical Engineering*, page 771, December, 1936.

stallation in a locomotive in conjunction with a turbine-electric drive.

The cost of operation [3] of four Diesel-electric trains of the Burlington Railroad for the first million miles was about 31 cents per train mile. This figure includes wages of crews, fuel, maintenance of power plant and train, lubricating oil and water, and miscellaneous supplies and expenses; the figure does not include depreciation and track maintenance. The cost of operation of a conventional steam train of the same capacity at the same speeds is about 70 cents per train mile, or more than twice that of the Diesel-electric trains.

One of the above trains carried an average of 49 passengers per train mile per month at an average revenue for passenger per mile of 1.9 cents. Average daily gross revenues from operations during this month amounted to $467.00 from passenger and $150.00 from mail and express, a total of $617.00 per day. When a steam train was in this service, the passengers per train mile averaged 25 during the preceding month and the cost of operation was about twice that of the Diesel-electric train. The steam train yielded an operation profit for the 30-day period preceding installation of the Diesel-electric train's schedule of $36.00 a day average, or $1080.00 for the month; the Diesel-electric train for this same service showed an operation profit per day of $441.00 or $13,230.00 for the following month. Based on an average monthly profit of less than this amount the $250,000.00 cost of a four-car Diesel-electric train could be amortized easily in about two years.

108. Diesel Power for Industrial Service. Many industrial concerns are striving to reduce operating expenses and manufacturing costs to a minimum; in many cases the cost of power is one of the major expense items. The large plant can buy power, in most cases, at much more favorable rates, and if care is used to keep the peak loads within reason, can often buy it for less than the cost of generating with Diesel-electric units if all fixed charges are included. There are many plants that have requirements of 100 to 500 kw which are buying power under unfavorable conditions or producing it with obsolete steam-driven equipment at a high cost. If electrified, these plants would probably use 3-phase, 60-cycle current at 220 volts; and a Diesel-electric unit could be specified to run at 1200 to 1800 rpm and be equipped with a 75-kw generator. Diesel engines are being used to furnish industrial power to dredges, cotton gins, flour mills, refrigeration plants, etc. Consider a typical installation in a refrigeration plant, where a three-cylinder, 175-hp Diesel engine running at 300 rpm is belted to a 50-kw, 1200-rpm generator and also

[3] "A Million Miles of New Railway Travel," Ralph Budd, *Civil Engineering*, page 231, April, 1936.

separately belted to a 100-hp, 150-rpm compressor (refrigerant). The generator supplies electric power to motor-driven pumps, agitators, air blowers, hoists, lights, and other auxiliary services in and about the plant. This dual drive at a high speed and a low speed makes possible the most effective distribution of the Diesel power to the various plant uses.

The question of maintenance and standby equipment is usually more serious in a small plant than in a large plant. An extra unit in a plant requiring 200-kw capacity would increase the investment about one-third. Routine maintenance could be taken care of by the engine operator, but major overhaul and repairs would require outside help. Inasmuch as engine manufacturers do not (as yet) maintain service stations whereby a replacement unit could be put in the place of a plant unit needing overhaul and the latter unit taken to the service station and reconditioned by experts, it is necessary for each plant to provide its own spare equipment if it desires continuous service.

Consider an industrial plant with an average load of 180-kw and a peak load of 220-kw that operates 2000 hr per year with four 75-kw units installed. Fuel oil would cost about 6 cents per gallon. A unit of this size would probably deliver about 10 kw-hr per gal of fuel oil. The cost of these units, complete with auxiliaries and switchboard, would be about $120.00 per kw installed, and the fixed charges (interest, depreciation, insurance, and taxes) would approximate 12 per cent. Assuming lubricating oil to cost 0.12 cent per kw-hr, maintenance and supplies 0.16 cent per kw-hr, labor 0.17 cent per kw-hr, and fuel oil (as above) 0.60 cent per kw-hr, the total operating cost would be 1.05 cents per kw-hr. For an annual load of 2000 hr × 180 kw = 360,000 kw-hr the total operating cost for Diesel power would be $3780.00 per year.

The annual cost of purchased electric power (under a simple rate) for the above plant in an average American city would probably be as follows. A demand charge of $3.00 per month per kw of maximum demand would amount to $3.00 × 12 × 220 kw = $7920.00 per year. An energy charge of 3 cents per kw-hr would amount to 360,000 kw-hr × $0.03 = $10,800.00 per year. A discount for a 220-kw demand of 50 per cent, to apply to both demand and energy charges, would make the annual net cost for electric power $9360.00.

The comparative studies made, for the use of Diesel power and purchased power for a typical industrial plant, in the preceding paragraphs show a saving in operating cost by Diesel-electric generation of $5580.00 per year, which is a return of 15.5 per cent on the $36,000.00 investment, or 3.5 per cent more than that allowed for fixed charges. Although the figures given apply only to the case assumed, they do show the possibilities of a multiple-engine installation in an industrial plant where the cost of power is 2 cents or more per kw-hr.

109. Diesel Engines for Auxiliary Service. One of the most attractive phases of Diesel engine application is its use as auxiliary power to the main generating units. The Diesel engine may serve as a peak-load carrier, as a generating unit at the end of a transmission system, or as a protection against service interruption in an industrial plant or large office building. Certain characteristics of the Diesel engine, such as absence of standby losses, the ability to start quickly with no preliminary expense, and operation with a minimum of attention, are recognized as favoring its increased use to carry intermittent loads.

For large electrical distributing systems, the cost of current may be decreased by the use of Diesel engines of moderate capacity, that is, from 300 to 1500 kw, installed in suitably located outlying sub-stations and operated regularly over peak-load periods. In this way they serve the double purpose of peak-load equipment and standby or emergency equipment. Installation costs are relatively low, even though the capacity is small, because such installations generally consist of compact units which permit the lowest installed cost per kilowatt capacity. The operating costs are low, because they run at or near full load and have automatic starting and control with minimum attendance charges. For such plants the operating periods vary in different localities, but in general they run from 7 to 9 A.M. and from 4 to 10 P.M.

In recent years, the gradual refinement of the Diesel engine has made available engines that operate satisfactorily under automatic control. Thus Diesel power for auxiliary service can be obtained from one or more Diesel-electric generating units operating in parallel with a storage battery and a battery booster, each of which is started and controlled entirely automatically. These Diesel generating units can be automatically started, controlled, and stopped in sequence in accordance with load requirements; line voltage can be maintained constant regardless of load surges, and all of the supply services to the engine are fully controlled and safeguarded. Now that the automatic Diesel-electric generating unit has proved successful, additional installations will probably be made in increasing numbers in department stores, hotels, office buildings, warehouses, industrial plants, etc., either to replace central-station service or to augment it to provide protection against interruption of service.

Other suitable applications for automatic Diesel power plants include: the development of outlying sections by electric public utilities where the load at present is too light to justify the construction of transmission and distribution lines; the reinforcement of heavily loaded transmission lines; for suburban and country estates where central-station service may be unavailable, costly, or unreliable; for emergency service; for some industries, such as rayon, in which a power outage causes great losses because it interrupts important continuous processes. An automatic Diesel plant could be used which would start instantly when outside power service was interrupted. The laws of some states require that underground mines be provided with auxiliary power in case of emergency. Automatic Diesel plants are also applicable to railroad water-pumping stations, which are often located in sections isolated from public-utility lines. Many sections of the country are now being supplied with cheap natural gas; the same principles and methods now used in automatic Diesel plants can readily be applied to automatic gas power plants. Also in combination with steam power plants, where there is need for low-pressure steam for heating or process work, Diesel engines can be used effectively to improve the heat balance and reduce over-all costs. In such plants the Diesel engines can be arranged to start automatically as soon as the process-steam demand decreases and causes a rise in the steam-engine back pressure.

The economic justification for automatic Diesel power plants is the low cost of fuel combined with low cost of attendance. By reducing the need for human supervision it becomes practical for a few specially qualified experts to give highly competent inspection at regular intervals to the equipment in a group of plants, and also to take care of any derangement signaled to the central office. This assures the plant's being continuously maintained in first-class condition. Such expert supervision and maintenance, if properly organized, would cost much less than the attendance in manually operated Diesel power plants. This applies particularly to plants smaller than 500 kw and plants requiring 24-hr service, as in such manually operated plants the attendance costs per kilowatt-hour would be quite large.

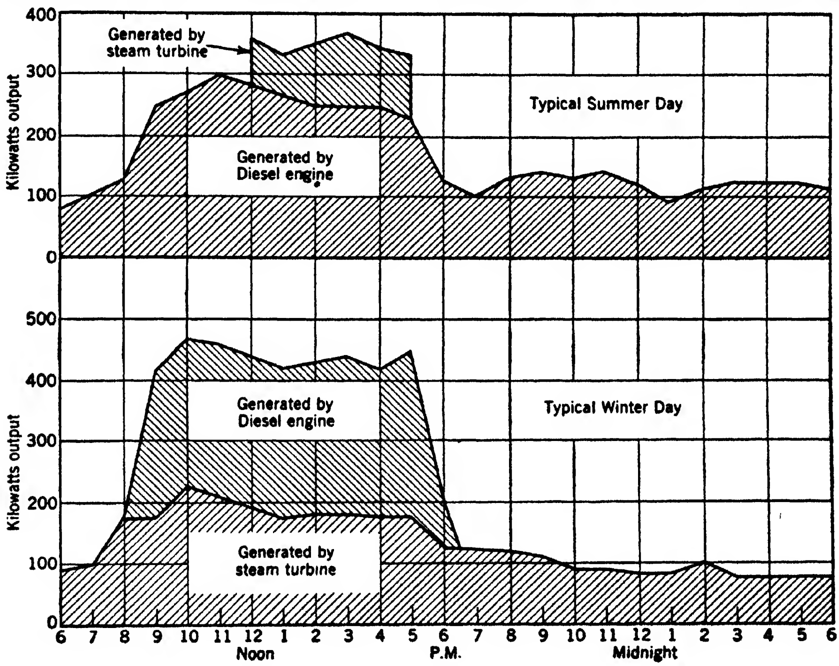
110. Combined Diesel and Steam Power Plants. Diesel engines can be used effectively in conjunction with certain types of steam power plants to produce power and heat at low total cost. Their combined usefulness consists in enabling the steam plant to produce power from steam which is to be used again for heating and process work, whereas the Diesel units generate the remaining power at lower cost than a steam plant would. In such joint steam and Diesel power plants, the total plant capacity need not exceed that of a corresponding all-steam power plant. Despite some loss in the thermal economy, the decreased investment results in lower over-all costs. The Diesel engine as an auxiliary unit should receive consideration when its fuel cost per kilowatt-hour is less than that of a steam power plant whose exhaust heat is being wasted. Whether the addition of Diesel power is a better solution than a major alteration in the steam power plant itself depends upon so many variable and conflicting factors that it can be decided only after careful analysis of each individual installation.

By suitably selecting the steam power equipment for some industrial plants and city buildings, the relative consumption of the steam may be so adjusted that the same steam may be used for producing both heat and power. Such effective utilization of the steam will reduce to a minimum the over-all cost of the heat and power requirements. However, there are many plants where this desirable balance has never been obtained. There are others where the heat and power balance that once existed has been changed; in many industrial and commercial plants the demand for power is increasing more rapidly than that for heat. This also applies to large city buildings where heat demands have changed very little, whereas while power requirements have grown rapidly to provide the ever-increasing human comforts and services demanded by their patrons. In general, these industrial plants and large city buildings have heavy load demands during the day with heat, light, and power peaks, see Fig. 160, and low load through the night, and on holidays. Also the steam load is good during the winter months, almost negligible in the summer; thus steam plants waste heat in the summertime and must always have standby and peak-load capacity—all costing money.

When Diesel power and steam power work together, the steam plant provides power to the extent of steam demand, the Diesel carrying the rest of the load and peaks. The steam plant carries most of the load during the winter, and the Diesel (particularly if waste heat is used for heating service water) most, or all, during the summer months. As shown by Fig. 160, one prime mover is a standby for the other, and Diesel flexibility and economy supplement by-product steam economy; the Diesel engine may be used as a peak-load or a base-load unit, the former being the more popular application. The advantages of the Diesel engine which make it particularly suitable as a standby or peak-load unit include: no operating standby costs, quick starting, great flexibility, low operating costs, and small dimensions per horsepower.

In planning a power plant, too many engineers think only in terms of electrical energy, instead of realizing that electric power may be secondary to steam for heating process work, hot water, laundry, etc. It is in this type of plant that mixed steam and Diesel units have the greatest opportunity. Most engineers believe that power plants should contain but one type of prime mover. Some contend that a plant should contain only one make of apparatus of a given type. One of the principal reasons advanced is that operators capable of handling one kind of equipment cannot handle another nearly so well, and that two types of prime movers cannot be made to operate successfully in synchronism. Many American plants refute this fallacy. Notable examples are the Diesel-steam combinations in the R. H. Macy and Co. department store, Hotel New Yorker, Singer Building, New York University, Doubleday-Doran Co., Dennison Manufacturing Co., Union Stock Yards (Chicago), and the municipal plant in Marshall, Minnesota. Most of these installations were the result of a desire to modernize an old plant with minimum capital investment, using old equipment for standby and peak-load service and the new equipment to generate most of the power.

In any case, there is no cheaper electric power than that obtained from steam used for heating also; it is therefore highly desirable to get as



Courtesy of *Diesel Power.*

FIG. 160. Power distribution for combination Diesel and steam power plant; 440-hp Diesel engine carries base load during summer months and peak load during winter months.

much power from heating steam as possible. It is also desirable to get the remaining power from the most economical source. These two purposes must be accomplished without excessive investment in equipment, as additional fixed charges could easily exceed any saving in operating cost. There are many opportunities for new combination plants; modern developments indicate that the economical plant is frequently Diesel engines combined with a high-pressure non-condensing steam turbine. Pressure and superheat should be selected so that steam at the exhaust end of the turbine has a few degrees of superheat and dry steam is delivered to the heating system.

Several incidental savings are possible, as for example the Diesel engine exhaust gases may be passed through a heat exchanger and used to heat the boiler feedwater. When the Diesel engine is operating at 75 per cent load or higher, it is possible to recover in this manner about 10 to 15 per cent of the heat in the fuel. Circulating water and boiler-feedwater pumping costs are reduced over similar full steam-plant operation. This combination may permit placing the jacket-water coolers in the condenser discharge lines, thereby obviating the pumping of water in the outer circuit of the Diesel-engine cooling system. Another advantage is that the Diesel unit permits quickly placing the steam plant back on the line in a forced shut down. Where natural gas is available at an attractive price, natural-gas engines can be substituted for Diesel engines.

The city of Marshall, Minnesota, has owned and operated a municipal power plant for many years. The original equipment consisted of steam-engine generating units exhausting into the municipal buildings' steam heating system, or exhausting to atmosphere when heat was not required. In 1932, an investigation was made as to possibilities for expansion and modernization; a new steam turbine to be operated condensing during the summer months, a bleeder type turbine for low-pressure steam, and a Diesel-steam plant were given consideration. A Diesel engine was installed to operate in combination with the existing steam equipment, a better heat balance was obtained by eliminating the exhaust steam losses during the summer months, and the city saved[4] about $18,000.00 in operating costs the first year. Providing steam, heat, and water in addition to power, the combined plant has shown an average annual profit of $25,000.00 after deducting all interest and depreciation. In addition free lighting of streets and public buildings, pumping services, and heating of public buildings represent an additional saving.

Table XXVI shows the equipment installed in this plant, operating conditions, and load data; the heating system is of the vacuum type. No low-pressure steam is required during the summer months and only a small amount during April, May, September, and October. The Diesel-generating unit operates alone during the summer, spring, and fall months. During the remaining five months the steam-engine units generate electricity and the exhaust steam enters the heating system at a pressure of 1½ lb per sq in. gage, with the Diesel unit available for standby and peak-load service during this period.

The accompanying table also shows the comparative generation cost for eight-months' operation of the steam plant in 1932 and the cost

[4] "Adding Diesel to Steam Saves $18,000.00 for Marshall, Minn." H. N. Anderson, *Power*, page 635, December. 1933.

TABLE XXVI

SUMMARY OF DATA AND COSTS FOR COMBINED DIESEL-STEAM POWER PLANT

General Data	
Steam engine, 125 lb supply, $1\frac{1}{2}$ lb back pressure, 19 yr old, kw....	125
Steam engine, 125 lb supply, $1\frac{1}{2}$ lb back pressure, 17 yr old, kw....	120
Steam engine, 185 lb supply, $1\frac{1}{2}$ lb back pressure, 5 yr old, kw....	400
Diesel engine, two-cycle, solid injection, 1 yr old, kw..............	605
Total combined Diesel-steam plant capacity, kw...............	1250
Peak load, kw..	660
Average plant load, kw...	230
Normal daytime load, kw..	250 to 400
Normal evening load, kw..	400 to 500
Generating Cost	
Steam Plant, 8 months (Feb. to Sept.), 1932:	
Kilowatt-hours generated.....................................	1,384,100
Generation cost; salaries, fuel, lube oil, repairs, etc.............	$28,627.00
Steam plant operating cost per kw-hr.........................	$0.0207
Diesel-steam Plant, 8 months (Feb. to Sept.), 1933:	
Kilowatt-hours generated.....................................	1,421,900
Fuel cost; coal at $4.96 per ton, fuel oil at 3.8c per gal.........	$9,394.00
Fuel cost per kw-hr for Diesel-steam plant....................	$0.0066
Generating cost; salaries, fuel, lube oil, repairs, etc.............	$16,666.00
Diesel-steam plant operating cost per kw-hr...................	$0.0117

for the operation of the Diesel-steam plant during the same period of the following year. The steam sales were approximately 25,000,000 lb per year; losses were estimated at 20 per cent, since about 31,000,000 lb were sent out annually. The annual electric generation was about 2,000,000 kw-hr and the sales approximated 1,750,000 kw-hr. If a further comparison of the generating costs for the two generating methods were made, based on the 1933 plant output, the Diesel-steam installation would show a probable annual saving of about $18,000.00 over the former all-steam installation.

APPENDIX

A.S.M.E. TEST CODE FOR INTERNAL-COMBUSTION ENGINES[1]

Introduction. Tests for determining the performance of internal-combustion engines by this code apply to all forms of internal-combustion engines, but are limited to the engines alone. Separately driven auxiliaries that are essential to the operation of the engine, such as scavenging pumps or blowers and injection air compressors, must be included in the tests, but the testing of such auxiliaries must follow the other codes appropriate to the auxiliary to be tested and the results in horsepower absorbed included in this code.

Excluded from this code are the following:

(*a*) Complete internal-combustion engine stationary power plants using gaseous or liquid fuels, which have power-generating or power-absorbing auxiliaries separate from the engine.

(*b*) Motor-boat and motorship power plants and the vessels themselves.

(*c*) Aircraft and their power plants.

(*d*) Land propelled vehicles, with or without speed-change gears, including motorcycles, automobiles, motor trucks, tractors and rail locomotives, or self-contained cars.

(*e*) All internal-combustion engine "sets," consisting of an internal-combustion engine so combined with a driven unit as to make it impossible to determine separately the performance of the engine itself, or such as falls within the scope of other codes, excepting only electric generating sets. Among these sets herein excluded are internal-combustion-engine pumps, compressors and hoisting outfits, whether used for hoisting or for hauling.

(*f*) All internal-combustion engines, the construction or operating conditions of which prevent an output determination in terms of either brake horsepower or kilowatts, for a generating set.

Included within this code are the following internal-combustion engines not excluded by the preceding paragraphs:

(*a*) Stationary engines, and electric-generating sets even if not intended for stationary use, burning gaseous or liquid fuels, including the scavenging pumps of two-cycle engines, and injection air compressors, if used and separately driven.

(*b*) Marine engines exclusive of propeller, and burning light or heavy liquid fuels.

(*c*) Aircraft engines limited to block tests, exclusive of propeller, and burning light liquid fuels.

(*d*) Land vehicle engines limited to block tests and burning light and heavy fuels.

The object of the test, and if more than one, then each separately, should be determined and recorded. This code is limited to tests in which the object is of a commercial nature. It excludes all tests forming part of experimental investigations,

[1] Prepared and published by the A.S.M.E. Power Test Codes Committee, American Society of Mechanical Engineers, 29 West 39th Street, New York City. The reproduced material is an abstract of the original publication.

research, or development. Tests of the kind contemplated by this code and for which it has been prepared are those in which the object of the test is *definition or identification* of an engine by its performance characteristics.

When the object of the test is wholly or in part the determination of fulfilment of a contract guarantee, an agreement should be made before the test between the interested parties on all matters that involve any possible elements of uncertainty, or that may later be a cause of dispute; and the points agreed upon should be stated in the report of the test.

Agreements between interested parties should also be made and recorded even when a contract guarantee is not involved. This should be done before any test is undertaken in which an element of judgment is involved in the selection of any instrument, piece of apparatus, physical constant, test method, operating condition or other matter which will promote the accuracy, value, conclusiveness, or general acceptability of the results when obtained. This matter is one of special importance in tests on internal-combustion engines because of their great variety.

Sub-objects of the main object may often be of value in improving the value of results obtained in the test for the main object, and may themselves be made the subject of separate tests, but in all cases *only* when made the subject of previous agreement as to the nature of the sub-objects and the test procedure. Sub-objects of a test must be separately reported as such to distinguish them clearly from the main object, and it is advisable that reasons be given for making such special tests. This is especially important when the special test for the sub-object is to be conducted at the same time as the test for the main object. Sub-objects frequently considered are: exhaust-gas analysis and temperature, amount of air taken into engine, properties and suitability of fuels for engine, solid deposits in combustion chambers, amount of air required to start engine, electrical starting requirements, power consumed by each or all of accessories and auxiliaries, smoke quantity and color or density, etc.

The measurements that must be made in a performance test of an internal-combustion engine will be the following: (*a*) Diameter, stroke, and clearance volume of power cylinders. (*b*) Diameters of piston rods and tail rods of power cylinders of double-acting engines. (*c*) The brake horsepower or shaft horsepower output. (*d*) The kilowatt output, if engine is direct-connected to a generator. (*e*) The speed in revolutions per minute. (*f*) The horsepower to drive the independent cooling water pump and fuel pump (if any). (*g*) The horsepower to drive the independent scavenging pump or blower. (*h*) The horsepower to drive the independent injection air compressor. (*i*) The amount of fuel supplied; cubic feet of gas for gas-burning engines, or pounds of liquid fuel for liquid-fuel engines. If more than one kind at the same time, the amount of each kind. (*j*) The high heating value of the fuel.

The measurements which may be made in addition to those just stated, will be some or all of the following as may be agreed upon in advance by the test engineers. (*a*) Cylinder diameters and strokes of injection air compressor. (*b*) Two-cycle engine scavenging pump diameter and stroke. (*c*) Diameters of piston and tail rods of scavenging pumps of two-cycle engines. (*d*) Diameters of piston rods and tail rods of injection air compressor. (*e*) The indicated horsepower of the engine. (*f*) The number of explosions per minute, or fuel injections per minute, or in general, combustion times per minute, for each and for all working cylinders. (*g*) The spray-air pressure of air-injection oil engines. (*h*) The exhaust back pressure at the common exhaust nozzle. (*i*) The scavenging-air pressure of two-cycle engines, constant or maximum values in header. (*j*) The manifold vacuum of carburetor engines, (*k*) The jacket water or oil-supply pressure at the engine supply; and, if jackets are divided with separate supplies, the pressure at the supply point to each. (*l*) The jacket water or oil back pressure at the main outlet; or, if jackets are divided with separate discharges, the pressure at the outlet of each. (*m*) The pressure, vacuum or the hydraulic head, positive or negative of the supply of liquid fuel at the carburetor connection of carburetor engines, and at the injection-pump suction of injection oil engines. (*n*) The pressure or vacuum of the gas supplied to gas-burning engines at the stop valve. (*o*) The total pressure, static and velocity of cooling-air supply of air-cooled engines. (*p*) Barometric pressure. (*q*) Atmospheric humidity. (*r*) The pressure of lubricating-oil supply to bearings of engines lubricated by pump force-feed circulation. (*s*) The temperature of the atmospheric-air supply. (*t*) The temperature of the fuel for gas at the mixing valve, or for liquid fuel at the carburetor or injection-pump suction. (*u*) The temperature of the jacket water or oil as supplied to the engine as a whole, or if jackets are divided with separate supplies, the temperature of each separate supply. (*v*) The temperature of the jacket water or oil leaving the engine as a whole, or if jackets are divided with separate outlets, the temperature at each outlet. (*w*) The temperature of the scavenging air of two-cycle engines. (*x*) Temperature of injection air in header. (*y*) The temperature of the lubricating oil supplied to the engine at as many points as there are different supplies of pump-circulated lubricating oil, or in the crankcase of engines without pumps or having pump deliveries that do not permit of temperature measurement. (*z*) The temperature of the cooling air supplied to an air-cooled engine. (*aa*) The amount of lubricant consumed. (*ab*) The amount of jacket water or oil supplied, for engines as a whole, and when jackets are divided for each if both supply and discharge are separate. (*ac*) Amount or cubic feet of cooling air supplied to air-cooled engines. (*ad*) Amount of jacket water evaporated in jackets of hopper-cooled engines. (*ae*) Amount, pounds, of injection water supplied internally to cylinders of some oil engines, as a liquid or as steam. (*af*) The compression pressure in cylinders when hot and cold, at normal speed at wide-open air throttle.

The instruments and apparatus required for a performance test of an internal-combustion engine as prescribed by this code will include some or all the following: (*a*) Scales with or without special auxiliary apparatus for weighing liquid fuel, with or without tanks to be used in cooperation. (*b*) Gas meters for measuring gaseous fuel, or gas-metering methods with indirect observation apparatus. (*c*) Gas calorimeters for determining the heating value of gaseous fuels. (*d*) Baumé oil hydrometer for petroleum liquid fuel, and for indirect determination of heating value. (*e*) Pressure gages and mercury or water columns for measuring small pressures and vacua. (*f*) Thermometers. (*g*) Barometers. (*h*) Gas-engine and oil-engine indicators for working cylinders and steam-engine indicators or special low-scale springs for cylinders of two-cycle scavenging pumps and low-pressure cylinders of injection air compressors. (*i*) Pressure-indicating or -recording instruments for compression pressures or injection pressures in cylinders. (*j*) Planimeters. (*k*) Tachometers, revolution counters or other apparatus for measuring speed or number of fuel admissions per minute. (*l*) Absorption dynamometers of appropriate type for the horsepower, speed, and torque of the engine to be tested, especially hydraulic brakes and electric dynamometers. (*m*) Appropriate electrical instruments and apparatus, if the engine is direct-connected to a generator, to provide a suitable electrical load and to measure it.

Indicators cannot be regarded as precision instruments on an internal-combustion engine to the same degree as on a reciprocating steam engine. Limits of use are these. (*a*) When the engine speed exceeds 400 rpm an indicator must not be used for determining the cylinder mean effective pressure of an internal-combustion engine. (*b*) When the engine speed exceeds 400 rpm indicators may be useful as a means of adjustment of valve or spark timing, but have no other value, and are of little use even for such adjustments in the case of multi-cylinder engines. (*c*) On hit-and-miss governed engines the indicator must not be used for determination of the cylinder mean effective pressure. (*d*) When indicator diagrams taken at speeds of 400 rpm or less are not constant in shape and area over long periods of time while a constant load is maintained, the indicator must be rejected as a means of determining cylinder mean effective pressures. When the largest area differs from the smallest area by 4 per cent or more of their arithmetical mean, all diagrams shall be rejected. At intervals, at least ten successive diagrams shall be taken on one card in addition to those taken singly at regular intervals, for comparison as to constancy. (*e*) When indicator diagrams taken at speeds of 400 rpm or less are apparently constant in area, according to *d*, they may still be useless as a means of determining the cylinder mean effective pressure, when the cylinder pressures are high, the spring scale large, and the diagram area is small. Under these conditions the precision will be very low and the indicated horsepower much in error. This condition makes it hopeless to undertake determinations of indicated horsepower for many oil engines of the high-pressure type, developing small mean effective pressures.

Brakes or absorption dynamometers when used must be suitable for the engine to be tested, in capacity, in speed-torque relations, and in ability to maintain an adjustment without necessity for constant hand manipulation to hold the scale beam steady. (*a*) For testing aircraft and automotive engines, nothing but an electric dynamometer shall be used to determine the brake horsepower. (*b*) For brake horsepower tests of marine engines, hydraulic or electric dynamometers shall be used. On engines of this class of large size and lower speed, hydraulic brakes of the type in which brake torque varies with the nth power of the speed, and in which n is equal to or greater than 1.5, shall be used as the preferred form. (*c*) Other forms of brakes may be properly used with stationary engines. (*d*) When brakes of any kind cannot be mounted directly on the engine shaft without other supports, adequate precautions must be taken to avoid errors due to these supports. (*e*) In no case may a brake driven by a belt, or otherwise than by direct coupling to or mounting on the engine shaft, be used for engine brake horsepower measurements. (*f*) Under proper provisions brake horsepower determinations may be made by the indirect or substitution method; in some cases it is the only means available. The engine may be tested with a load such as a propeller (air or water type), a club or a fan, which propeller or fan may later be, or previously has been tested for its speed-torque characteristic curve, and the horsepower of the engine determined by reading from this curve the horsepower for the speed maintained by the engine. Assurances must be obtained of identity of conditions of flow through and resistance of the propeller or the fan under the two separate tests of use and of calibration.

The **heating value** of fuels to be used in calculations of results of tests of internal-combustion engines must be the "high value," products condensed, or the direct reading of a water calorimeter. The following recommendations are made. (*a*) When gaseous fuel is burned in an engine under test, a standard form of gas calorimeter shall be used as one of the instruments of the test and its determination shall be used as the heating value of the gas supplied. Calculations of heating value from the gas analysis are not permissible according to the code, but with all details of apparatus, methods and physical constants involved may be made the subject of previous agreement. (*b*) When liquid fuel is burned in an engine under test, its heating value shall be determined in a standard bomb calorimeter by a recognized physical or chemical laboratory, the selection of which should be a matter of previous agreement, and in the report the "high value" shall be used in calculating final results. In no case shall such a determination be part of the duty of a test engineer, nor should his results be regarded as acceptable. (*c*) The heating value of gasolines and kerosenes may be estimated by the Sherman and Kropf formula, modified by Strong, from the Baumé reading without the bomb test, if such procedure is made the subject of previous agreement.[2]

Gas measurements for engines burning gaseous fuel must be made with meters or by methods yielding the greatest possible accuracy consistent with availability and cost in the large sizes. In no case can the results of proportional meters be regarded as satisfactory, nor those of vane meters of anemometer type. Within the limits of size and cost, positive displacement meters are to be preferred, but these are not available for measurements of large capacity.

Large capacity measurements are best made by means of one of the various types of flow-rate measurements, and that one, applied to mixed gases, should be used that requires the simplest and most accurate determination of physical and chemical properties. When physical or chemical properties of a mixed gas must be determined for use in a metering calculation, even if only gas density direct or computed from the analysis, a recognized chemical laboratory shall be agreed upon to make the determination and its report shall be accepted and used. In no case shall the test engineer make a determination of physical or chemical properties of gases, nor shall his results be acceptable if he does so.

If measurements of volume of *injection air* or of *scavenging air* are desired, the orifice-flow method will be sufficiently correct, if used as prescribed in the section "Instruments and apparatus," but the formula and conditions for such test must be agreed upon in advance between the engineers.

Liquid fuel supplied to an engine should never be measured by volume; it must be weighed directly as used. Combinations of weighing and volume measurements are acceptable if the volume of the tank itself is not the measure but only a reservoir from which the engine draws a supply, starting with a predetermined level marked by a hook gage or on a glass, and a weighed quantity subsequently supplied to restore the original level. In this case the weight is the measure and the result acceptable if proper precautions are used to limit the weight error of reading levels to something less than the error of the scale weighing. In no case shall meters be used to measure liquid fuel supplied to an engine.

When volatile fuels are used, and especially when heavy liquid fuels containing light volatiles must be heated to make them flow, precautions must be taken to prevent losses of weighed fuel by evaporation, and the main supply must be protected against evaporation that may fractionate or concentrate it and change its quality with time.

Preparations. The dimensions and physical conditions of all parts of the engine should be determined and recorded. A clean jacket-water supply should be assured in proper quantity and at as nearly constant a supply temperature as possible. For engines operating with electric ignition, assurance of continued supply at proper

[2] The Sherman and Kropf formulas for the heats of combustion (Btu per pound) of gasoline and kerosene as modified by Strong, U. S. Bureau of Mines, Bull. 43, page 19, are as follows (B = deg. Baumé):

$$\text{Gasoline: high value} = 18{,}320 + 40\,(B - 10)$$
$$\text{Kerosene: high value} = 18{,}440 + 40\,(B - 10)$$

voltage for the whole test must be provided. When liquid fuel is to be used, there should be a sufficiently large supply to last for the whole period of the test; in the viscous oils, which do not mix well without heating, special care should be taken to assure the mixing of new oils supplied to the tanks with that already in the tanks. When the lubricating system includes a tank or sump, this should be drained and new oil provided or new oil in sufficient quantities properly mixed with it to prevent any changes during the test, other than those normal and incident to its operation.

In the absence of any agreement to the contrary, it will be assumed that before starting the test the engine will be put in proper condition so far as can be done by cleaning, by stoppage of all leaks, and by adjustment, including replacement of small parts, such as spark plugs or spray valves. The test engineer should in all cases record not only the condition in which the equipment was found, but also everything done to change the condition. Attention should be given to the following suggestions. (*a*) Jacket surfaces on the heat-receiving side must be free from water scale, rust or other deposits. (*b*) Combustion-chamber walls, including piston heads and all inward-facing parts, such as spark plugs, spray valves, air starting valves, relief valves, as well as inlet and exhaust valves and the parts of four-cycle engines, and the parts of two-cycle engines, should be free from deposits especially carbon, oil, gum or tar, and solids derived from uncleaned air. This also applies to cylinder walls and rings. (*c*) The lubricating-oil system should be free of all deposits of whatever kind, and all flow passages freely and fully open, especially with circulating forced-feed systems. (*d*) The exhaust system, including expansion chambers, mufflers, and pipes, should be clean and free and so arranged as to develop no excessive back pressure. Other engines exhausting into the same system must not interfere with the engine under test. It is always preferable, however, that no other engines should exhaust into the system of the engine under test. (*e*) Fuel-supply and regulating systems must be clean and free of sediment, tar, or other foreign matter, and passages full open, especially for liquid-fuel engines, at the carburetor or injection pumps.

Leakages must be located and corrected. Among important leakages to be investigated attention is called to the following. (*a*) Cylinder leakage outward may be checked by an indicator if the engine is one suitable for indicator use. When there is any clear space between the compression and expansion lines of a diagram taken when there is no combustion and when cooling water is not running, it may be assumed that leakage is excessive. The several possible sources of such leakage must be separately checked. These sources are piston rings, cylinder, cylinder-head gasket, air starting valve, relief valve, spray valve or spark-plug seat, inlet or exhaust valves of four-cycle engines, or air scavenging valves of valve-scavenging two-cycle engines. (*b*) Water leakage into the cylinder may occur from the jackets; compressed air from the starting air system through leaky air starting valves; fuel oil past spray valve; and high-pressure spray air when air spraying is used in oil engines. (*c*) Miscellaneous leakages in the manifold system of carburetor engines anywhere between carburetor and inlet-valve seat, especially at flanges and valve-stem guides; fuel oil in injection oil engines at pump valves, pump plunger, oil-delivery pipe, or spray valve; lubricating oil at pump, tanks, or in piping; spray air at compressor storage bottles, valves, or piping; starting air at compressor storage reservoir, valves, or piping. Insulation of electrical circuits must be checked as a leakage item.

Operating conditions during the test include mainly matters of load and speed to be maintained, with other points incidental to these. The conditions should be agreed upon in all details and should not be inconsistent with the following test procedure: (*a*) Engines of the stationary class, including all engine-driven generator sets, shall be tested at constant speed under governor control or as near constant as the governor will maintain it, and at whatever load may be required by agreement or specific object of the test. (*b*) All engines for the propulsion of land vehicles, except such tractor engines as are fitted with speed governors and operate normally under governor control, shall be tested with throttle wide open, with brake torque regularly varied from nothing to a maximum and back to nothing, at whatever speeds may result, and the horsepower-speed curve typical of the engine shall be determined instead of the horsepower at any given speed. In no case should a *safe* speed be exceeded and, when the highest speed used is the safe limit, it should be so stated. The maximum torque applied shall in all cases be greater than that which will produce maximum horsepower unless the speed required for this is too high to be safe. (*c*) Marine engines and aircraft engines may be tested according to the conditions prescribed for engines of land vehicles, which are the same as the conditions prescribed for stationary engines except that the speed shall be maintained by the brake torque and not by a governor. Marine engines may also be tested at constant speed for full-load operation fixed by their propellers.

Special attention should be given to questions of *safety* that may arise during operation, such as excess speed, or torsional vibrations at a given speed within the normal range, and the procedure to be adopted in such cases should also be included in the agreement.

Starting, Stopping, and Duration. The test cannot be considered as having started until the engine has been in operation for a sufficiently long time to have attained its steady state for the conditions of the first run, and until preliminary observations have been made and recorded to prove that such steady state has been reached. If successive runs are to be made, each under some different conditions, these preliminary observations must be repeated for each run. The engine shall not be regarded as having attained its steady state unless jacket and lubricating-oil temperatures are substantially constant. In no case shall steady state for the first run be regarded as established in a shorter time than 1 hr after the operating conditions have been imposed and before test observations are started. For successive runs at other conditions the minimum time for any engine shall be 10 min, the actual time being determined by the extent to which conditions are changed in successive runs. Maximum time to reach the steady state shall not be taken as greater than 24 hr, which may be regarded as permissible for the first run of the largest sizes of low-speed Diesel engines for motorships and double-acting stationary gas engines.

The duration of the test after the establishment of steady state for the set operating conditions shall be a matter of previous agreement, but should be greater for engines where reliability is a question of importance and which require the longest time to reach a steady state. In no case shall the length of run be less than the period required to reach the steady state, subject to the additional condition that the length of run shall be great enough to insure the accuracy of fuel measurement within 1 per cent, except when gaseous fuel is metered and when metering accuracy is not improved by lengthened runs.

Records. Instruments should be read at least quarter-hourly when the conditions are uniform, and oftener when there is much variation. If there are wide fluctuations in readings, they should be shown by recording instruments. Indicator diagrams, if taken, must be taken from every working cylinder of multi-cylinder engines. If indicator diagrams are taken, the areas, lengths, mean effective pressures, compression pressures, and maximum cylinder pressures should all be recorded in the log. One or more sets of specimen indicator diagrams should be carefully selected for inclusion in the record.

The data and results should be reported in accordance with the form of Table XXVII for tests made at constant speed, and in accordance with Table XXVIII for tests made over the whole speed range of a variable-speed engine. The method of calculation used for obtaining results from observations and physical constants is indicated in the tables for the various items and explained on the following pages.

TABLE XXVII

Data and Results of Internal-combustion Engine Test at Constant Speed, A.S.M.E. Test Code

General Information

1. Date of test__________ 2. Location__________
3. Owner__________ 4. Builder__________
5. Test conducted by__________
6. Object of test__________

Description, Dimensions, Etc.

7. Type of engine, two- or four-cycle, single- or double-acting, horizontal or vertical. If four-cycle, the valve arrangement L, T, or I head; if two-cycle, the type of scavenging; if single-acting, whether crosshead or trunk piston; if multi-cylinder the arrangement of cylinders and cranks, gas or liquid fuel, carburetor type or other class name fixing manner of treating fuel, such as, for example, air injection, mechanical injection, or primary combustion__________
8. Class of service, stationary and special feature, marine, aircraft or vehicle, and what kind__________
9. Auxiliaries attached, such as magneto, fuel-injection pump, fuel-circulating pump, lubricating-oil pumps, jacket circulating pumps, scavenging pumps, spray-air compressor, maneuvering or starting air compressor, radiator fans, oil or fuel coolers or heaters__________
10. Auxiliaries, independent or separately driven, and power__________
11. Details of type, capacity, maker's name, and other features of auxiliaries __________
12. Rated brake horsepower of engine, or kilowatts of electric-generating set, and speed__________
13. Grade of fuel for which engine is designed, kind of gas or specification of liquid fuel, and what was used in test__________
14. Special structural features for fuel utilization__________
15. Special structural features of speed and power control, and governor of reversing gear__________
16. Number of working cylinders__________
17. Bore and stroke of power cylinders__________
18. Diameters of piston rods and tail rods of power cylinders__________
19. Head-end horsepower constant for power cylinders (stroke $\times$ net piston area) $\div$ 33,000__________
20. Crank-end horsepower constant for power cylinders (stroke $\times$ net piston area) $\div$ 33,000__________
21. Capacity of generator or other apparatus consuming power of engine __________
22. Characteristics of generator—d-c or a-c, volts, cycles, phase__________

(Continued on next page)

Test Data and Results

Items	Test 1	Test 2
23. Duration of test, hr		
24. Barometric pressure, lb per sq in		
25. Spray-air pressure (air injection Diesel engines), average gage, lb per sq in		
26. Exhaust back pressure at exhaust nozzle, in. of water		
27. Jacket-water supply pressure at crankshaft center, lb per sq in		
28. Manifold vacuum (carburetor engines), in. of water		
29. Gas pressure at meter (gaseous fuel), in. of water		
30. Lubricating oil pressure circulating forced-feed system, at crankshaft center, lb per sq in		
31. Scavenging-air pressure, average gage, in engine header (two-cycle engines), lb per sq in		
32. Engine-room temperature, F		
33. Temperature of fuel at meter (gaseous fuel), F		
34. Temperature of main air supply entering engine, F		
35. Temperature of mixture at intake port (carburetor engine), F		
36. Temperature of main jacket-water or oil inlet, F		
37. Temperature of main jacket-water or oil outlet, F		
38. Temperature of piston-cooling water or oil at inlet, F		
39. Temperature of piston-cooling water or oil at outlet, F		
40. Heating value of gas, high value at standard pressure and temperature, Btu per cu ft		
41. Heating value of liquid fuel, high value Btu per lb		
42. Total gaseous fuel at meter pressure and temperature, cu ft		
43. Correction factor for gas (absolute meter pressure × absolute standard temperature) ÷ (absolute standard pressure × absolute meter temperature)		
44. Total gaseous fuel at standard pressure (29.92 in. Hg) and standard temperature (60 F), item 42 × item 43, cu ft		
45. Total liquid fuel used during test, lb		
46. Total gaseous fuel per hour at standard pressure and temperature, cu ft		
47. Total liquid fuel per hour, lb		
48. Total heat in fuel supplied per hr, item 46 × item 40, or item 47 × item 41, Btu		
49. Mean indicated pressure, average of all power cylinders, lb per sq in		
50. Maximum pressure, average of all power cylinders, lb per sq in		
51. Revolutions per minute, average for test, rpm		
52. Piston speed (mean) of power pistons, ft per min		
53. Indicated horsepower of all power cylinders, ihp		

(Continued on next page)

Items	Test 1	Test 2
54. Brake horsepower developed by whole engine, by brake or dynamometer measurement, bhp		
55. Brake mean effective pressure, lb per sq in		
56. Horsepower input to motor or output of engine driving independent scavenging pump, hp		
57. Horsepower input to motor or output of engine driving independent injection air compressor, hp		
58. Horsepower input to motors of other independent auxiliaries essential to engine operation (tabulated), hp		
59. Net or actual brake horsepower of engine, item 54 − items 56, 57, and 58, bhp		
60. Fuel consumption per ihp-hr, cu ft or lb		
*61. Fuel consumption per bhp-hr, cu ft or lb		
62. Heat consumed per ihp-hr (high value), Btu		
*63. Heat consumed per bhp-hr (high value), Btu		
64. Indicated thermal efficiency, per cent		
65. Brake thermal efficiency, per cent		
66. Sample indicator diagram from each power cylinder		
67. Average volts each phase, volts		
68. Average amperes each phase, amp		
69. Total electrical output corrected for winding 1, 2, or 3 phase, kva		
70. Power factor, per cent		
71. Total electric output, kw		
72. Separate excitation, kw		
73. Net electric output, kw		
74. Fuel consumed per net kw-hr, cu ft or lb		
75. Heat consumed per net kw-hr, item 48 ÷ item 74, Btu		

* According to contractual conditions, the bhp will be either item 54 or 59.

TABLE XXVIII

Data and Results of Internal-combustion Engine Test at Variable Torque and Speed, A.S.M.E. Test Code

General Information

1. Date of test__________ 2. Location__________
3. Owner__________ 4. Builder__________
5. Test conducted by__________
6. Object of test__________

Description, Dimensions, Etc.

7. Type of engine, two- or four-cycle, single- or double-acting, horizontal or vertical; if four-cycle, the valve arrangement L, T, or I head; if two-cycle, the type of scavenging; if single-acting, whether crosshead or trunk piston; if multi-cylinder the arrangement of cylinders and cranks, gas or liquid fuel; and if radial cylinders whether cylinders are fixed or rotate__________
8. Class of service, aircraft, automobile, truck, tractor, railroad or marine, for gear, electric or direct propeller drive, single direction or reversing__________
9. Auxiliaries, attached and independent, kind, maker's name, capacity, etc. __________
10. Rated horsepower and speed, if any, or speed at maximum torque__________
11. Maximum safe speed__________
12. Bore and stroke of power cylinders__________
13. Diameters of piston and tail rods of power cylinders__________

Test Data and Results

Items	Test 1			Test 2		
	Run 1	Run 2	Run 3	Run 1	Run 2	Run 3
14. Brake horsepower, bhp						
15. Speed, rpm						
16. Total fuel for run, lb						
17. Duration of run, hr						
18. Fuel per hour, lb						
19. Fuel per hour per bhp, lb						
20. Heating value of fuel, high value, Btu per lb						
21. Heat supplied per hour, item 18 × item 20, Btu						
22. Heat supplied per hour per bhp, item 21 ÷ item 14, Btu						
23. Brake thermal efficiency, 2545 ÷ item 22, per cent						
24. Mean effective pressure equivalent to brake horsepower, lb per sq in						
25. Torque at one-foot radius equivalent to brake horsepower, lb-ft						

(Continued on next page)

Items	Test 1			Test 2		
	Run 1	Run 2	Run 3	Run 1	Run 2	Run 3
26. Fig. 1. Curve of rpm plotted horizontal against vertical						
(*a*) Brake horsepower, item 14						
(*b*) Fuel per hour, item 18						
(*c*) Fuel per hour per bhp, item 19						
(*d*) Brake thermal efficiency, item 23						
(*e*) Brake mean effective pressure, item 24						
(*f*) Torque, item 25						

TABLE XXIX

S.A.E. Gasoline Engine Testing Form *

Specification Sheet

Name and Model ______________________ Date of Test __________
Manufacturer ______________________
(1) General Type ______________________ Cycle __________
(2) No. of Cyl. ________ Bore ________ in., Stroke ________ in., Piston Displ. per Cyl. ________ cu in., Total ________ cu. in
(3) Compression Vol. (V_c) ________ cu in., Total Vol. of Cyl. (V) ________ cu. in., Compression Ratio = $\frac{\text{Comp. Vol.}}{\text{Total Vol.}} = \frac{V_c}{V} =$ ________
Compression Pressure ________ lb gage at ________ rpm.
(4) Type of Cyl. Casting ______________________ Matl. __________
(5) Type of Valves ______________________ Location __________
(6) Cooling System ______________________
Fan Diam. ________ in., No. of Blades ________ Projected Width ________ in., Ratio of Fan to Engine Speed __________
(7) Piston, Type ______________________ Matl. __________
Wt. with Rings and Pin ________ lb, Length ________ in., Distance Center of Pin to Top of Piston ________ in.
(8) Piston Rings, No. per Piston ________ Type ______________________ Width ________ in.
(9) Connecting Rod, Type ______________________
Length, c. to c. ________ in., Weight, Upper End ________ lb, Lower End ________ lb, Total ________ lb.
(10) Piston-rod Bearings, Diam. ________ in., Total Length ________ in., Matl. ________ Location __________
(11) Connecting-rod Bearings, Diam. ________ in., Length ________ in., Matl. ________ Type __________
(12) Crankshaft Bearings No. ________ Diam. ______________________
Material ________ Lengths ______________________
(13) Camshaft Bearings, No. ________ Diam. ______________________
Material ________ Lengths ______________________
(14) Type of Cams ______________________ Type of Valve Lifters __________
(15) Inlet Valves, No. per Cyl. ________ o.d. ________ in., Port Diam. ________ in., Lift ________ in., Seat Angle ________ deg

(16) **Exhaust Valves, No. per Cyl.** ________ o.d. ________ in., Port Diam. ________ in., Lift ________ in., Seat Angle ______ deg

(17) **Weight of Valve Reciprocating Parts, Inlet** ____________ lb, Exhaust ____________ lb

(18) **Valve-spring Tension, Inlet Open** __________ lb, Closed __________ lb, Exhaust Open __________ lb, Closed ____________ lb

(19) **Valve Timing, Inlet Valve Opens** ___________ deg __________ Top Center, Closes ___________________ deg after Lower Center

Exhaust Valve Opens _______________ deg before Lower Center Closes ____________ deg ________________________ Top Center

(20) **Flywheel, o.d.** ______________ in., Weight ______________ lb, Moment of Inertia ____________________

(21) **Weight of Engine** ______________ lb, Including ____

CARBURETION

(22) **Carburetor, Name and Model** ____ Nom. Size ______________ in.

(23) **Specifications (Size of Nozzles, etc.)** ____

(24) **How Heated** ____

(25) **General Principles of Operation** ____

____

(26) **Description of Intake Pipe** ____

____

IGNITION

(27) **Name and Type of System** ____

(28) **Type of Distributor** ____ Firing Order ______________

(29) **Type of Breaker** ______________________________ Maximum Spark Advance ________ deg, Retard ________ deg

(30) **Spark Plugs, Name and Type** ______________________________________ Size ______________ in.

(31) **Location** ____ Gap ______________ in.

LUBRICATION SYSTEM

(32) **Type and Description** ____

____

ACCESSORIES

(33) **Accessories Attached during Test** ____

____

* Prepared and published by the S.A.E. Standards Committee, Society of Automotive Engineers, 29 West 39th Street, New York City.

TABLE XXX

S.A.E. Gasoline Engine Testing Form *

Log Sheet

Name________________ Fuel______ Btu per Lb______ Sp Gr______ at______ deg F
Model________ No. Cyl.______ Dynamometer______ Arm (L)______ ft
Bore______ In. Stroke______ In. Displ. (V_d)______ Cu In. Humidity______ per cent
Laboratory______ Date______ Oil______ Grade______ Cold Test______ deg F
Observers______ Saybolt Univ. Vis. at deg F______ At deg F______

BRAKE HP AND FUEL CONSUMPTION

Run Number	Symbol	Formula	1	2	3	4	5	6	7	8	9	10
Time Started		†										
Duration of Run—Min	t	†										
Counter Start	C_o	†										
Counter Finish	C_t	†										
Total Rev	r	$C_t - C_o$										
Average Rpm	N	$\frac{r}{t}$										
Barometer In. Mercury		†										
Room Temp		†										
Correction Factor	CF	$\frac{p_s}{p_o} \times \sqrt{\frac{T_o}{T_s}}$										
Brake Load at Arm L	W	†										
Brake Load Corrected	W_c	$CF \times W$										
Torque Lb-Ft	T_q	WL										
Brake Mep	bmep	$\frac{150.8 T_q}{D}$										
Brake Hp	bhp	$\frac{WLN}{5252.1}$										
Friction Hp at N	fhp	fhp Curve										
Indicated Hp	ihp	bhp+fhp										

BRAKE HP AND FUEL CONSUMPTION	Mechanical Efficiency	e_m	$\frac{\text{bhp}}{\text{ihp}}$										
	Temp Jacket Water—In		†										
	Temp Jacket Water—Out		†										
	Temp Oil—In		†										
	Temp Oil—Out		†										
	Oil Press, Lb		†										
	Temp Air to Carburetor		†										
	Wt Fuel Start	W_o	†										
	Wt Fuel Finish	W_t	†										
	Lb Fuel Used	W_f	$W_o - W_t$										
	Lb Fuel per Bhp-hr	F	$\frac{60 W_f}{t \times \text{Bhp}}$										
	Thermal Eff. Re Bhp	e_t	$\frac{2545}{F \times \text{Btu}}$										
FRICTION HP	Time Started		†										
	Duration of Run—Min	t	†										
	Counter Start	C_o	†										
	Counter Finish	C_t	†										
	Average Rpm	n	$\frac{C_t - C_o}{t}$										
	Brake Load at Arm L	w	†										
	Friction Hp	fhp	$\frac{wLn}{5252.1}$										
	Mean Temp Jacket Water		†										

Above computed data corrected for barometer Yes/No and temperature Yes./No.

* Prepared and published by the S.A.E. Standards Committee.

† Laboratory readings

TABLE XXXI

S.A.E. Diesel Engine Testing Form *

Specification Sheet

Name and Model ________ Serial No. ________ Date of Test ________

Type: Automotive ________, Marine ________, Stationary ________, Crankshaft Balanced ________, Static ________, Dynamic ________

(1) General Type (Combustion and fuel-injection system) ________, Cycle ________, Governor Used ________, Type ________

(2) No. of Cyl. ________ Bore ________ in., Stroke ________ in., Piston Displ. per Cyl. ________ cu in., Total ________ cu in.

(3) Compression Vol. (V_c) ________ cu in., Total Vol. of Cyl. (V) ________ cu in., Compression Ratio = $\frac{\text{Comp. Vol.}}{\text{Total Vol.}} = \frac{V_c}{V} =$ ________

Compression Pressure ________ lb gage at ________ rpm with engine at normal full load operating temperature.

(4) Type of Cyl. Casting (single or en bloc) ________, Material ________, Liners ________, Material ________

(5) Type of Valves ________ Location ________

(6) Cooling System ________ Capacity ________ gal

Fan Diam ________ in., No. of Blades ________, Projected Width ________ in., Ratio of Fan to Engine Speed ________

(7) Piston, Type ________ Matl. ________

Wt. with Rings and Pin ________ lb, Length ________ in., Distance Center of Pin to Top of Piston ________ in.

(8) Piston Rings (a) Compression No. per Piston ________ Type ________ Width ________ in.

(b) No. per Piston ________ Type ________ Width ________ in.

(9) Connecting Rod. Type ________ Length, c. to c. ________ in., Weight, Upper End ________ lb, Lower End ________ lb, Total ________ lb

(10) Piston-pin Bearings, Diam. ________ in., Total Length ________ in., Material ________ Location ________

(11) Crank-pin Bearings, Diam. ________ in., Length ________ in., Material ________ Type ________

(12) Crankshaft Bearings, No. ________ Diam. ________ in., Material ________ Lengths ________ in.

(13) Camshaft Bearings, No. ________ Diam. ________ in., Material ________ Lengths ________ in.

(14) Type of Cams ________, Driven by ________, Type of Valve Lifters ________

(15) Inlet Valves,† No. per Cyl. ________ o.d. ________ in., Port Diam. ________ in., Lift ________ in., Seat Angle ________ deg

(16) Exhaust Valves,† No. per Cyl. ________ o.d. ________ in., Port Diam. ________ in., Lift ________ in., Seat Angle ________ deg

(17) Weight of Valve and Reciprocating Parts, Inlet ________ lb, Exhaust ________ lb.

(18. Valve-spring Tension, Inlet Open ________ lb, Closed ________ lb, Exhaust Open ________ lb, Closed ________ lb.

(19) Valve Timing, Inlet Valve Opens ________ deg ________ Top Center, Closes ________ deg after Lower Center

Exhaust Valve Opens ________ deg before Lower Center, Closes ________ deg ________ Top Center

(20) Flywheel, o.d. ________ in., Weight ________ lb, Moment of Inertia ________, No. ________

(21) Weight of Engine ________ lb. Including ________

FUEL-INJECTION SYSTEM

(22) Type of Fuel System ________

(23) Name and Model of Fuel Pump ________
Bore ________ in., Stroke ________ in., No. of plungers in Fuel Pump ________

(24) Normal Pressure in Fuel-supply System ________ lb. Injection pressure ________ lb

(25) Type of Injection Nozzle ________
Spray Orifices, Number ________, Size ________, Angle of Spray ________

(26) Location of Injection Valves, No. ________, in Cyl-Hd. ________, at Center ________, at Periphery ________
In Cyl. Wall ________, Vertical ________, Horizontal ________, Angular ________

(27) Type of Injection Valve, Open ________, Closed ________, How operated, Cam ________, Pump pressure ________
Fuel injection (engine at rest), full load begins ________ ends ________
Is injection-period timing fixed or variable? ________, Beginning ________, Ending ________

AIR SYSTEM

(28) Type of Air Injection Pump ________ Displacement ________
No. of Stages ________, Pressure of Injection Air ________ lb, Pump Speed, in Terms of Crankshaft Speed ________
Diameters, 1st stage ________ in. 2nd stage ________ in. 3rd stage ________ in. Stroke ________ in.

(29) Pressure of Scavenging Air ________ Supercharger, type ________ pressure ________ lb, temperature ________ deg
Type of Scavenging Air pump ________
Bore ________ in., Stroke ________ in., Number of Cylinders ________
Scavenging Air Pump, How Driven ________ Speed in Terms of Crankshaft Speed ________

LUBRICATION SYSTEM

(30) Type and Description, Splash ________, Wet Sump ________, Dry Sump ________, Force Feed ________, Outside Tank ________
If cooled, describe method ________, If filtered, describe method ________

METHOD OF STARTING

(31) Electric ________ Inertia ________ Air ________ Auxiliary Engine ________ Hand ________
Glow plug ________ Is incoming air preheated? ________ Method ________

ACCESSORIES

(32) Accessories, standard with the engine and used during test, air filter, type ________ fuel filter, type ________
Water circulating pump, type ________
Lubricating oil filter, type ________ Starting air compressor ________
Generator ________

* Prepared and published by the S.A.E. Standards Committee.
† If non-poppet type, describe and give dimensions from which areas can be computed.

TABLE XXXII

S.A.E. Diesel Engine Testing Form*

Log Sheet

Name________ No. Cyl________
Model________ Serial No.________
Bore________ In. Stroke ________ In. Displ. (V_d)________ Cu In.
Laboratory________ Date________
Observers________

Fuel________ Btu per Lb________ Sp Gr________ at________ deg F
Dynamometer________ Arm (L)________ ft
Humidity________ per cent
Lube Oil________ Grade________ Cold Test________ deg F
Saybolt Univ. Vis. at ____ deg F________ At ____ deg F________

Run Number	Symbol	Formula	1	2	3	4	5	6	7	8	9	10
Time Started		†										
Duration of Run—Min	t	†										
Counter Start	C_o	†										
Counter Finish	C_t	†										
Total Rev	r	$C_t - C_o$										
Average Rpm	N	$\frac{r}{t}$										
Barometer In. Mercury		†										
Room Temp		†										
Correction Factor	CF	$\frac{p_s}{p_o} \times \sqrt{\frac{T_o}{T_s}}$										
Brake Load at Arm L	W	†										
Brake Load Corrected	W_c	$CF \times W$										
Torque Lb-Ft	T_q	WL										
Brake Mep	bmep	$\frac{150.8 T_q}{D}$										
Brake Hp	bhp	$\frac{WLN}{5252.1}$										
Friction Hp at N	fhp	fhp Curve										
Indicated Hp	ihp	bhp+fhp										

BRAKE HP AND FUEL CONSUMPTION	Mechanical Efficiency	e_m	$\frac{\text{bhp}}{\text{ihp}}$										
	Water, Gal per Min		†										
	Temp Jacket Water—In		†										
	Temp Jacket Water—Out		†										
	Temp Oil—In		†										
	Temp Oil—Out		†										
	Temp Exhaust		†										
	Oil Press, Lb		†										
	Wt Fuel Start	W_o	†										
	Wt Fuel Finish	W_t	†										
	Lb Fuel Used	W_f	$W_o - W_t$										
	Lb Fuel per Bhp-hr	F	$\frac{60 W_f}{t \times \text{Bhp}}$										
	Thermal Eff Re Bhp	e_t	$\frac{2545}{F \times \text{Btu}}$										
FRICTION HP	Time Started		†										
	Duration of Run—Min	t	†										
	Counter Start	C_o	†										
	Counter Finish	C_t	†										
	Average Rpm	n	$\frac{C_t - C_o}{t}$										
	Brake Load at Arm L	w	†										
	Friction Hp	fhp	$\frac{wLn}{5252.1}$										
	Mean Temp Jacket Water		†										

Above computed data corrected for barometer $\genfrac{}{}{0pt}{}{\text{Yes}}{\text{No}}$ and temperature $\genfrac{}{}{0pt}{}{\text{Yes.}}{\text{No.}}$

* Prepared and published by the S.A.E. Standards Committee.

† Laboratory readings

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