RESTRICTED

FUNDAMENTALS OF DIESEL ENGINES U. S. NAVY

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FUNDAMENTALS OF DIESEL ENGINES-U.S. NAVY

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CHAPTER 1

INTRODUCTION

1-1. Purpose. The main purpose of this book is to serve as a text in diesel schools preparing diesel engineering officers and diesel engine service men for the United States Navy. It should be noted that the educational background of the men going through the various diesel schools differs; different numbers of hours are assigned to classroom work; and the schools themselves have different engine equipment around which their courses are built. Therefore, the book had to be written in such a way as to be readily understandable to all of the trainees and to contain all information required by all schools. In cases where the time allowed for the study of this material is not sufficient to cover the whole book, it will be necessary to leave out certain chapters or sections. It is hoped that those men who are really interested in diesel engines and in raising their own standing, will later find time to study those chapters and sections which were omitted during their school training.

1-2. Definition of a diesel engine. The internal-combustion engine is an engine from which work is obtained by the burning or combustion of fuel within the engine cylinders themselves. A diesel engine is an internal-combustion engine which uses fuel oil, injected in a finely divided state into the cylinder which contains air compressed to a comparatively high pressure and temperature. The temperature of the air must be so high as to ignite the particles of the injected fuel. No other means are used for ignition. Due to the method of ignition used, diesel engines are often called *compression-ignition* engines. This differentiates them from other internal-combustion engines called *spark-ignition* engines. These latter engines use gasoline as fuel and the mixture of gasoline and air is ignited by an electric spark.

1-3. Importance of internal-combustion engines. The importance of the internal-combustion engine, in general, can readily be seen from the fact that it made possible the automobile, airplane, and submarine. The Navy uses comparatively few gasoline engines except in airplanes and in some power boats. However, the use of diesel engines in naval vessels is very large. The total horsepower of diesel engines installed in the Navy for various applications is over 12,000,000 bhp. and is continuously growing.* The tremendous growth of diesel engine application can be seen from the following figures: from August, 1941, to June, 1943, the number of diesel-propelled seagoing vessels has been multiplied by 13. In August, 1941, one-third of the diesel-propelled ships were submarines; in June 1943 one-tenth of the diesel-propelled ships were submarines. Thus, while the number of submarines has grown four times, the number of surface vessels has grown 17.5 times.**

1-4. Usage of diesel engines. In the Navy, diesel engines were used successfully first in submarines, then in small boats. Lately, the application of these engines has been widely extended. Diesel engines are being used to drive tugs with from 400 to 3,000 hp., depending upon the size and type; various types of landing craft, varying from 175 to 2,000 hp.; submarine chasers with from 800 to 1,800 hp.; patrol craft and minesweepers with 3,000 hp.; seaplane tenders with 6,000 hp.; submarines with 6,400 hp.; escort vessels with from 6,000 to 12,000 hp., and auxiliary ships and tenders up to 12,000 hp.* In addition, diesel engines are used extensively as an auxiliary power supply or as stand-by power in battleships, cruisers, and aircraft carriers.

1-5. Advantages of diesel engines. The main advantages are: (a) high power per pound of engine-installation weight, particularly with present-day high-speed engines; (b) high reliability in operation;

*Data taken from Rear-Admiral E. W. Mills' paper: "Some Aspects of Diesel Engines for Navy Main Propulsion," Mechanical Engineering, Sept. 1943, p. 625.
**Data taken from Captain T. G. Reamy's paper: "Diesel Engine Maintenance in the Navy," Mechanical Engineering, Sept. 1943, p. 628.

(c) low fuel consumption per hp.-hr., which means an increased cruising radius for seagoing vessels; (d) reduced fire-hazard as compared with gasoline engines.

1–6. Trends in development. The first diesel engines were low-speed, low-pressure heavy engines. The first steps of development were: (a) to increase the power for a given bore and stroke by raising the operating speed, thus getting more power strokes per minute, and (b) to raise the gas pressure in the cylinders by improving the combustion, by obtaining a better utilization of the air inside of the cylinder. The next step was to reduce the weight of the engines by a more careful use of materials; avoiding unnecessary weight where possible; using materials of higher strength for a given weight, both in stationary and moving parts, such as using high-grade alloys instead of cast-iron for exhaust valves, aluminum alloys instead of cast-iron for pistons, high-grade alloy steels for carbon steel in connecting rods and crankshafts, welded steel instead of cast-iron for framework, nickel cast-iron for cylinder liners, etc. Particular attention had to be paid to lightening the reciprocating parts, in order to reduce the undesirable forces of inertia as the engine speeds were gradually being increased. Another step was changing the engine shape to get more power for the same overall bulk. This was done by shortening the engines by using a V-type cylinder arrangement or crowding the cylinders together by using an X-type arrangement with a vertical shaft and several banks of cylinder rows as seen in the pancake engine.

The latest step is supercharging, i.e., increasing the amount of air taken in which permits an increase in the amount of fuel burnt in the engine, thus raising the useful pressure and the horsepower developed. While present diesel engines are tremendously improved as compared with diesel engines built 20 and even 10 years ago, further progress undoubtedly will take place. The probable procedure will be a further increase in the engine speed and the obtaining of more efficient combustion. However, improvements become more and more difficult as the present engines have almost reached the safe limits of high temperatures and stresses in various parts.

1–7. Importance of fundamentals. A thorough understanding of, and familiarity with, the fundamentals of internal-combustion engines as presented in this book is important for several reasons. First, it is essential for an understanding of the operation of various engines and the functions of the different parts. It is much easier for a person to do something properly when he understands why he has to do it rather than to do it simply because he has been told to. The understanding of the fundamentals underlying the operation of different engine parts helps to prevent undesirable operating conditions and thus reduces the maintenance troubles. Second, it is impossible to cover every possible problem in a short school term. A proper understanding of the fundamentals helps to solve new problems of operation and maintenance, and suggests how to meet new conditions of operation. Furthermore, a good understanding of the fundamentals will help materially in dealing with an engine of a new type or a new design, because the basic principles for all engines are the same. The different shape of a certain part will not confuse a person who understands the purpose and operating conditions of a certain engine part or a piece of auxiliary equipment. In a case when a diesel operator has to use a helper who did not receive diesel-engine training, or has to teach a man to take his place in an emergency, a thorough understanding of the fundamentals will be of invaluable help. When asked by the green man why he should do such or such a thing, he will be able to answer the questions of the inexperienced man and this will increase his usefulness and will raise the professional pride of the unexpected teacher, making him a leader instead of a straw boss. At the same time it is well to remember that studying from a book is not enough, regardless of how good the book may be. It takes several years of practical experience in operating and servicing diesel engines of various types to become a real diesel man.

1–8. Questions. 1. What is the definition of a diesel engine?

2. What Naval vessels have the largest diesel-engine installations?

3. Enumerate the main advantages of a diesel engine.

4. What is the probable trend of further development of diesel engines?

5. Enumerate briefly the importance of knowing the fundamentals of diesel engines for an engineering officer and a diesel-engine operator or service man.

CHAPTER 2

BASIC PRINCIPLES

2-1. Units of measurement. All physical quantities can be expressed by means of three units—length, force, and time. In the English system, used in the United States, the standard units are the foot (ft.), pound (lb.), and second (sec.).

Derived units are units derived from the three standard English units and are used to measure quantities encountered in engineering practice.

2-2. Derived units. *Area* is a measure of surface and is expressed as the product of the length and width, or of two characteristic lengths of the surface. Areas are expressed in square units, such as square feet (sq. ft.) or (sq. in.).

Volume is a measure of space and is expressed as the product of area and length, or of three characteristic lengths of the space. Volumes are measured in cubic units, such as cubic feet (cu. ft.) or cubic inches (cu. in.).

Linear motion is the length of the line along which a point or a body has moved from one position to another. Linear distance is measured in units of length, such as feet or inches.

Rotary motion is the movement of a point or body along a circular path. The position of a point or body rotating about a fixed point in a plane may be expressed by the angle through which it has rotated. Thus in Fig. 2-1, when point a has moved to b, remaining all the time at a constant length r from the fixed point o, the position of b is determined by the angle of rotation c. Angles are measured in degrees, 360° corresponding to one complete revolution.

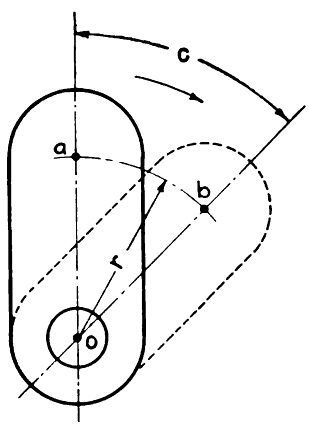


Figure 2-1. Rotary motion.

Velocity is the distance traveled by a moving point in a unit of time, as sec., min., or hour. Velocity is computed by dividing the distance traveled by the time used for the travel,

$$\text{Velocity} = \text{distance} \div \text{time} \qquad (2\text{-}1)$$

When the distance is expressed in feet and the time in minutes, the velocity will be expressed in feet per minute (ft. per min.) or ft./min. If the time is expressed in seconds, the velocity will be expressed in feet per second, (ft. per sec.) or ft./sec.

> *Example 2-1.* A point traveled 1,800 ft. in 2.5 min. Find its velocity.
> Velocity = distance ÷ time = 1,800 ÷ 2.5 = 720 ft./min.

Velocity may be *uniform* or *varying*. If the motion is uniform, i.e., when the velocity is constant, the above expression will give the *actual velocity*. If the motion, and hence also the velocity, is not uniform, as in the reciprocating motion of a piston in an engine cylinder, then the above expression will give the *average velocity*. The average piston velocity is referred to as piston *speed*. The velocity of a moving vehicle or aircraft is generally called *speed* and is expressed in miles per hour (mph.).

> *Example 2-2.* Find' the velocity in ft./min. of an automobile traveling at a speed of 50 mph.
> Since 1 mile = 5,280 ft. and 1 hour = 60 min., the speed of 50 mph. = 50×5,280 ÷ 60 = 4,400 ft./min.

When referring to the flow of fluids such as gas or water, the rate of flow is called *velocity* and is expressed in terms of feet per minute (ft./min.) or feet per second (ft./sec.). On the other hand, the term *speed* is applied when referring to the rotary motion of a mechanism. Thus, *engine speed* is said to be so many revolutions of its crankshaft per minute, and is designated as rpm.

Acceleration is the change of velocity of a moving body in a unit of time. Acceleration may be uniform or varying. It is considered positive when the velocity increases, and negative when the velocity decreases. Negative acceleration is called *deceleration.*

Acceleration is computed by dividing the change in velocity by the time during which this change takes place. If the acceleration is uniform, then this expression will give the *actual acceleration.* If the change in velocity is not uniform, then this expression will give the *average acceleration.* When the velocity is expressed in ft./min., acceleration will be expressed in ft./min. per minute or ft./min.^{2} If the velocity is expressed in ft./sec., the acceleration will be expressed in ft./sec. per second or ft./sec.^{2}

An example of constant acceleration is the acceleration of the earth's force of gravity, which for all technical calculations can be taken as 32.2 ft./sec^2. If a body is released above the surface of the earth, it will begin to fall, being attracted to the center of the earth. If the density of a body is high, such as that of a metal part, the resistance presented by the air is very small and generally may be disregarded. The acceleration of gravity means, that during each second, the velocity of a falling body will increase by 32.2 ft./sec. If the resistance of the air is large, as in the case of a falling body supported by a parachute, the acceleration will be reduced.

Example 2-3. A body at rest is acted upon by a force which after 4 secs. has caused it to acquire a velocity of 6,000 ft./min. What is the average acceleration in ft./sec.^{2}?

6,000 ft./min. = 6,000 ÷ 60 = 100 ft./sec.

$$\text{Av. Accel.} = \frac{\text{change in velocity}}{\text{change in time}} = \frac{100-0}{4} = 25 \text{ ft./sec.}^2$$

Pressure may be defined as a force acting on a unit of area. Pressure may be exerted either by a solid body or by a fluid.

Example 2-4. Determine the pressure on the subbase of an engine; the weight of the engine is 1,800 lbs.; the contact area between the engine and the subbase consists of two strips, each 2 in. wide and 40 in. long.

In this case the weight of the engine is a force pressing the engine against the subbase and the pressure p will be equal to

$$p = \frac{1{,}800}{2(2\text{x}40)} = 11.25 \text{ psi.}$$

In general this can be expressed as

Pressure = force ÷ area

or

$$p = F \div A \tag{2-2}$$

Where the force may be in pounds and the area upon which the force is acting may be expressed in square inches or square feet. Accordingly, pressure may be in pounds per square foot (psf.) or pounds per square inch (psi.).

In the case of contact between two solid bodies, the surfaces have a perfect uniform contact only in exceptional cases. The presence of uneven areas will give higher pressures at the high spots, and lower, if any, at the places of depression. In such a case, the pressure determined as in the above example will give only the average value. However, when a force is transmitted by a fluid, either liquid or gas, the pressure between the fluid and the walls of the container will be uniform and equal in all directions, regardless of the shape of the walls.

Example 2-5. Determine the pressure in an air compressor if the force acting upon the piston is 750 lbs. and the piston diameter is 3 in.

The area of a 3-in. circle

$A = \pi r^2 = \pi (d \div 2)^2 = (\pi d^2) \div 4 = 3.14 \times 3^2 \div 4 = 7.07$ sq. in. and the pressure will be according to equation (2-2)

$$p = 750 \div 7.07 = 106 \text{ psi.}$$

Specific gravity. The ratio of the weight of a certain volume of a liquid to the weight of an equal volume of water is called *specific gravity.* For practical use, it is well to remember that 1,728 cu. in. = 1 cu. ft., 1 cu. ft. of fresh water weighs 62.4 lbs. and since 1 gal. = 231 cu. in., 1 gal. therefore weighs

$$62.4 \times 231 \div 1{,}728 = 8.34 \text{ lbs./gal.}$$

Example 2-6. Determine the weight of 1 gal. of fuel oil which has a specific gravity of 0.84.

The weight of the oil will be equal to the weight of the water times the specific gravity, or $8.34 \times 0.84 = 7$ lbs./gal.

2-3. Work and power. Work is done when a force is moving a body through a certain distance. Work is measured by the product of the force F multiplied by the distance d moved in the direction of the force,

$$\text{Work} = \text{force} \times \text{distance}$$
$$W = F \times d \tag{2-3}$$

Work is expressed in ft.-lbs. or in in.-lbs.

Example 2–7. Find the work necessary to raise the weight of 100 lbs. a distance of 2¾ ft.

The work to be done is $100 \times 2.75 = 275$ ft./lbs.

Power is the rate at which work is performed, or the number of units of work performed in one unit of time. Power is measured in ft.-lbs. per min. In engineering calculations, 550 ft.-lbs. per sec. = 33,000 ft.-lbs. per min. and is called a *horsepower.*

$$\textit{Power} = \textit{Work} \div \textit{Time}$$

Example 2–8. Determine the power required to do the work of example 2–7 if the work is to be performed: (a) in 5 sec. or (b) in 25 sec.

For case (a): The rate of performing the work is 275 ft.-lbs. ÷ 5 sec. = 55 ft.-lbs. per sec.

Expressed in horsepower this is

$$\frac{55 \text{ ft.-lbs. per sec.}}{550 \text{ ft.-lbs. per sec.}} = 0.1 \text{ hp.}$$

Or, since 55 ft.-lbs./sec. × 60 sec./min. = 3,300 ft.-lbs./min.

then $\dfrac{3{,}300 \text{ ft.-lbs./min.}}{33{,}300 \text{ ft.-lbs./min.}} = 0.1$ hp. as before.

For case (b): The rate is 275 ft.-lbs. ÷ 25 sec. = 11 ft.-lbs. per sec., so

$$\frac{11 \text{ ft.-lbs./sec.}}{550 \text{ ft.-lbs./sec.}} = 0.02 \text{ hp.}$$

Electric Power is measured in units called watts; 1,000 watts are called 1 kilowatt (kw.). The conversion factors between hp. and kw. are 1 hp. = 0.746 kw. or 1 kw. = 1.341 hp.

2–4. Temperature. The temperature of a body is a characteristic which can be determined only by comparison with another body. When two bodies are placed in close contact, the one which is hotter will begin to heat the other and is said to have a higher temperature.

The scales of temperature are set arbitrarily. The temperature scale used in this country is the Fahrenheit scale. In this scale the two reference points are the temperature of melting ice, designated as 32° F, and the temperature of steam with the water boiling under normal barometric pressure, designated 212° F. The distance on the scale between these two points is divided into 180 equal parts called *degrees*. The scale is continued in both directions, above 212° and below 32°. Below 0° F the temperatures are designated by a minus (−) sign.

In theoretical calculations pertaining to gases, another scale is used, called the *absolute* or *Rankine* scale. In this scale the unit, the degree, is the same as in the Fahrenheit scale, but the absolute zero is placed at −460° F. Thus the relation between the absolute temperature, designated T, and the corresponding Fahrenheit temperature t is

Degree Rankine = degree Fahrenheit + 460

or $$T = t + 460 \tag{2–5}$$

In technical calculations pertaining to gases, 60° F is called *normal* or *standard temperature.*

2–5. Properties of gases. Pressure has already been discussed in Sec. 2–2. Additionally it should be mentioned that, the pressure of a gas is often expressed by the height of the column of a liquid which will balance the gas pressure in the space under consideration. The liquid used is either water or mercury. The relation between the various units can be established noticing that 1 cu. ft. of fresh water at room temperature weighs 62.4 lbs., or a column of water 1 ft. high and acting upon an area of 1 sq. ft. is equal to a pressure of 62.4 psf. Since 1 cu. ft. contains 1,728 cu. in., the weight of 1 cu. in. of water is 62.4 ÷ 1,728 = 0.0361 lb. Therefore, a column of water 1 in. high, acting upon 1 sq. in. will produce a pressure of 0.0361 psi. To obtain a pressure of 1 psi., the column must be higher in the proportion of 1 ÷ 0.0361 = 27.70 or 27.70 in., or also 27.70 ÷ 12 = 2.309 ft. Mercury is 13.6 times as heavy as water, and therefore a column of mercury must be shorter in this proportion or 1 psi. = 27.70 ÷ 13.6 = 2.036 in. mercury, and conversely 1 in. mercury = 1 ÷ 2.036 = 0.491 psi.

Gauge and absolute pressures. Instruments measure the pressure of gases in respect to the pressure of atmospheric air, also called barometric pressure. Pressures measured thus, are called *gauge* pressures and indicate pounds per square inch gauge (psig.) and pounds per square foot gauge (psfg.). The actual pressure exerted on the gas can be obtained by adding to the gauge pressure the barometric pressure. This pressure is called *absolute* pressure, and is indicated as pounds per square inch absolute and pounds per square foot absolute. If absolute pressure is designated p_a, gauge pressure p_g, and barometric pressure b, then the relation can be written as

abs. pressure = gauge pressure + barometric pressure

or $$p_a = p_g + b \tag{2–6}$$

The barometric pressure b is not constant, since it changes with the altitude and weather conditions. *Normal* or *standard barometric* pressure at sea level is taken as 29.92 in. of mercury or 29.92 ÷ 2.036 = 14.70 psia.

Volume is the space occupied by a body, either solid, liquid, or gas. If the body is a vapor or gas, its volume must be confined from all sides. In engines the volume of gas is usually confined by a *cylinder* having one end closed by a stationary cylinder head, and the other end closed by a movable head called a *piston*. The *piston* has provisions for a gastight seal. When the piston changes its position, the volume of the gas changes. When the piston approaches the cylinder head, the volume is being decreased, the gas is *compressed;* when the piston moves away from the cylinder head, the volume increases, the gas *expands*.

Gas characteristics. In dealing with gases, the three measurable quantities, pressure, volume, and temperature, are called gas properties or characteristics. The three characteristics are connected by a simple relation, which for any gas can be written as

$$pV = WRT \qquad (2\text{–}7)$$

where p is the absolute pressure in pounds per square foot absolute, V is the volume in cu. ft., W is the weight of the gas in lbs., T is the absolute temperature in degrees Rankine, and R is a constant called the gas constant. The numerical value of R is known for all gases. It is expressed in ft.-lbs. per lb. per degree Rankine. Equation (2–7) shows that if the three characteristics for a certain amount of gas are known, the weight can be found; or if the weight is known, any one of the three characteristics can be found if the two others are measured.

Example 2–9. Find the weight of air contained in a 2-cu. ft. cylinder, at a pressure of 100 psig. and having a temperature of 72° F; the gas constant of air $R = 53.3$

The absolute pressure, by expression (2–6), assuming a normal barometric pressure $b = 14.7$ psia., is $p = 100 + 14.7 = 114.7$ psia. or 114.7×144 16,500 psfa.

The absolute temperature, by expression (2–5), is $T = 72 + 460 = 532°$ Rankine. Solving equation (2–6) for W and substituting the corresponding numerical values gives

$$W = pV \div RT = 16{,}500 \times 2 \div (53.3 \times 532) = 1.16 \text{ lbs.}$$

Example 2–10. Determine what will happen if the air in Example 2–9 is heated to 150° F.

The new absolute temperature is $T = 150 + 460 = 610°$ R.

The characteristic which will change is p. Solving equation (2–7) for p, and substituting the corresponding values gives

$$p = WRT \div V = 1.16 \times 53.3 \times 610 \div 2 = 18{,}900 \text{ psfa.}$$

Converting 18,900 psfa. to psia. gives $18{,}900 \div 144 = 131.2$ psia.

The gauge pressure will be, according to equation (2–6)

$$p_g = 131.2 - 14.7 = 116.5 \text{ psig.}$$

or an increase of 16.5 psi. over the original pressure of 100 psig.

2–6. Energy. Energy of a body is the amount of work it can do. Energy exists in several different forms; a body may possess energy through its *position*, *motion*, or *condition*. Energy due to a position occupied by a body is called *mechanical potential* energy. An example of mechanical potential energy is a body located at a higher level, such as water behind a dam. When a body is moving with some velocity, it is said to possess energy of motion or *kinetic energy;* for example, a ball rolling upon a level floor. A third form of energy is *internal* energy, or energy stored within a body, either gas, liquid or solid, due to the forces between the molecules or atoms composing the body, such as in steam or gas under pressure. *Chemical* energy in fuel or in a charged storage battery is also classified as internal energy.

These three forms of energy, mechanical potential, kinetic, and internal, have in common the characteristic of being forms in which energy may be stored away for future use.

Work can be classified as mechanical or electrical energy in the *state of transformation* or *transfer*. Work done by raising a body stores mechanical potential energy in the body due to the force of gravity; work done to set a body in motion stores kinetic energy; work done in compressing a gas stores internal energy in the gas; electrical work can be transformed into mechanical work by means of an electric motor and after that it may undergo other changes the same as mechanical work.

Heat, like work, is energy in the *state of transfer* from one body to another, due to a difference in temperature of the bodies.

Units of energy. There are two basic independent units of energy:

1. The *foot-pound* (ft.-lb.) is the amount of energy as shown by work and is equivalent to the action of a force of 1 lb. through a distance of 1 ft.

2. The British thermal unit (Btu.) is the energy required to raise the temperature of 1 lb. of pure water by 1° F at a standard atmospheric pressure of 14.70 psia.

The conversion factor from ft.-lb. to Btu. units, often called the *mechanical equivalent of heat*, is 1 Btu. = 778 ft.-lbs.

There are two other energy units used in engineering calculations derived from the basic unit of ft.-lbs.:

3. *The horsepower-hour* (hp.-hr.) which is the transfer of energy at the rate of 33,000 ft.-lbs. per min. during 1 hr., or a total of 1,980,000 ft.-lbs., or, using the factor 778, 1 hp.-hr. = 1,980,000 ÷ 778 = 2,544 Btu.

4. The *kilowatt-hour* (kw.-hr.) which is the transfer of energy at the rate of 1,000 watts per hour or 1.341 hp. per hour which is equivalent to 44,253 ft.-lbs.

per min. during 1 hr., or a total of 2,655,180 ft.-lbs. or also 1 kw.-hr. = 2,655,180 ÷ 778 = 3,412 Btu.

Kinetic energy of a body is computed by the formula

$$KE = \tfrac{1}{2}(W \div g)v^2 \qquad (2\text{-}8)$$

where g is the acceleration due to gravity, 32.2 ft./sec.^{2} and v is the velocity of the body, ft./sec. Work done by kinetic energy is due to a change of velocity from an initial value of v_1 to a final value v_2.

Example 2-11. Find the work done by 500 lbs. of exhaust gases discharged upon the blades of a supercharger turbine if the initial velocity of the gases was 9,000 ft./min. and the exit velocity was 5,400 ft./min.

First the velocity must be changed to ft./sec.:

$v_1 = 9{,}000 \div 60 = 150$ ft./sec.;

$v_2 = 5{,}400 \div 60 = 90$ ft./sec.

By expression (2-8) the kinetic energies before and after the turbine are

$KE_1 = \tfrac{1}{2}(500 \div 32.2) \times 150^2 = 175{,}000$ ft.-lbs.

$KE_2 = \tfrac{1}{2}(500 \div 32.2) \times 90^2 = 63{,}000$ ft.-lbs.

and the work done = 175,000 − 63,000 = 112,000 ft.-lbs.

The principle of conservation of energy states that energy may exist in many varied and interchangeable forms but may not be quantitatively destroyed or created. Thus, mechanical energy may be transformed into heat, or vice versa, but only in a definite relation as given before, 1 Btu. = 778 ft.-lbs. Potential or internal energy may be changed to kinetic energy, etc.

2–7. Heat flow. As stated before, heat is a form of energy in a state of change; it is expressed in Btu. Quantitively a flow of heat is determined by the change of temperature of a body—heat is conveyed if the temperature of the body rises, and taken away if the temperature goes down. A quantitive measurement of heat is possible only by comparison with the behavior of some other body selected as a standard. Since the heat unit, Btu., is determined with the aid of water, water is used as a standard for determination of the behavior of all other substances in respect to a change of heat.

Specific heat. The specific heat of a substance, is the ratio of heat flow required to raise by 1° the temperature of a certain weight of the substance, to the heat flow required to raise by 1° the temperature of an equal weight of water. Due to the definition of 1 Btu., the specific heat of water is unity, or 1 Btu./lb.-deg. F, and numerically the specific heat of a substance is equal to the heat flow, in Btu., required to raise by 1° the temperature of 1 lb. of the substance. Denoting the specific heat by c, the heat flow Q, required to raise the temperature of W lb. of a substance from t_1 to t_2 degree F is:

Heat = Weight (of body) × specific heat × temperature difference or

$$Q = Wc(t_2 - t_1) \qquad (2\text{-}8)$$

In general, specific heat varies with the temperature, and for gases depends also upon conditions of pressure and volume. For many calculations, a mean value of specific heat can be used.

Example 2-12. Find the heat which is transferred to 53 gals. of lubricating oil when its temperature rises from 70° F. to 165° F. The specific heat of the oil is 0.5 Btu./lb.-deg. F. and its specific gravity is 0.925.

1 gal. of this oil weighs 8.34 × 0.925 = 7.71 lbs. and 53 gals. of it weigh 53 × 7.71 = 408 lbs. Therefore by expression (2-9) the heat transferred

$Q = 408 \times 0.5(165 - 70) = 19{,}380$ Btu.

Heat transfer. Generally speaking, heat is transferred by three methods: *conduction*, *radiation*, and *convection*.

Conduction is energy transfer by actual contact from one part of a body having a higher temperature, to another part of it or to a second body having a lower temperature.

Radiation is energy transfer through space from a hotter body to a colder body.

Convection is not a form of energy transfer. By convection is designated a process in which a body and the energy in it are moved from one position to another without change of state. An example of convection is the movement of heated air from one part of a room to another.

Basic principle of heat flow is that heat can flow from one body to a second body only if the temperature of the first body is higher than the temperature of the second body.

2–8. Problems. 1. Find the area of a piston which has a diameter of 6¼ in. *Ans.* 30.68 sq. in.

2. Find the volume of air contained in a pipe that has an inside diameter of 3.07 in. and is 12 ft. long. *Ans.* 1,066 cu. in. or 0.617 cu. ft.

3. Find the velocity of the water in a river if a piece of driftwood floats a distance of 6.5 miles in 2 hr. 10 min. *Ans.* 3 mph. or 264 ft./min.

4. Determine the force which tends to stretch the bolts holding a round cover 12 in. in diameter on a pressure vessel if the inside pressure is 400 psig. *Ans.* 45,240 lbs. or 22.62 tons.

5. Find the weight of 54 gals. of lubricating oil which has a specific gravity of 0.915. *Ans.* 412 lbs.

6. Find the gauge pressure in psi. corresponding to a column of (a) 12 in. of mercury, (b) 25 in. of water. *Ans.* 5.89 psig.; 0.90 psig.

7. Find the absolute pressure for the conditions of problem 6 if the barometric pressure is 29.6 in. mercury. *Ans.* 20.42 psia.; 15.43 psia.

8. An engine cylinder has a bore of 6¼ in. and a piston stroke of 8 in.; when the piston is in its highest position, nearest to the cylinder head, the volume of the gases enclosed in the cylinder is 49.09 cu. in. Find the volume of gases when the piston is in its lowest position, farthest away from the cylinder head. *Ans.* 294.53 cu. in.

9. The absolute temperature of a gas is 1,095° Rankine. What is its temperature in degrees Fahrenheit? *Ans.* 635° F.

10. Find the weight of the gases in the engine cylinder of problem 8, assuming that at the lowest position of the piston the gases are air, the pressure and temperature are atmospheric, standard conditions. *Ans.* 0.01303 lb.

11. Find the work necessary to lift 12,500 gals. of oil to a level 12 ft. 7 in. higher. One gal. of oil weighs 7 lbs. *Ans.* 1,100,000 ft.-lbs.

12. Determine the power required to do the work of problem 11 if the oil must be lifted in one-half hour. *Ans.* 1.11 hp.

13. Express the work of problem 11 in Btu. *Ans.* 1,413 Btu.

14. Find the kinetic energy of a truck moving with a speed of 45 mph.; its weight is 6,750 lbs. *Ans.* 457,000 ft.-lb.

15. Find the heat carried away per hour by the jacket cooling water of an engine, if the water flows at a rate of 11 gals. per min., and the average inlet temperature of the water is 124° F, and the outlet temperature is 156° F. *Ans.* 176,200 Btu./hr.

2–9. Questions. 1. What are the standard units used in the United States?

2. What is the difference between standard and derived units?

3. What is velocity?

4. What is the difference, if any, between speed and velocity?

5. What is acceleration?

6. What is pressure? How is it computed?

7. What is specific gravity?

8. What is work? How is it computed?

9. What is power?

10. What are the units of power used in engineering?

11. What is absolute pressure? How is it computed?

12. In what calculations must absolute temperatures be used?

13. Enumerate the three main forms of energy.

14. Can work or heat be stored as such?

15. Enumerate the units of energy used in engineering.

16. What is the principle of conservation of energy?

17. How is heat flow measured?

18. What is specific heat?

19. What are the methods of heat transfer?

20. What is the difference between conduction and convection?

21. What is the basic principle of heat flow?

CHAPTER 3

PETROLEUM PRODUCTS

3–1. Crude petroleum. Petroleum or crude oil is usually a dark brown liquid and is a mixture of a large number of compounds. The main chemical elements which form all these compounds are *hydrogen* and *carbon*. For this reason the petroleum compounds are called *hydrocarbons*. The amount of hydrogen present in the compounds varies from about 11 to 15 per cent, by weight, the balance being carbon. Some of these hydrocarbons, when separated from the crude oil, are gases at ordinary temperatures, and are stored under pressure in steel cylinders and used as fuel for household use. Other compounds, when separated from the crude oil, exist as solids at ordinary temperatures, for example, asphaltum used as a road surfacing material, or paraffin wax, used for sealing jars of home-canned vegetables and preserves. The majority of the compounds are liquids that vary considerably in color, specific gravity, and viscosity, such as gasoline and lubricating oil. Crude petroleum also contains varying amounts of impurities, such as sulfur, oxygen, nitrogen, water, salt, sand, and clay.

Crude petroleum is found in many parts of the world. The crude oil found in a certain locality normally exhibits some properties which distinguish it from crude oils found in other places. In the continental United States, there are three main oil-producing areas, and the characteristics of petroleum from these areas differ considerably.

Eastern crude oils consist chiefly of compounds containing much light oil or gasoline and considerable amounts of paraffin wax. They are nominally termed paraffin-base oils. They contain very little asphaltic material and sulfur. The proportion of hydrogen in products from this crude is usually higher than in similar products from crudes from other fields.

Mid-continent crudes vary within wide limits. In general, they contain a slightly lower amount of paraffinic compounds and more asphaltic materials. They are nominally classified as *mixed-base* oils. In some cases these crudes contain sulfur in such quantities as to require special treatment for its removal.

Western or *California* crudes contain large amounts of asphaltic materials and relatively high percentages of sulfur. They are termed *asphalt-base* oils.

Large producing areas exist in other parts of the United States. These sources produce crudes that, according to their characteristics, can be classified with one of the three main types.

3–2. Distillation. The separation of petroleum into its products is accomplished by what is essentially *fractional distillation*. The means by which this is accomplished depends upon the type of crude oil and the products desired. Fractional distillation can be briefly described as follows: The crude oil contained in a closed vessel is heated by a coil through which steam or hot gases are circulated. At first, the low-boiling compounds are driven off the crude in vapor form. The vapors are taken away by a pipe connected to the top of the vessel, condensed by cooling with a coil, through which relatively cold water or oil is circulated, and drained into another vessel. The temperature of the crude oil is maintained constant. After all compounds which boil under and at this temperature are driven off or distilled, the flow of hot gases through the heating coil is increased, the temperature of the crude goes up, and the vapors which are distilled are drained, after condensing, into another vessel, etc. In the distillation process the first three main products obtained, in the order of their increasing boiling temperature, called the *boiling point*, are (1) gasoline, (2) kerosene, and (3) gas and fuel oil. Lubricating oils are distilled off later or left in the unvaporized residue.

3–3. Diesel fuels. Fuel oils suitable for modern high-speed diesel engines and meeting the Navy's specifications are obtained either by a straight distillation of petroleum, or by special treatment of certain gas oils, or by blending several oils. They range from a heavy kerosene to a medium gas oil.

The properties of fuel oils have considerable influence on the performance and reliability of a diesel engine. The chief fuel properties affecting engine operation are:

1. Ignition quality.
2. Volatility.
3. Carbon residue.
4. Viscosity.
5. Sulfur content
6. Ash.
7. Water and sediment.
8. Flash point.
9. Pour point.
10. Acidity.
11. Heating value.

Ignition quality is one of the most important properties of a diesel fuel. It is expressed by an index called the *cetane number*. Present high-speed diesel engines require a cetane number of about 50. Its value as a diesel-fuel characteristic is similar to the octane number of gasoline. More information about cetane number is given in Sec. 3–5. The ignition quality of a fuel determines not only the ease of ignition and of starting cold engines, but also the kind of combustion obtained from the fuel. A fuel with a better ignition quality, a higher cetane number, gives easier starting, even at low temperatures, together with a quicker warm-up, smoother and quieter operation, lower maximum cylinder pressures, and more efficient combustion, hence lower fuel consumption.

Volatility of a diesel fuel is indicated by the *90 per cent distillation temperature*. This is the temperature at which 90 per cent of a sample of the oil has distilled off. The lower this temperature, the higher the volatility of the fuel. In small diesel engines a higher fuel volatility is necessary than in larger engines to obtain a low fuel consumption, low exhaust temperatures, and minimum smoke. For high-output naval diesel engines the maximum 90 per cent distillation temperature is 675° F.

Carbon residue is left after burning off volatile matter in a sample of oil under certain conditions. It indicates the tendency of the fuel to form carbon deposits on engine parts. Naval specifications allow a maximum carbon residue of 0.10 per cent.

Viscosity of a fluid is the measure of its internal friction or its resistance to flow. In practice, viscosity is expressed by the number of seconds required for a certain volume of liquid at some standard temperature to flow out through an orifice or hole of a definite small diameter. The greater this number of seconds is, the higher is the viscosity of the fluid. The instrument used in the United States for determining viscosity of oils is the *Saybolt Viscosimeter* with the *Universal* orifice, and the data it gives are designated as *S.S.U.*, seconds, Saybolt Universal. Viscosity controls lubrication and friction between the moving parts and hence controls their wear. Lubrication of the parts in the fuel-injection system depends entirely upon the fuel oil, and therefore its viscosity cannot be below a certain minimum value. For Navy diesel fuel a viscosity of 35–45 S.S.U. at 100° F is prescribed.

Sulfur in the fuel burns with the fuel oil in the engine cylinder and produces highly corrosive gases that are condensed by the cooled cylinder walls, especially when the engine operates under a low load and the cylinder temperature drops. Corrosion due to sulfur gases from fuel oil is frequently found in the exhaust system of diesel engines. Navy specifications do not permit a sulfur content over 1.00 per cent.

Ash and *sediment* in a fuel are a source of abrasive material which will cause excessive engine wear. Sediment may also cause clogging of the fuel system. Wear may be increased due to corrosion when the fuel contains water, especially salt water. A maximum permissible content of ash is 0.01 per cent and of water and sediment together, 0.05 per cent.

Flash point is the minimum temperature to which an oil must be heated in order to give off inflammable vapors in a sufficient quantity to flash or momentarily ignite when brought in contact with a flame. A fuel oil having an excessively low flash point is dangerous in storage and handling. The minimum flash point for diesel fuel is 150° F.

Pour point is the temperature at which an oil solidifies or congeals. It is of particular importance in connection with cold-starting an engine and for handling the oil between the storage and the engine. The maximum pour point for diesel fuel is 0° F.

Acidity. A fuel oil must not be corrosive, *must not contain free acids*, in order not to damage metal surfaces with which it comes in contact in storage or in the engine.

Importance of specifications. Some of the harmful effects, which may result from a fuel oil which does not comply with the specifications as outlined are:

1. Difficult cold-starting of the engine.
2. Reduced maximum power output.
3. High fuel consumption.
4. Smoky exhaust.
5. Rough and noisy engine operation.
6. Piston-ring and valve sticking.
7. Deposits of carbon and gummy substances on pistons and cylinder liners.

8. Contamination of lubricating oil.
9. Excessive wear of various engine parts.

3–4. Combustion. Combustion is a chemical reaction in which certain elements of the fuel combine with oxygen of the air, causing an increase in temperature of the gases. The main combustible elements are carbon and hydrogen; another combustible element, undesirable and contained in small amounts, is sulfur. The oxygen necessary for combustion is obtained from air which is a mixture of oxygen and nitrogen. Nitrogen is an inert gas and does not participate in the combustion process. During the process of combustion, the fuel oil particles are split into their component elements, hydrogen and carbon, and each element combines with oxygen of the air separately. Hydrogen combining with oxygen forms water, and carbon and oxygen combine to form carbon dioxide. If there is not enough oxygen present, part of the carbon will combine with oxygen, forming carbon monoxide. When carbon monoxide is formed, the amount of heat developed by each unit of carbon is only 30 per cent of the heat developed by the formation of carbon dioxide.

The actual diesel engine produces only a very small amount of carbon monoxide. Free carbon, or a greasy soot, is the usual result of incomplete combustion.

The heat value of diesel fuel varies, but an average is about 18,600 Btu. per lb. on the basis of the *lower heat value*, when the water formed by the combustion of hydrogen is in the state of vapor, as in the case of exhaust gases from an engine.

Theoretically, about 14.5 lbs. of air are required for the combustion of 1 lb. of fuel oil. However, under such conditions, some particles of oxygen, diluted by nitrogen and products of combustion, due to the exceedingly short time during which combustion must take place, will not be able to participate in the process of combustion. Free carbon will be released and some carbon monoxide will be formed. Therefore, to insure complete combustion of the fuel and avoid the heat loss due to formation of carbon monoxide and release of free carbon, an excess amount of air is always supplied to the cylinder. The ratio of the amount of air supplied to the quantity of fuel injected during each power stroke, both measured in pounds, is called the *air-fuel ratio*. The air-fuel ratio is very important in the operation of an internal-combustion engine. When a diesel engine is operating at light loads, the actual air-fuel ratio is several times greater than the theoretical value of 14.5. As the load is increased, the air-fuel ratio decreases; but even when the engine is over-loaded, the air-fuel ratio must be at least 25–30 per cent greater than 14.5. There should be present that much of an excess of air over the minimum required for complete combustion.

3–5. Ignition. In diesel engines, the fuel is ignited and combustion started by the heat of compression alone. The procedure is as follows: The fuel enters the cylinder as a fine mist or fog; the cylinder at this moment is filled with almost pure air whose temperature is raised by compression to about 1,000-1,050° F; the particles of fuel pick up heat from the air, begin to turn into vapor, and soon some vapor particles ignite. This igniting develops additional heat and helps to ignite other vapor particles.

Ignition delay. The time which it takes to heat the fuel particles, turn them into vapor, and bring about combustion is called *ignition lag*, or ignition *delay*. It depends upon the ignition quality of the fuel and certain characteristics, particularly engine speed and compression ratio. In present high-speed engines it takes 0.0012 to 0.0018 sec. This time decreases with an increase of the engine speed due to the improvement of air turbulence, and resultant better heating of the admitted fuel. It also decreases with an increase in compression pressure.

Cetane number. The cetane number serves to measure the ignition quality of a given fuel. The cetane number of the fuel is the percentage of cetane in a mixture of *cetane* and *alpha-methyl-naphthalene* that has the same ignition quality as the fuel which is tested. Both cetane and alpha-methyl-naphthalene are hydrocarbons, produced chemically from tar oil. Cetane has an excellent ignition quality but alpha-methyl-naphthalene has a very poor ignition quality. The scale runs from 0 to 100, pure alpha-methyl-naphthalene corresponding to 0 and pure cetane to 100. A cetane number of 48 would mean a fuel equivalent to a mixture composed of 48 per cent cetane and 52 per cent alpha-methyl-naphthalene.

The cetane number of a fuel sample is determined by using it in a special single-cylinder test engine with a variable compression ratio. The test procedure is based on the fact that the ignition-delay period in a given engine at a fixed engine speed decreases with an increase of the compression ratio. The delay period is measured from the moment when the fuel-injection valve leaves its seat until the ignition of the fuel produces a measurable pressure rise in the cylinder. An ignition-delay period of standard length, 13 crank-angle degrees, is used as reference, the test fuel is burned in the engine, and the compression ratio is raised in the engine until the 13-degree ignition-delay period, as indicated by special instruments, is reached and the required compression ratio is noted. Next, the engine is run using two mixtures of cetane and alpha-methyl-naphthalene, one having a cetane number about 5 units higher and the other about 5 units lower than the expected cetane number of the fuel. The compression ratios of these mixtures to obtain a 13-degree ignition delay are found, and by proportioning or interpolation, the exact cetane number of the sample is easily computed. Fuels with good ignition qualities require lower compression ratios for the 13-degree ignition delay and have a higher cetane number. Fuels with poorer ignition qualities require higher compression ratios for the 13-degree ignition delay and have a lower cetane number.

3–6. Gasoline as diesel fuel. A good gasoline with a high octane number, does not knock in a spark-ignition engine because it burns comparatively slowly, hence it does not ignite as readily, and has a correspondingly low cetane number. In general, the higher the octane number of a fuel, the lower its cetane number, and vice versa.

Thus, gasoline is not a desirable diesel fuel. However, tests have shown that diesel engines can run on gasoline, although the engine will be rather noisy and rough and the fuel consumption will be higher than with regular diesel fuel oil. By adding 10 to 15 per cent of regular diesel fuel, the performance of the engine can be considerably improved. Another drawback of running a diesel engine on gasoline is the danger of wear and seizure of the fuel-injection pumps because of the low viscosity of gasoline. Addition of lubricating oil to gasoline will raise the viscosity and help to overcome this trouble.

3–7. Lubricants. All lubricants used in diesel engines may be divided into two groups: the important group of *oils* and a small group of *greases*. Both oils and greases have as their main object the separation of metal surfaces in motion and thus prevention or at least reduction in wear on such surfaces.

1. *Lubricating oils.* Lubricating oils used in diesel engines are obtained by distillation of the stock left after the lighter fractions, gasoline, kerosene, and gas oils, which include diesel fuel oils, are driven off the crude petroleum. They are called *mineral* oils to differentiate them from vegetable and animal oils used with other machinery. Lubricating oils are hydrocarbons, the same as diesel fuel oil, but differ from it by the internal structure of their particles which is shown chiefly in their greater viscosity and heavier specific weight or gravity. The desired properties of lubricating oils are obtained by mixing, or more accurately, blending, of oils distilled from different stocks and at different temperatures.

Properties of lubricating oils, both physical and chemical, should be such that the oil will give satisfactory engine performance. These properties are determined by tests similar to those used for testing fuel oils. Approximately in the order of their importance they are:

a. *Viscosity.* This is the most important property. Viscosity decreases with an increase of temperature and is determined with the already mentioned Saybolt Viscosimeter with the Universal orifice. Viscosity of Navy diesel oils for various engines varies from 90 seconds to about 500 seconds at 130° F.

Friction, engine wear, and oil consumption depend principally on the viscosity of the oil.

Viscosity index indicates the influence of the temperature on the viscosity of a particular oil. Viscosity index depends on the source of the crude and the methods used in refining. Straight paraffin-base oils have a viscosity index in the neighborhood of 100. A lower figure denotes an oil whose viscosity decreases faster with an increase of temperature. Fig. 3–1 illustrates the behavior of two oils—one having a high viscosity index, high V.I., and the other a low V.I. At 80° F their viscosity is the same, 280 S.S.U.; but at 180° F, the viscosity of the high V.I. oil is 160 S.S.U., while that of the low V.I. oil is only 100 S.S.U.

b. *Pour point.* Pour point is the lowest temperature

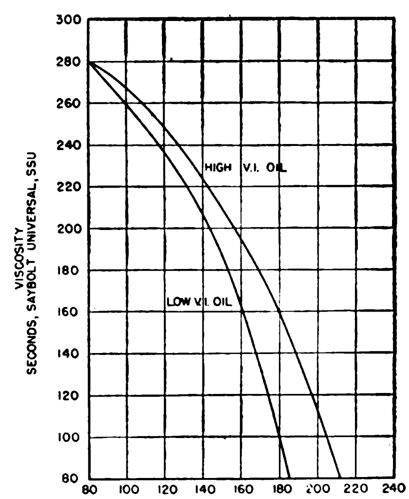


Figure 3–1. Change of viscosity with temperature.

at which the oil will barely pour from a container. This property affects the ability to pump the oil through the system with the numerous small-size tubings and orifices. A relatively high pour point may cause starting difficulties in cold weather.

c. *Carbon residue.* Carbon residue is the amount of carbon left after the volatile matter has been evaporated and burned off by heating the oil. It gives an indication of the amount of carbon that may be deposited in an engine, which will lead to operating troubles.

d. *Flash point.* Flash point of lubricating oil is determined in the same way as for fuel oil; however, its object is chiefly to indicate the quality of oil. The flash point of various naval diesel lubricating oils varies from 340° to 430° F.

e. *Water* and *sediment.* The oil is tested by centrifuging and should be free of water and sediment.

f. *Acidity.* Lubricating oils should show a neutral reaction when tested with litmus paper. An acid oil may corrode or pit the engine parts and has a tendency to form emulsions with water, and sludge with carbon. In service, oils have a tendency to become acid due to oxidation.

g. *Emulsion.* The oil should not form an emulsion with water. If shaken with water it should readily separate from it. This ability of separation is important in an oil after it has been used for some time. When exhaust gases, which always contain water vapor, enter into the crankcase due to blow-by, water vapor is condensed and mixed with oil in the crankcase.

h. *Oxidation.* The oil should not have an excessive tendency to oxidize, as oxidation causes formation of sludge.

i. *Ash.* Ash in oil is a measure of the foreign matter which may cause abrasion or scoring of the moving parts in contact.

j. *Sulfur.* Free sulfur or corrosive compounds of sulfur are not allowed in lubricating oils, because they have a tendency to form acids with water vapor. Non-corrosive compounds of sulfur are permitted to a limited extent.

k. *Color.* The color of a lubricating oil has no relation to its lubricating quality.

Additive oils. Higher cylinder pressures and temperatures, higher loads and speeds, used in present-day high-output diesel engines, have increased the requirements as to the qualities of the lubricating oils. It is difficult to meet these requirements with the use of straight mineral oils. One of the main troubles encountered with high-output engines is the sticking of piston rings which in turn causes a number of other troubles, such as reduction of power, smoky exhaust, and contamination of lubricating oil. In trying to overcome such troubles, the oil refineries found that certain chemicals added to the oil, and called additives, increase the resistance to oxidation, and other additives help materially in keeping the piston rings from sticking, serving to cleanse or to *deterge* them. This explains why these heavy-duty additive diesel engine oils are often called *detergent* oils. They have taken the place of the straight mineral oils in the operation of naval diesel engines.

The chemicals in the additive oils do their work by combining with the undesirable contaminations. Therefore, the amount of free additives in the oil gradually decreases, the oil "wears out," and its ability to resist oxidation and deterge the piston rings gradually diminishes. Worn-out oil must be removed and replaced by fresh oil.

Navy symbols for identifying lubricating oils consist of four digits, of which the first classifies the oil according to its use and the last three indicate the viscosity in S.S.U. Forced-feed oils with the viscosity referred to as 130° F, have the first digit 2. Thus, a Navy symbol 2190 oil is a forced-feed oil with an approximate viscosity of 190 S.S.U. at 130° F. The symbol digit for detergent diesel oils is 9 and the viscosity is referred to 130° F. Thus Navy symbol 9370 oil is a detergent diesel oil with a viscosity of 370 S.S.U. at 130° F. The Navy uses three grades of additive diesel lubricating oils, designated by the symbols 9170, 9250, and 9370. Their viscosity corresponds approximately to SAE viscosity numbers 20, 30, 40, respectively.

2. *Greases.* Lubricating greases are emulsions or intimate mixtures of a lubricating oil with soap. With Navy diesel engines, greases generally are used only in two places—for the shafts of some centrifugal water pumps and on the inside of some reverse gears. The pump grease must contain soap which will not dissolve in water. Reverse-gear greases must resist high temperatures created by the friction in the gears. These greases may contain graphite which has the ability of lowering the friction coefficient.

3-8. Diesel engine lubrication. As already mentioned, the successful lubrication of modern high-speed diesel engines has become an increasingly difficult problem. On the other hand, good lubrication is even more vital for the proper operation and life of modern diesel engines. Some of the reasons for the encountered difficulties are:

1. Smaller engines with smaller oil reservoirs and higher speeds have increased engine temperatures.

2. Higher piston-ring temperatures have a tendency to cause ring sticking and subsequent piston seizures.

3. Higher engine speeds and temperatures have made proper cylinder and piston lubrication more difficult.

4. Higher cylinder pressures have resulted in greater bearing loads.

5. Higher engine temperatures, loads, and speeds have put new strain on bearing materials which in some cases have shown a tendency to corrosion and fatigue failure in spite of the use of special new bearing metals.

6. Bearings with the new precision bearing shells are more sensitive to solid particles in the lubricating oil and require improved oil purification.

7. The use of copper in bearings contributes to oil oxidation.

8. The increased use of two-stroke engines with their increased power output and supercharging of four-stroke engines have increased the cylinder pressures and temperatures.

9. The all-enclosed construction of high-speed diesel engines has made the prevention of fuel-oil leaks more difficult and has increased the rate of crankcase oil dilution with fuel oil.

10. The increased clearance necessary with aluminum-alloy pistons has resulted in higher oil consumption, particularly in larger engines when operating at partial loads.

11. The increased use of mobile diesel engines has brought up the problem of the viscosity of lubricating oil especially when diesel engines are started in subfreezing weather.

The best test of any lubricating oil is the way it behaves in the engine. The oil must have a viscosity which remains within suitable limits throughout the operating-temperature range of the engine. In addition to maintaining a sufficient oil film between the moving parts, the oil which unavoidably burns in the cylinder must leave a minimum of carbon residue. The oil must be stable, resist oxidation, acidity, and emulsification.

An ideal diesel-engine lubricant must fulfill all of the following requirements:

1. Maintain a good oil film on cylinder walls and thus prevent excessive wear of cylinder liners, pistons, and rings.

2. Prevent sticking of piston rings.

3. Seal compression in the cylinders.

4. Leave no carbon deposit on the crown and upper part of the piston.

5. Must not lacquer the piston or the cylinder surfaces.

6. Leave no carbon deposits in the exhaust and scavenging ports.

7. Prevent wear of bearings.

8. Clean the interior of the engine.

9. Must not form excessive sludge, nor clog oil lines, strainers, and filters, nor leave deposits in the oil cooler.

10. Be usable with any approved filter.

11. Show a low consumption rate.

12. Have good cold-starting properties.

13. Permit long intervals between renewals.

Evidently no single oil can meet all these requirements, and the oil that comes nearest to meeting the most important requirements is the one selected. In the Navy, the oils to be used in the various engines are specified by the Bureau of Ships after exhaustive tests have been conducted to determine the oil most suitable for a particular engine. Since the oil refineries are constantly trying to improve their products, new tests are continuously going on in an effort to find a better oil.

3–9. Problems. 1. Find the air-fuel ratio for an engine which uses per hour, 41 lbs. of fuel oil and 828 lbs. of air. Ans. 20.2.

2. Find the per cent of excess air for an engine which uses per hour, 762 lbs. of fuel oil and 14,100 lbs. of air. Ans. 27.6 per cent.

3–10. Questions. 1. Why are petroleum compounds called *hydrocarbons?*

2. What are the first three products obtained by fractional distillation of crude petroleum?

3. Enumerate four or five of the more important fuel properties.

4. What are the main products into which fuel oil is transformed during combustion?

5. What is the meaning of the term *cetane number?*

6. If a diesel engine must be run on gasoline, how can its performance be improved?

7. What is *viscosity* of a lubricating oil?

8. What oils are called *additive oils?*

9. What is a lubricating grease?

10. Enumerate a few of the factors which make satisfactory lubrication more difficult in modern high-speed, diesel engines.

11. Enumerate a few of the more important requirements of a diesel-engine lubricant.

CHAPTER 4

ENGINE CONSTRUCTION

4-1. Engine parts. Diesel engines vary greatly in outside appearance, size, number of cylinders, cylinder arrangement, and details of construction. However, they all have the same main basic parts which may look different but perform the same functions. There are only a very few basic main working parts to a diesel engine, and all the rest of the engine is composed of auxiliary parts which assist the main working parts in their performance, and the connecting parts necessary to hold the working parts together. The main working parts are:

1. Cylinder.
2. Piston.
3. Connecting rod.
4. Crankshaft.
5. Bearings.
6. Fuel pump and nozzle.

Naturally, there are a number of other parts without which an engine could not operate, but their functions are more or less subordinate and they will be discussed later.

Fig. 4-1 is a schematic drawing of a typical diesel engine. Its purpose is to show the main working parts and their relation to other parts.

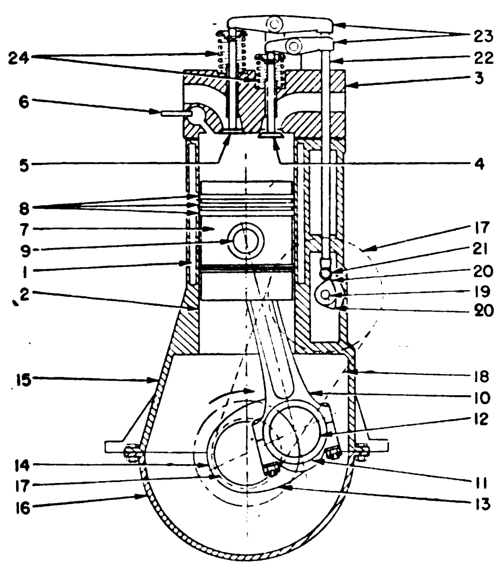


Figure 4-1. Schematic drawing of a diesel engine.

Parts shown on Figure 4-1.

1. Cylinder.
2. Cylinder liner.
3. Cylinder head.
4. Intake valve.
5. Exhaust valve.
6. Fuel injector.
7. Piston.
8. Piston rings.
9. Wrist or piston pin.
10. Connecting rod.
11. Crankpin bearing.
12. Crankpin.
13. Crank cheek or crank web.
14. Crankshaft.
15. Engine frame.
16. Crankcase.
17. Timing chain sprocket.
18. Timing chain.
19. Camshaft.
20. Cams.
21. Cam follower.
22. Push rod.
23. Rocker arm.
24. Valve spring.

Cylinder. The heart of the engine is the cylinder where the fuel is burned and the power developed. The inside of the cylinder is formed by the *cylinder liner* or *sleeve*, and the *cylinder head* which seals one end of the cylinder and often, although not always, contains the valves to admit fuel and air and to eliminate the used gases. The diameter of the cylinder is known as the *bore*.

Piston. The piston seals the other end of the working space of the cylinder and transmits to the outside the power developed inside of the cylinder by the burning of the fuel. A gastight seal between the piston and the cylinder liner is produced by *piston rings* lubricated with engine oil. The distance that the piston travels from one end of the cylinder to the other is known as the *stroke*.

Connecting rod. One end, called the *small end* of the connecting rod or conrod eye is attached to the *wrist pin* or *piston pin* located in the piston; the other or *big end* has a bearing which encloses the crankpin. The connecting rod transmits the reciprocating motion of the piston to the continuously rotating crankshaft during the working stroke, and vice versa during the other strokes.

Crankshaft. The crankshaft obtains its rotary motion from the piston through the connecting rod and *crankpin* located between the crank *webs* or crank *cheeks*. The work of the piston is transmitted to the propeller or generator drive shaft. A *flywheel* of a sufficient mass is fastened to the crankshaft in order to reduce speed fluctuations by storing kinetic energy during the periods when power is developed and giving it back during the other periods.

A *camshaft* is driven from the crankshaft by a chain drive or by *timing gears*. Through *cam followers*, *push rods*, and *rocker arms*, the intake and exhaust *valves* are opened by cams on the camshaft. Valve *springs* serve to close the valves.

A *crankcase* is constructed to protect the crankshaft, bearings, connecting rods, and related parts, to catch the oil escaping from the bearings of the moving parts, and to provide a reservoir for lubricating oil. If the crankcase is constructed to support the whole engine, it is called a *bed plate*.

Fuel for diesel engines is delivered into the combustion space of the cylinder by an injection system consisting of a *pump*, *fuel line*, and *injector*, also called *injection* or *spray nozzle*.

4–2. Engine types. Diesel engines may be divided into several classes using different bases for the division: (1) operating cycle; (2) cylinder arrangement; (3) piston action; (4) method of fuel injection; (5) speed, etc.

Operating cycles. Diesel engines can be divided into two groups based on the number of piston strokes per cycle, in (1) *four-stroke-cycle*, or, for short, *four-stroke* engines, and (2) *two-stroke-cycle*, or *two-stroke* engines. The meaning of these terms and the difference between these two engine types is explained in Chapter 5.

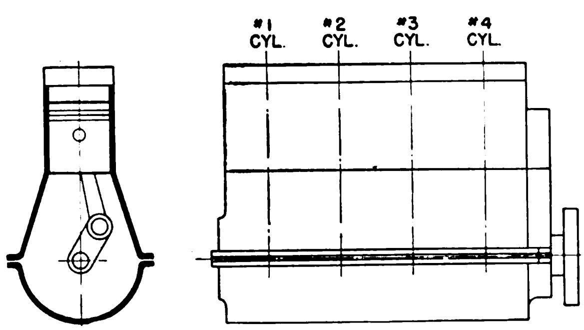


Figure 4–2. In-line engine.

Cylinder arrangement. Cylinders-in-line. This is the simplest arrangement with all cylinders parallel, in line, as shown in Fig. 4–2. This construction is used for engines having up to 10 cylinders.

V-arrangement. If an engine has more than 8 cylinders, it becomes difficult to make a sufficiently rigid frame and crankshaft with an in-line arrangement.

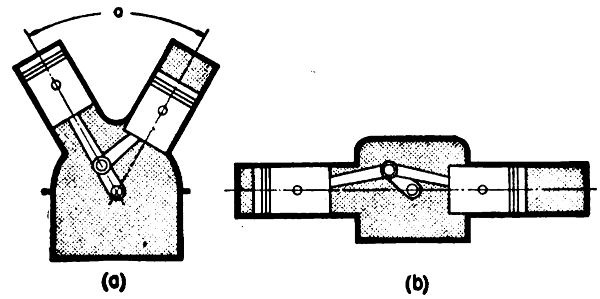


Figure 4–3. Vee-type engines.

The V-arrangement, Fig. 4–3a, with two connecting rods attached to each crankpin, permits reducing the engine length by one-half, thus making it much more rigid, with a stiff crankshaft. This is a common arrangement for engines with 8 to 16 cylinders. Cylinders lying in one plane are called a bank, and the angle *a* between the banks may vary from 30° to 120°, most common angle being between 40° and 75°.

Flat engine, Fig. 4–3b, is a V-engine with an angle of 180° between the banks. This arrangement is used mostly for trucks and buses.

Multiple-engine units. In order to increase the engine power without increasing its bore and stroke, two and four complete engines, having six or eight cylinders each, are combined in one unit by connecting

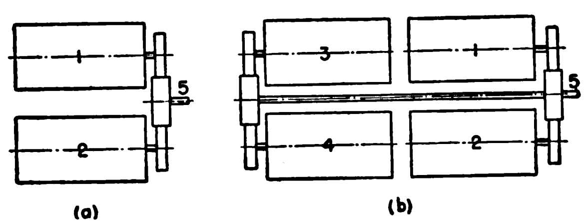


Figure 4–4. Multiple-unit engines.

each engine to the main drive shaft 5, Figs. 4–4a and 4–4b, by means of clutches and gears or clutches and roller chains. Fig. 4–4a shows a twin-engine and Fig. 4–4b a quadruple-engine, or quad.

Vertical-shaft engines. A recent development is an engine with four connecting rods attached to one crankpin, Fig. 4–5. The four cylinders are all in one horizontal plane, the crankshaft thus being vertical. Four banks located one on top of the other and using one crankshaft with four cranks form a compact

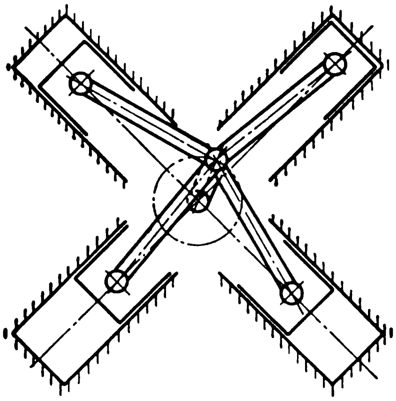


Figure 4–5. Top view of pancake engine.

16-cylinder engine used in the Navy under the name of the pancake engine.

There exists a great number of other cylinder arrangements but they are not at present used in United States Navy diesel engines.

Piston action. Single-acting engines use only one end of the cylinder and one face of the piston for the development of power. The engines in Figs. 4–1 and 4–2 are all single-acting. Nearly all diesel engines are single-acting.

Double-acting engines use both ends of the cylinder and both faces of the piston for the development of power. Double-acting engines are built only in large and comparatively low-speed units. A diagrammatic sketch of a double-acting engine is shown in Fig. 5–5.

Opposed-piston engines. Recently, many engines having two pistons per cylinder, driving two crankshafts, Fig. 5–6a, have been placed in Naval service. The design presents many advantages from the viewpoint of combustion of the fuel, engine maintenance, and accessibility of all parts, except the lower crankshaft.

Fuel injection. Diesel engines are divided into *air-injection* engines and *solid-* or *mechanical-injection* engines. The meaning of these terms and the differences between these two types is discussed in Chapter 9. Here we need only to mention that air-injection engines are gradually disappearing both from naval and land installations.

Speed. All diesel engines can be divided into three classes: low-speed, medium-speed, and high-speed engines. The basis for the division is discussed in Chapter 16. Here it is enough to state that the present trend is away from low- and even medium-speed engines toward increasingly higher-speed engines.

4–3. Engine designation. In the United States Navy, engines are designated by the number of cylinders, bore, stroke, and speed, if the engine operates at a definite speed, in the order named. Thus a 6-cylinder × 3¾-in. × 5 in. × 1,500 rpm. or often simply a 6×3¾×5×1,500 designation means an engine with 6 cylinders, a 3¾ in. bore, 5-in. stroke, normally operated at 1,500 rpm.

Another designation is by *piston displacement*, or, for short, by displacement. The engine displacement is computed as the product of the piston area times the piston stroke and times the number of cylinders. In English units, displacement is expressed in cubic inches. The engine speed does not enter in the determination of the engine displacement. Sometimes the displacement is indicated as the number of cylinders and the displacement of one cylinder.

Example 4–1. Find the piston displacement of an 8×8×10×720 naval engine.

The piston area $= \pi \times 8^2/4 = 50.27$ sq.in.; the stroke is 10 in. and the displacement of one cylinder is $50.27 \times 10 = 502.7$ cu. in.; since the number of cylinders is 8, therefore the total

$$\text{displacement} = 50.27 \times 10 \times 8 = 4{,}021.6 \text{ cu.in.}$$

or the engine may be designated as 8–503, to avoid decimal fractions in the designation.

4–4. Problems. 1. Write the Navy description for an engine which has 6 cylinders with a bore of 6¾ in., and a stroke of 8¾ in. and develops 323 hp. at 900 rpm. *Ans.* 6×6¾×8¾×900.

2. Find the piston displacement of the engine of problem 1. *Ans.* 1,876 cu.in. or 6–313 cu.in.

4–5. Questions. 1. Enumerate the main working parts which are necessary in every diesel engine.

2. What are the main functions of the *crankcase?*

3. State on what bases diesel engines are divided into various types.

4. What arrangement of cylinders are encountered in naval diesel engines?

5. How are diesel engines designated in the United States Navy?

CHAPTER 5

DIESEL ENGINE PRINCIPLES

5-1. Four-stroke-cycle events. A round of events which reoccurs regularly and in the same sequence, is known as a *cycle*. The cyclic sequence of events in a diesel engine is: (1) filling of the engine cylinder with fresh air; (2) compression of the air charge in order to raise its pressure and temperature to that necessary to ignite and burn the fuel efficiently; (3) combustion of the fuel and expansion of the hot gases; and (4) emptying the cylinder of the burned gases by exhausting them. When these four events are completed, the cycle is repeated. When each of these events takes place, roughly speaking, during one stroke of the piston, the cycle is called a *four-stroke* cycle.

Piston dead centers. The positions of the piston when it is nearest to the cylinder head and farthest away from it are called *top* and *bottom dead center*, respectively, or for short, *top* and *bottom center*, indicated *t.c.* and *b.c.* The reason for this designation is that at these positions the connecting-rod center-line coincides with the crank-throw center-line and the piston cannot be moved by gas pressure acting upon its surface. The motive force must come from the rotating crank acting through the connecting rod.

Main events. The four main events are shown dia-

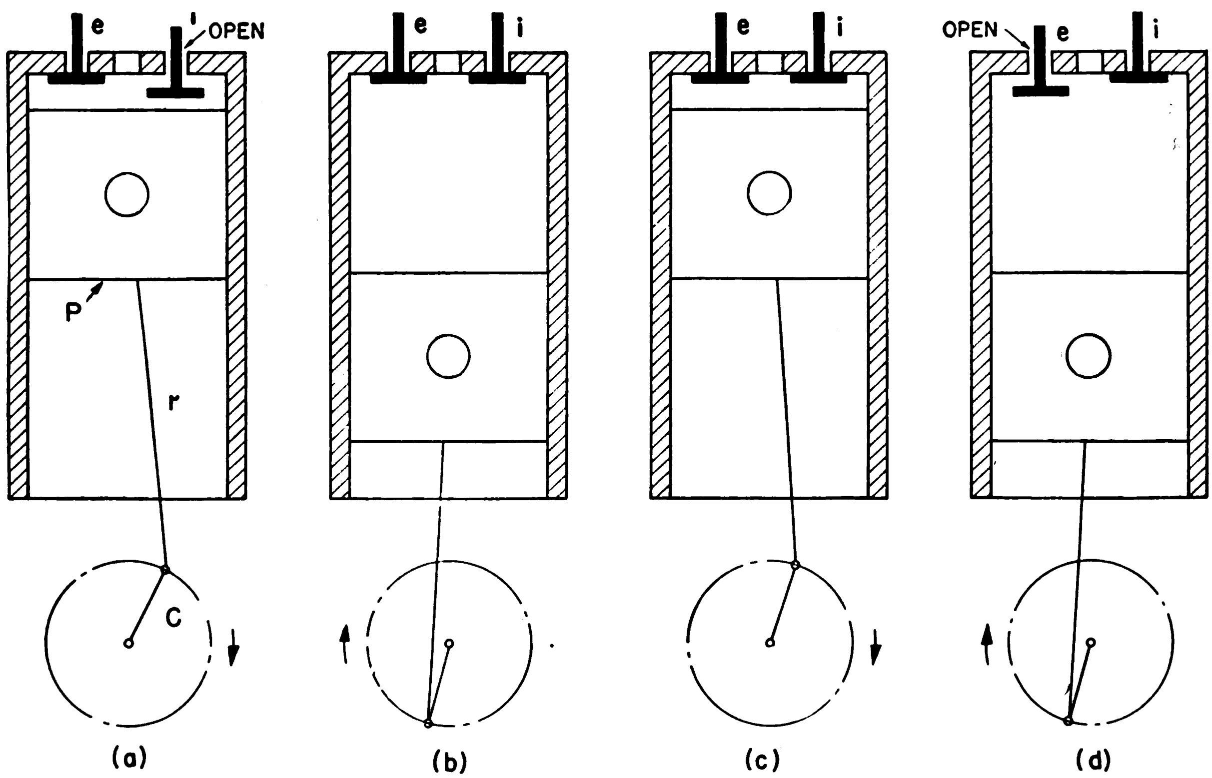


Figure 5-1. Four-stroke-cycle events.

grammatically in Fig. 5–1. During the first or *suction stroke*, Fig. 5–1a, the piston P moves downward, pulled by the connecting rod r, the lower end of which is moved by the crank C. The piston motion creates a vacuum in the cylinder, and outside air is drawn or sucked into the cylinder through the intake valve i which opens at about the beginning of the suction stroke and stays open until the piston reaches the lower or bottom center, *b.c*.

When the piston has passed *b.c.*, the second or *compression stroke* begins, Fig. 5–1b; the intake valve is closed and the upward motion of the piston pushed by the crank and the connecting rod begins to compress the air charge in the cylinder.

Shortly before the piston reaches top center, *t.c.*, liquid fuel in a finely atomized spray is admitted into the cylinder containing hot compressed air. The fuel is ignited by the heat of the air compressed in the cylinder and burns during the first part of the downward piston stroke. During this downward or third stroke, called *working* or *power stroke*, Fig. 5–1c, the hot gases whose pressure was considerably increased by the combustion of the fuel charge, force the piston downward and expand due to the increasing cylinder volume.

Shortly before the piston reaches the bottom center, the exhaust valve e opens, Fig. 5–1d, and the hot products of the combustion, having a relatively high pressure in spite of the previous expansion, begin to rush out through the exhaust ports into the outside atmosphere. During the following fourth or *exhaust stroke*, the piston moves upward, pushed by the crank and connecting rod, expelling the remaining products of combustion, until near top center the exhaust valve is closed, the intake valve is opened, and the whole cycle starts again. As can be seen, the four strokes require two engine revolutions. Thus in a four-stroke-cycle engine, one power stroke is obtained for every two engine revolutions, or the number of power impulses per min. are equal to one-half of the rpm. of the engine.

Timing of events. Actually, the dividing points between the four main events do not come at the very beginning and end of the corresponding strokes. The differences are smaller in low-speed engines and increase as the engine speed increases. The intake valve is opened *before* top center, 10 to 25 crank-angle degrees; it is closed from 25 to 45 crank-angle degrees *after* bottom center. The fuel injection starts some 7 to 26 degrees before *t.c*. In order to release the exhaust gases in proper time, the exhaust valve begins to open 30 to 60 degrees before *b.c*. and closes 10 to 20 degrees after *t.c*.

5–2. Compression. There are two reasons for compressing the air charge during the second or compression stroke: first, raising the thermal or over-all efficiency of the engine by increasing the final temperature of combustion; this applies to all internal-combustion engines, both of the spark-ignition as well as the so-called diesel type; second, increasing the temperature of the air charge so much that when the finely atomized fuel is injected into the hot air, the fuel will ignite and begin to burn without any outside source of ignition such as the spark plug used in automobile engines.

Compression ratio of an internal combustion engine is termed the ratio of the volume V_1, cu. in., of the gases in the cylinder when the piston is in its bottom center, to the volume V_2 of the same gases when the piston is at top center. Compression ratio is designated by r,

Compression ratio = volume at bottom center ÷ volume at top center or

$$r = V_1 \div V_2 \qquad (5\text{–}1)$$

The volume V_2 is called the compression or *combustion space*. The volume V_1, is equal to the sum of the piston displacement of one cylinder plus the combustion spaces.

Limits for compression ratios. Diesel engines use compression ratios of 12:1 up to about 19:1. Theoretically, the higher the compression ratio, the higher is the thermal efficiency of the engine, the lower its fuel consumption. However, an increase of the compression ratio is accompanied by higher gas pressures and combustion temperatures. This causes stresses and pressures in various engine parts, and to counteract these ill effects requires stronger, heavier parts and increases unduly the weight of an engine. Higher temperatures and pressures also increase the wear and tear of an engine and thus decreases its reliability.

In order to avoid the problems of high compression, a certain oil engine, the Waukesha-Hesselman, is built with such a low compression ratio, about 5:1, that the compression temperature becomes too low to produce self-ignition of the fuel when it is injected, and a spark-ignition system, as used in gasoline engines, is required. Naturally, the fuel consumption of this low-compression oil engine is considerably higher than that of a similar compression-ignition or diesel engine.

5–3. Combustion. There exist two distinctly different methods of burning the fuel in an engine cylinder:

(a) at a *constant volume* and (b) at a *constant pressure*. Combustion at constant volume means that during combustion the volume does not change and that all the heat energy developed by the fuel goes into an increase of the gas temperature and pressure. In the case of an engine, combustion at constant volume means that combustion proceeds at such a high rate that the piston has practically no time to move during combustion. Such a combustion is obtained when the piston is passing top center. The advantage of this method of fuel combustion is a high thermal efficiency. Its disadvantage is a very sudden pressure increase and resulting noisiness of the engine. Such combustion is somewhat approached by spark-ignition gasoline engines.

Combustion at constant pressure means that during combustion the temperature increases at such a rate that the resulting increase of pressure is just enough to counteract the influence of the increasing volume, and the pressure does not change. The heat energy developed by the fuel goes partly into an increase of the gas temperature and partly into performing outside work. In the case of an engine with constant-pressure combustion, the fuel is burned gradually so that the pressure attained at the end of the compression stroke is maintained during the greater part of the combustion event. Such a combustion was used in the original low-speed, air-injection diesel engine. Its advantage is a smoothly running engine producing a more even torque due to the extended combustion pressure. However, it is not suitable for high-speed oil engines.

High-speed diesel engines of the present operate on a cycle which is approximately a combination of the above two methods; part of the fuel is burned rapidly, almost at a constant volume near the dead center, the rest is burned while the piston begins to move away from the top center. However, the pressure does not remain constant, but usually increases and then decreases. In general this cycle resembles more the constant-volume combustion cycle than the cycle of the original diesel engines. Its advantage is a high efficiency, with low fuel consumption. Its drawback is in the difficulty of preventing rough and noisy operation of the engine.

5–4. Two-stroke-cycle events. A two-stroke cycle is completed in two strokes, or one revolution of the crankshaft, whereas a four-stroke cycle requires two revolutions. The difference between the two-stroke

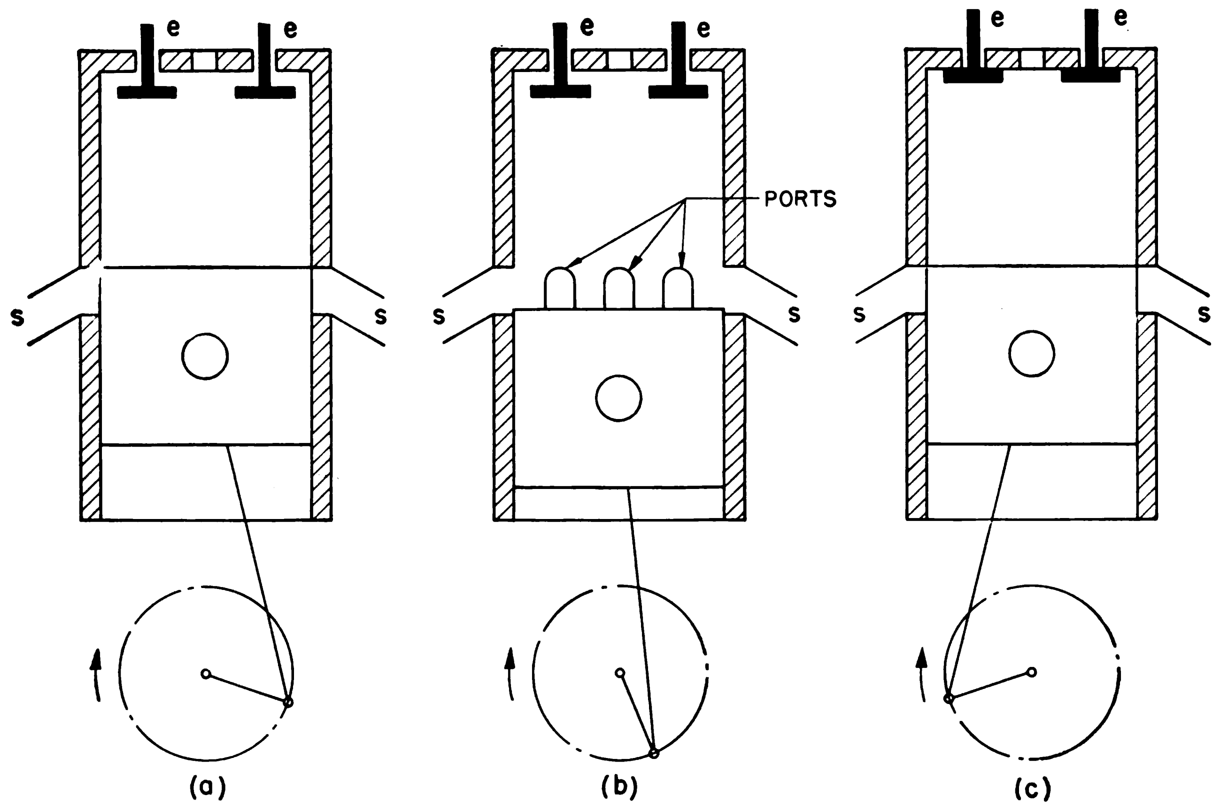


Figure 5–2. Scavenging of a two-stroke engine.

and four-stroke engine is in the method of removing the burned gases and filling the cylinder with a fresh charge of air. In a four-stroke engine these operations are performed by the engine piston during the exhaust and suction strokes. In a two-stroke engine these operations are performed near the bottom dead center by means of a separate air pump or blower.

The compression, combustion, and expansion events do not differ from those of a four-stroke engine. The filling of the cylinder, called *scavenging*, with a fresh charge may be explained as follows: when the piston has travelled 80 to 85 per cent of its expansion stroke, exhaust valves *e*, *e*, Fig. 5–2a, are opened, the exhaust gases are released and begin to escape from the cylinder. The piston continues to move toward the bottom center and soon uncovers ports *s*, *s*, through which slightly compressed air begins to enter the cylinder. This air, having a slightly higher pressure than the hot gases in the cylinder, pumps out the hot gases through valves *e*, *e*, Fig. 5–2b. This operation is called *scavenging*, the air admitted is called *scavenge air* and the air admittance ports, *scavenge ports*. About the time when the piston on its upward stroke closes ports *s*, *s*, the exhaust valves *e*, *e*, are also closed, Fig. 5–2c, and the compression stroke begins.

The *advantage* of two-stroke operation is the elimination of one scavenging and one charging stroke required in four-stroke-cycle operation. Thus, the cylinder delivers one power stroke for every revolution of the engine as compared with one power stroke for every two revolutions in a four-stroke-cycle engine. Theoretically, if all other conditions such as bore, stroke, speed, and gas pressures are equal, a two-stroke-cycle engine should develop twice the power of a four-stroke-cycle engine. This means also that a two-stroke engine should weigh only one-half as much as a four-stroke engine of the same power, and should produce a more even torque. Practical factors prevent the attainment of these ideal figures.

These advantages are very important in ship installations, and, therefore, two-stroke engines are used in the Navy much more than four-stroke engines, particularly in larger power units. A *disadvantage* of two-stroke operation is the higher working temperatures of the piston and cylinder head due to combustion occurring every revolution and resulting in distortion of these and related parts.

5–5. Scavenging methods. Fig. 5–2 illustrates only one of several methods of cylinder scavenging. In some engines the exhaust gases are let out through ports, uncovered by the piston the same as the scavenge ports *s*, *s*, Fig. 5–2. Depending upon the location of the exhaust ports in respect to the scavenge ports, there exist two basically different methods of scavenging: *direct* or *cross-flow* scavenging, Fig. 5–3, and *loop* or *return-flow* scavenging, Fig. 5–4.

In *cross-flow scavenging* the piston uncovers first the exhaust ports, *e*, *e* and releases the pressure; going down farther, the piston uncovers the scavenge ports *b*, *b* and begins to admit slightly compressed air whose stream is directed mainly upward, as indicated by the arrows, and thus pushes out the exhaust gases through ports *e*, *e*. Having passed the dead center, the piston closes first the scavenge ports and soon afterward the exhaust ports. The fact that the exhaust ports are closed after the scavenge ports permits some of the air charge to escape from the cylinder. This is a disadvantage of this scavenge scheme. However, it has also the decided advantage of simplicity of construction, and of maintenance, due to the absence of valves which must be kept tight.

Loop-scavenging, Fig. 5–4, is similar to cross-flow scavenging in the sequence of the port opening; the direction of air flow is different as indicated by the arrows. Its advantage is that the bulky scavenge-air and exhaust-gas receiver are located on one side of the cylinder, thus giving better accessibility. This scheme is particularly suitable for double-acting engines, since with them the operation of the exhaust valves, Fig. 5–2, for the lower combustion space, becomes very complicated. When used for double-acting engines, Fig. 5–5, the scheme is improved by the introduction of rotary exhaust valves *r*, *r*. During the release of the exhaust gases, valve *r* is open but is

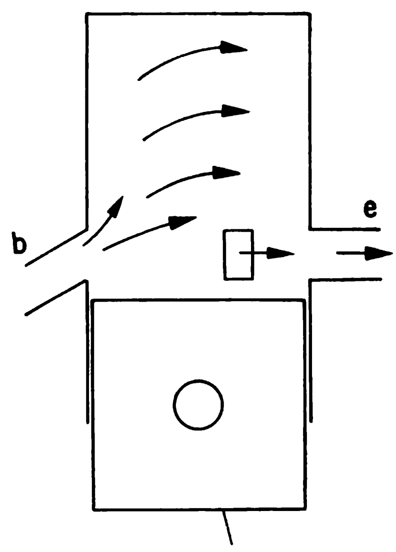


Figure 5–3. Direct or cross-flow scavenging.

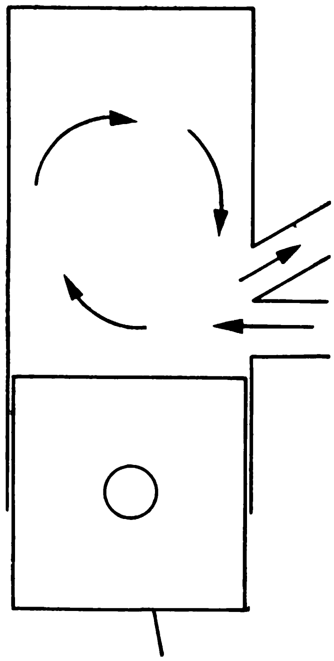


Figure 5–4. Return or loop scavenging.

being closed when the piston on the return stroke covers the scavenge ports. By this arrangement the escape of air charge is eliminated during the beginning of the compression stroke, when the exhaust ports are not yet covered. Sometimes, after the exhaust ports are covered by the piston, the rotary valve is opened, getting it ready for the next cycle. As can be seen from Fig. 5–5, the length of the piston is exactly equal to the length of the stroke in order to control the exhaust and scavenge events alternatingly by the upper and lower edges of the piston.

Opposed-piston scheme is shown in Fig. 5–6. The lower piston controls the exhaust ports, the upper one the scavenge ports. In order to obtain the necessary preliminary release of the exhaust gases, or an uncovering of the exhaust ports *e* ahead of the scavenge ports *s*, the crank of the lower crankshaft is advanced in respect to the crank of the upper crankshaft, or leads the upper crank by some 10 to 15 degrees. In this way the exhaust ports are opened first, Fig. 5–6a; when the pressure is sufficiently lowered, the scavenge ports are uncovered, Fig. 5–6b, and scavenging begins to take place. After the exhaust ports are closed, additional admission of air takes place, Fig. 5–6c, until the scavenge ports are also covered and compression of the air charge takes place, Fig. 5–6d. Slightly before the pistons reach the point at which they are closest together, fuel is injected, ignited, and burns while the expansion stroke starts, Fig. 5–6e. The power delivered by the upper pistons to the upper crankshaft is transmitted to the lower main crankshaft by means of an intermediate vertical shaft and two pairs of bevel gears. The advantages of this scheme are:

1. Very efficient scavenging of the cylinder and hence greater power is developed.
2. Absence of valves and valve-operating gears.
3. Absence of cylinder heads which are complicated castings and a source of trouble in engine operation.
4. Good accessibility for the inspection and repair of most parts, except of the lower crankshaft.

The two scavenge schemes, Figs. 5–2 and 5–6, are also classified as *uniflow* scavenging. In both cases the exhaust gases and scavenging air are flowing in the same direction, with less chance for formation of turbulences which are unavoidable with cross- and return-flow scavenging.

Most two-stroke diesel engines in the United States Navy operate on one of the two uniflow-scavenge schemes.

5–6. Problems. 1. Determine the compression ratio of a diesel engine with a 5¾-in. bore and a 6-in. stroke if the volume of the combustion space is 11.13 cu.in. *Ans.* 15:1.

2. Determine the compression ratio of the engine of problem 1, if, due to wear of the bearings, the piston at the top center is 0.024 in. lower than in a new engine. *Ans.* 14.26:1.

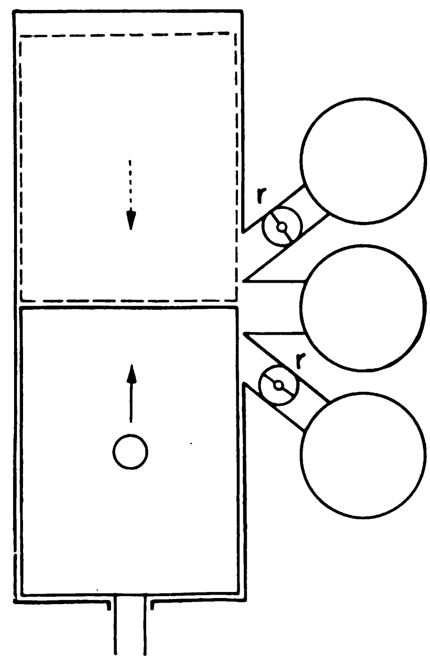


Figure 5–5. Return scavenging in a double-acting engine.

5–7. Questions. 1. Enumerate the main events of a four-stroke-cycle engine operation.

2. Explain the objects of compressing the air charge in a diesel engine.

3. What is meant by *compression ratio* in an internal-combustion engine?

4. What are the methods of burning the fuel in an engine cylinder?

5. What method of combustion is used in present diesel engines?

6. Explain the difference between two-stroke- and four-stroke-cycle operation.

7. Enumerate the methods of scavenging used in two-stroke engines.

8. What methods of scavenging are used in most two-stroke naval diesel engines?

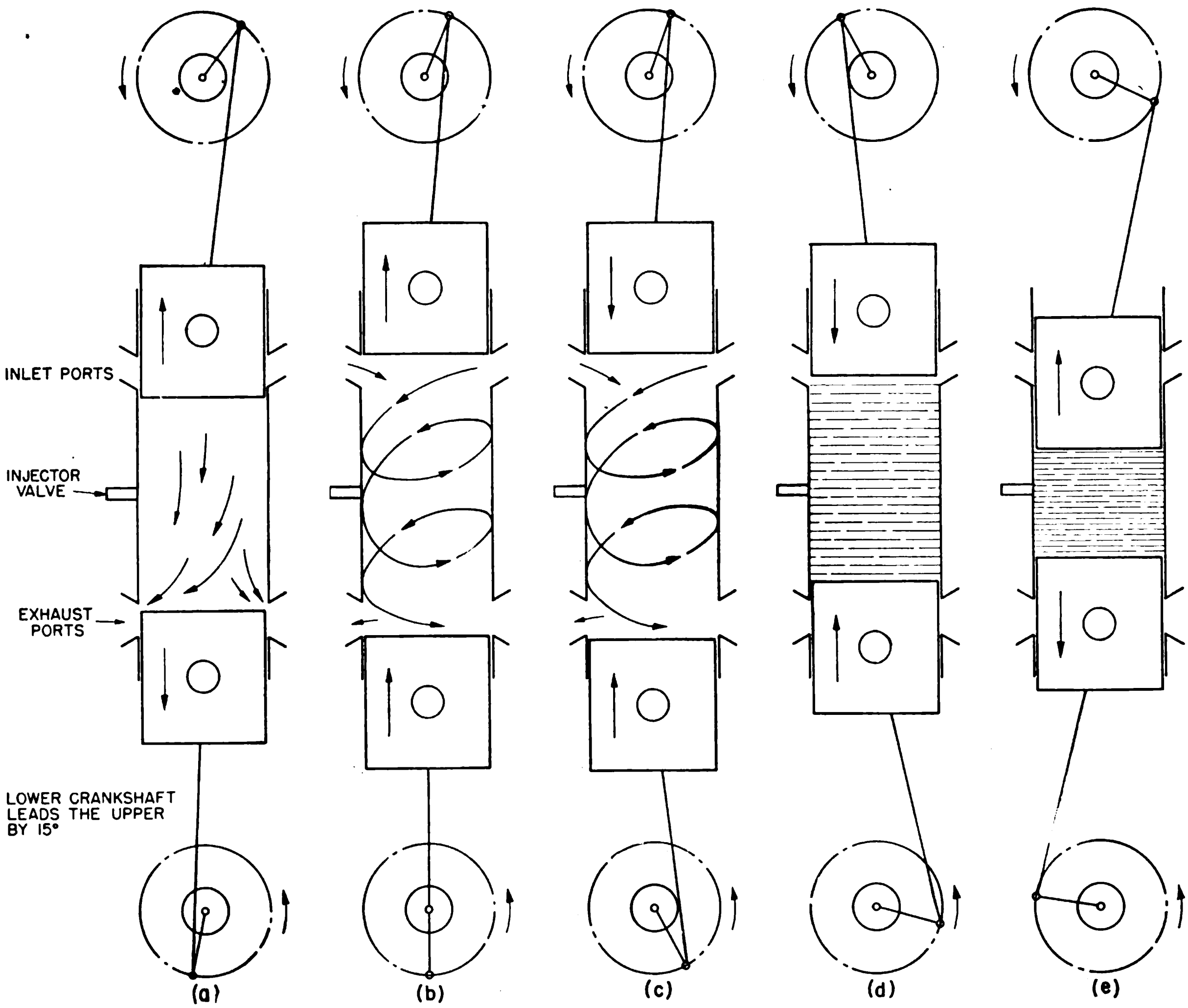


Figure 5–6. Opposed-piston engine operation.

CHAPTER 6

ENGINE PERFORMANCE

6–1. Pressure-volume diagrams. The understanding of what occurs in an engine is made clearer by a graphic representation of pressures and volumes, called a *pressure-volume* or *p-V* diagram. In such a diagram the volumes are represented by horizontal distances or *abscissas* measured by an elected scale, and the pressures by vertical distances or *ordinates* measured by another scale. These distances are measured from two basic lines called *zero-volume* and *zero-pressure* lines.

Example 6–1. Show in a *p-V* diagram the change that 3.5 cu. ft. of air, having a pressure of 55 psia., will undergo if the air is heated and expands to 9.5 cu. ft. keeping the same pressure.
Suppose the scale of volumes is selected as 1 in. = 4 cu. ft. and the scale of pressure is 1 in. = 40 psi. Then the initial condition of the air will be represented by a point which is $3.5 \div 4 = 7/8$ in. to the right from the zero-volume line *O-P*, Fig. 6–1, and $55 \div 40 = 1\frac{3}{8}$ in. above the zero-pressure line *O-V*, point *1* on Fig. 6–1.
The final condition of the air will be shown by a point with an abscissa $9.5 \div 4 = 2\frac{3}{8}$ in. from the line *O-P*, point 2, and the change of state will be represented by the horizontal line *1–2*.

Work. A volume of air or gas contained in a cylinder, with one end having a fixed cover such as the cylinder head of an engine, and the other end closed by a movable cover, as an engine piston, can be presented as the product of the piston area A, times the distance l, from the fixed cover to the moving piston. The change of volume, as in Example 6–1, can be imagined as a product of the piston area A, times the difference between the first and second positions of the piston or travel of the piston $(l_2 - l_1)$. On the other hand, the area A of the piston, multiplied by the pressure p is equal to the total force F acting on the piston, $A \times p = F$, which is only another form of expression (2–3). As stated before, expression (2–5), the product of a force, times the distance traveled by it, represents the work W done by that force; therefore, it may be stated also that the product of the pressure, times the change of volume, represents work done

$$\text{Work} = \text{pressure} \times \text{change of volume}$$

or

$$W = p(V_2 - V_1) \qquad (6\text{–}1)$$

Naturally, pressure and volume must be referred to the same units: if the pressure is in psi., the volumes must be in cu. in. and the expression (6–1) will give the work in in.-lbs.; if the volumes are in cu. ft., the pressure must be in psf. and the work found will be in ft.-lbs.

Example 6–2. Find the work in ft.-lbs. done in Example 6–1.
The pressure 55 psi. $= 55 \times 144 = 7{,}920$ psf.; and the work done, by expression (6–1):

$$W = 7{,}920\,(9.5 - 3.5) = 47{,}520 \text{ ft.-lbs.}$$

It can readily be seen that the product of pressure times the change of volume represents an area, such as the rectangle *1-2-3-4-1*, shown in Fig. 6–1. It can also be seen, that the pressure does not have to be constant; if the pressure changes with a change of volume, curve *1–2* in Fig. 6–2, the area under the curve *1–2* limited by two ordinates *1–4* and 2–3 drawn

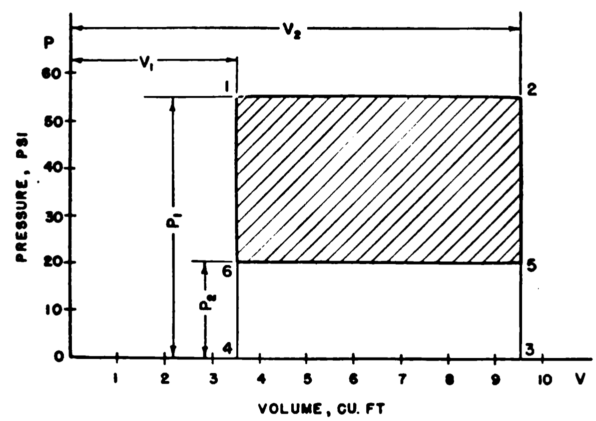


Figure 6–1. Pressure-volume diagram.

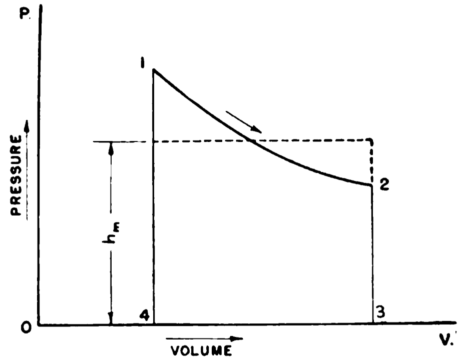


Figure 6–2. Change of state.

through the end points of the curve perpendicularly to the line of zero-pressure and by the line of zero-pressure, still represents work done during the change from *1* to 2. In order to find the numerical value of the work, the mean pressure p_m must first be found. This can be done either by measuring the area *1–2–3–4–1* in sq. in. and dividing it by the length, distance *4*–3, or by dividing the area in a number of vertical strips of equal width, Fig. 6–3, and finding their mean

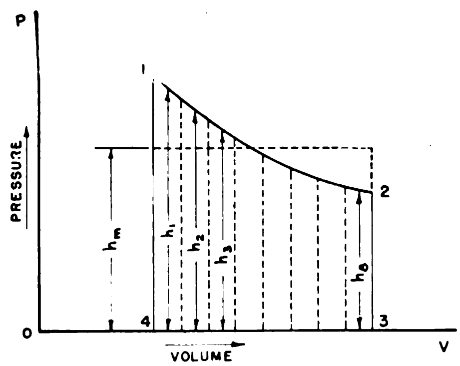


Figure 6–3. Finding of mean height.

height by taking each height h_1, h_2 . . . , adding them together, and dividing by the number of strips.

When the mean height h_m, in., is found, the mean pressure p_m is calculated by multiplying h_m by the number of psi. represented by 1 in. on the scale of pressures. If 1 in. $= m$ psi., then the mean pressure is found as

$$p_m = h_m \times m \qquad (6\text{–}2)$$

After p_m is found, the work done is computed using expression (6–1).

If the final volume V_2 is greater than the initial volume V_1, expression (6–1) gives a positive value, the work is *positive* or *performed by* the gas, such as during the expansion stroke of a diesel engine. If V_2 is smaller than V_1, the difference $V_2 - V_1$ is negative, the work by expression (6–1) is *negative* or *done on* the gas, such as during the compression stroke of a diesel engine.

Work of a cycle. If a certain amount of gas undergoes various changes of volume and pressure, but at the end its volume and pressure, respectively, will be the same as at the start, and this procedure is repeated over and over, it is said that the gas is *undergoing cyclic changes*. The work of a cyclic change or *cycle* is equal to the difference between the positive work done *by* the gas during the increase of its volume, its expansion, and the negative work done *on* the gas during the decrease of its volume, its compression. The simplest case is illustrated by Fig. 6–1. Air, whose conditions are given by point *1*, expands first at constant pressure to point 2, then the pressure drops from p_1 to p_2 and the air is compressed at constant pressure to point *6* and finally the pressure returns to the initial valve, point *1*. The positive work done by the air from point *1* to point 2 was determined as area *1-2-3-4-1*. During the changes from 2 to *5* and from *6* to *1* no positive work is done or negative work absorbed since there is no change of volume. To clarify the nature of the change of pressure at constant volume, one can imagine that the change is due to cooling the air when the pressure drops, and heating it when the pressure increases. The work from point *5* to point *6* is represented by the rectangle *5-6-4-3-5* and since the volume decreases, is negative, or done on the air. As a result, the work of the cycle is the difference between the positive and negative works and is represented by the rectangle *1-2-5-6-1* outlined by the consecutive changes of state.

On the basis of Fig. 6–1, the work of the cycle can be found as follows: using the expression (6–1) first for the positive work, gives

$$W_1 = p_1 (V_2 - V_1)$$

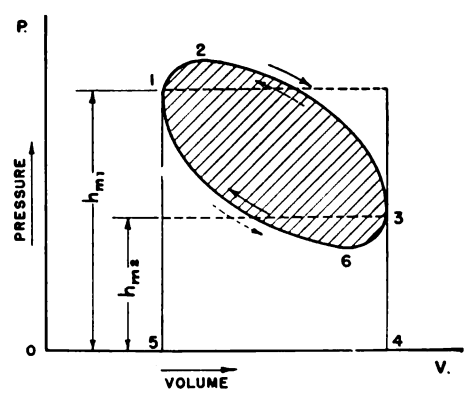


Figure 6–4. Work of a cycle.

and for the negative work

$$W_2 = p_2(V_2 - V_1)$$

subtracting W_2 from W_1 gives the work of the cycle

$$W_1 - W_2 = {}_1W_2 = p_1(V_2 - V_1) - p_2(V_2 - V_1)$$
$$= (p_1 - p_2)(V_2 - V_1) \qquad (6\text{–}3)$$

Evidently, the same as when dealing with a single change of state, such as *1*-2, Fig. 6–2, the outline does not have to be presented by straight lines. Thus, the work of the cycle of Fig. 6–4 will be equal to the positive work *1*-2-3-*4*-*5*-*1*, minus the negative work 3-*6*-*1*-*5*-*4*-3, or to the area *1*-2-3-*6*-*1*.

Numerically, the work of a cycle as shown in Fig. 6–4, can be calculated by finding first the mean height h_{m1} for the change *1*-2-3 and next the mean height h_{m2} for the return charge 3-*6*-*1* by one of the methods explained before. Knowing the scale of pressure of Fig. 6–4, the mean heights h_{m1} and h_{m2} can be converted to mean pressures p_{m1} and p_{m2} using equation (6–2). These values of p_{m1} and p_{m2} can be used now in expression (6–3) to find the work done by the cycle.

The change of state does not always occur in the direction indicated by the solid arrows in Fig. 6–4. If it goes in the opposite direction, as indicated by the broken-line arrows, the positive work will be represented by the area *1*-*6*-3-*4*-*5*-*1*, the negative work by 3-2-*1*-*5*-*4*-3, and their combined result will be *negative* work represented by the area limited by the curve 3-2-*1*-*6*-3. Such a diagram means, that as the result of the cycle, work must be *done* on the gas. This is the case of a compressor cycle such as is used for compressing air or in a refrigeration installation.

6–2. Indicator cards. The investigation of the various events in a diesel engine and of its performance in general is greatly helped by the *p-V* diagram. Fig. 6–5 shows the diagram of a two-stroke compression-ignition engine; curve *1*-2 shows the gradual rise of pressure during the compression stroke; the line 2-3 indicates the rapid pressure rise during the first part of combustion; 3-*4* is the second part of combustion at constant pressure; *4*-*5* is the expansion of the burned gases; release occurs at point *5* with the resulting drop of pressure and following scavenging, lower part of curve *5*-*6* and part *6*-*1* of the return stroke.

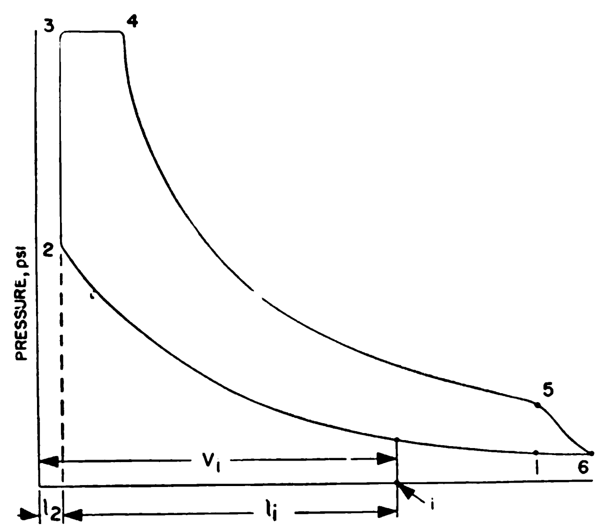


Figure 6–5. Indicator diagram of a two-stroke cycle engine.

As already mentioned, the gas volume in an engine cylinder can be presented as $V_n = A \times l_n$, where A is the constant piston area and l_n changes from l_1 at *b.c.* to l_2 at *t.c.*, Fig. 6–6. From Fig. 6–5, $l_1 - l_2 = l$, where l is the total piston stroke. The volume at any piston position can be presented as $A(l_2 + l_i)$.

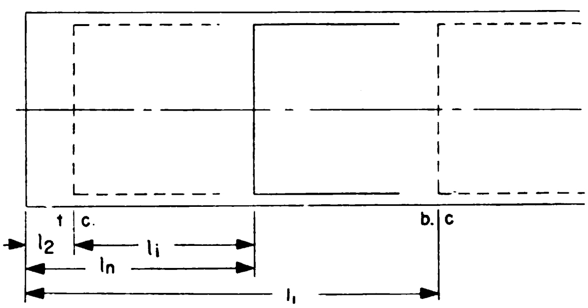


Figure 6–6.
Piston positions in connection with an indicator diagram.

Based on this fact, a *p-V* diagram may be obtained from an engine in operation by means of an instrument called an *indicator*. Fundamentally, a cylinder-pressure indicator consists of a cylinder, one end of which is connected to the combustion space of the engine, and the other closed by a gastight but free-moving piston. The piston is fastened to a spring which is compressed or stressed in tension, depending upon the construction, when the piston is moved by the gas pressure. This motion of the piston is traced on a piece of paper fastened to a drum. At the same time, the drum revolves back and forth imparting to the paper attached to it a motion proportional to the piston motion. Such a diagram is called an *indicator diagram* or *indicator card*. In conformity with this designation, a *p-V* diagram presenting the events in an engine cylinder, even if it is not drawn by an indicator but from theoretical computations, as in Fig. 6–5, is also called an indicator diagram. This is more correctly designated as a *theoretical indicator diagram*.

Fig. 6–7 gives the reproduction of an actual indicator diagram taken of a four-stroke diesel engine:

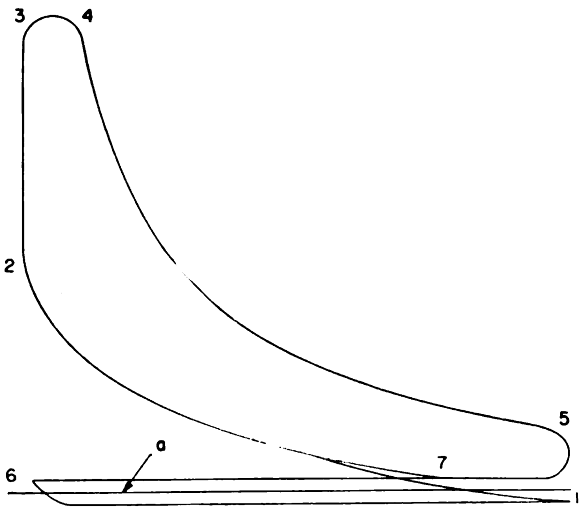


Figure 6–7. Indicator diagram of a four-stroke engine.

1-2 is the compression stroke; 2-3-*4* is the combustion and *4*-*5* the expansion; *5*-*6* is the exhaust stroke with the pressure slightly above the atmospheric line *a* which is also drawn by the indicator pencil; *6*-*1* is the suction stroke.

Fig. 6–8 is an indicator of a two-stroke engine, the events being the same as those referred to for the theoretical diagram, Fig. 6–5.

An indicator card serves two purposes. First, it shows what happens in an engine cylinder and thus gives an indication of whether the engine operates as it should or what adjustments must be made to improve its performance. Second, it affords the means for computing the power developed by the engine. The procedure is based on the fact that the area enclosed by the contour of an indicator diagram represents the work done during one cycle, as explained in Sec. 6–1.

Mean indicated pressure is the term applied to a constant pressure which would develop the same power as the variable pressure recorded on the indicator card. Mean indicated pressure is found as follows: the area of the diagram, such as *1*-2-3-*4*-*5*-*1*, Fig. 6–8, is measured and found to equal *a* sq. in.; this value of *a* is divided by the length *l*, in., of the card and gives the mean height h_m; the mean indicated pressure p_m, psi., is found by multiplying h_m by the scale of the ordinates m_1 called in this case *spring scale* or *spring number* and showing how many psi. are represented by 1 in. Because the mean pressure found in this manner would do the same work as the variable pressure of the diagram, i.e., have the same *effect*, the mean indicated pressure is often referred to as the *indicated mean effective pressure* or simply *mean indicated pressure.*

The indicator card of a four-stroke engine may be divided into two areas: area *1*-2-3-*4*-*5*-*1*, which represents the positive work developed, and area *5*-*6*-*1*-*5*, which represents the negative work of removing the burned gases and filling the cylinder with a fresh charge of air. The vertical line *5*-*1* does not appear on a diagram taken by an indicator but represents the drop of pressure which would occur, if there were no resistance to the gas flow through the exhaust valves. Their combined result is the difference between these two areas, and since the area *5*-7-*1*-*5* is part of both the positive and negative area, it cancels out, and the net work is represented by the difference of the areas of the loop 7-2-3-*4*-*5*-7 and loop 7-*6*-*1*-7. When the area is measured by a planimeter and the diagram contour is followed in one sweep *1*-2-3-*4*-*5*-7-*6*-*1*, the subtraction is done by the instrument automatically, and its reading gives the net area *a*. When the indicator diagram is taken with a stiff spring, which has a small scale, the exhaust and suction line may merge with the atmospheric line and the negative loop area practically disappears.

Indicated horsepower. The piston area *A*, sq. in., multiplied by the mean indicated pressure p_m, psi., gives a force *F*, lb., which, moving through a distance equal to the piston stroke l_1 in., does the work $W = F \times l = A \times p_m \times l$, in.-lb., during each cycle. If this work is multiplied by the number of power strokes per min., which for a two-stroke engine is equal to the number of revolutions per minute *n*, rpm., the product will give the power *P*, in.-lbs. per

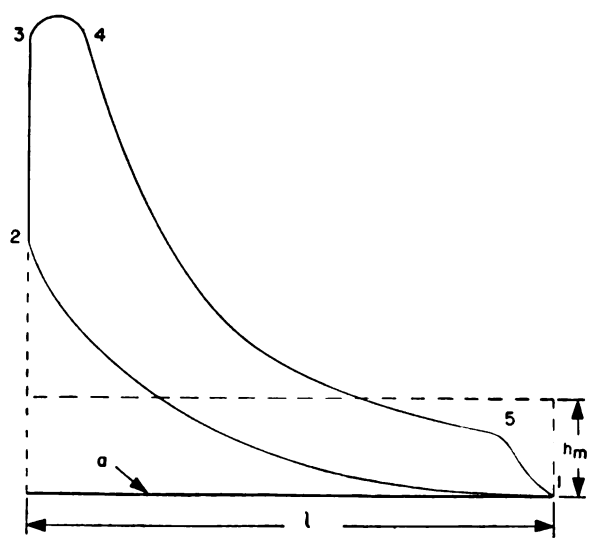


Figure 6–8. Indicator diagram of a stroke engine.

min. Dividing P by 12 will convert it to ft.-lbs. per min., and dividing by 33,000, the equivalent of 1 hp., will give the *indicated horsepower* (ihp.) developed by one cylinder:

$$\text{ihp.} = Ap_m\, ln \div (12 \times 33{,}000)$$

Changing the designations p_m to p and A to a and rearranging the order of the multipliers gives the standard expression:

$$\text{ihp.} = \text{plan} \div (33{,}000 \times 12) \qquad (6\text{–}4)$$

In a four-stroke engine the power occurs for every two revolutions and n in expression (6–4) must be divided by 2.

The indicated horsepower, ihp., developed by a multicylinder engine evidently is equal to the sum of ihp. for each cylinder; if it is assumed that the mean effective pressure p is the same in all cylinders, it is equal to the ihp. of one cylinder multiplied by the number of cylinders. In a double-acting engine the piston area upon which the gas acts from the underside is reduced by the area of the piston rod, and the ihp. of the engine is equal to the sum of ihp. developed by each of the two combustion spaces times the number of cylinders.

Example 6–3. Determine the horsepower developed by a 6-cylinder, four-stroke, 5¾ in.×6-in. diesel engine operating under full load at 1,200 rpm. The indicator card taken with a 320-psi./in. spring has an area of 0.523 sq.in., and its length is 1¾ in.

The mean indicated pressure, by expression (6–2), Sec. 6–1,

$$p_m = \frac{0.523}{1.75} \times 320 = 95.5 \text{ psi.}$$

The piston area $a = \pi \times \frac{(5.75)^2}{4} = 25.97$ sq. in.

Using expression (6–4) with $n = 1{,}200 \div 2 = 600$ gives for one cylinder:

$$\text{ihp.} = 95.5 \times 6 \times 25.97 \times 600 \div (33{,}000 \times 12) = 22.5 \text{ ihp.}$$

and the 6-cylinder engine will develop $22.5 \times 6 = 135.0$ ihp.

Pressure-time diagrams. The pressure-volume indicator diagram gives the cylinder events as referred to the piston stroke. As will be shown in the following chapter, the piston motion is not uniform, being slow near the dead center, with the speed increasing and reaching a maximum about mid-stroke, after which it decreases to zero at the other dead center. Due to such conditions, a small motion of the piston near one of the dead centers corresponds to a considerably larger time element than a similar motion farther away from one of the dead centers. However, in all engines the important events of ignition and combustion, and in two-stroke engines the scavenging, takes place near the dead center. Hence, these events are not shown clearly enough on a regular indicator card. These events can be investigated much better if the horizontal distances on the card are proportional to time. Such an indicator is built with the paper drum being revolved by a clock mechanism. Since the rotation of the crankshaft is rather uniform, the paper drum motion can be obtained also from the rotation of the shaft with a suitable speed reduction. The indicator card obtained by such an instrument will have the abscissas proportional to time or to crank-angle degrees, which is practically the same. A sample of such a diagram is shown in Fig. 6–9. It shows clearly the decreasing rate of pressure rise toward the end of the compression stroke, the ignition, and at first a slow combustion and soon after dead center an accelerated combustion, then again retarded combustion and expansion.

A pressure-time or pressure-crank angle diagram is useful for investigation of an engine's performance but cannot be used for hp.-determination, unless redrawn to a pressure-piston stroke or p-V type.

6–3. Engine efficiencies. Efficiency in general may be defined as the ratio of the output to the input. In the case of a heat engine, the output is the work delivered and the input is the heat supplied in the fuel. Efficiencies are expressed as decimal fractions and are always less than unity. Another way to express efficiency is by percentage, in which case efficiency is less than 100 per cent. The difference between the

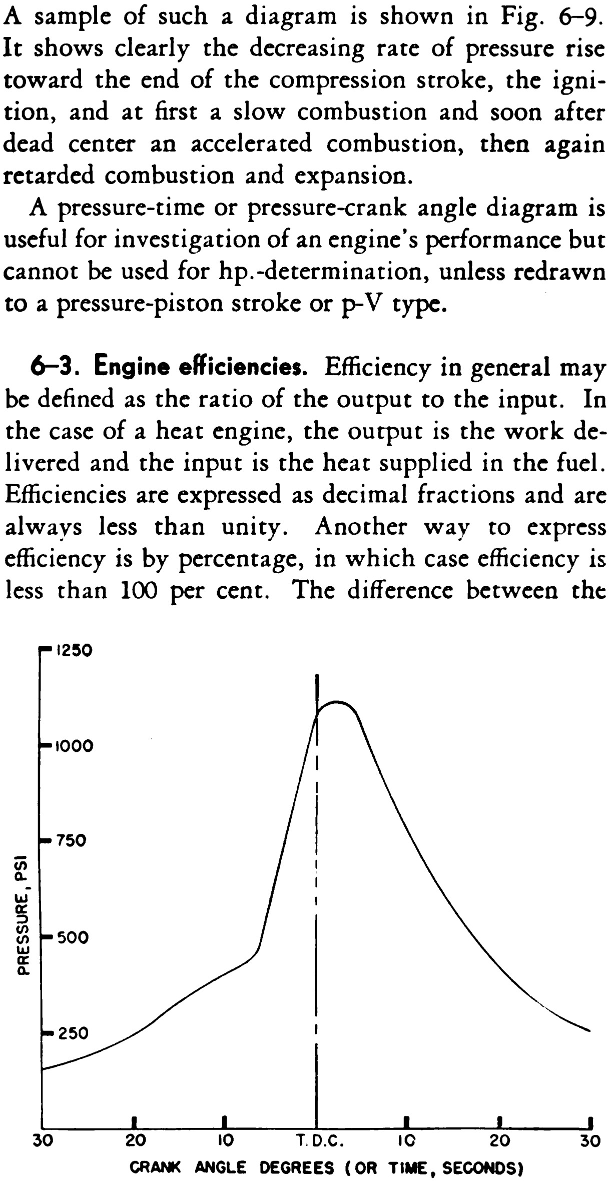


Figure 6–9. Open indicator diagram.

efficiency fraction and unity or percentage efficiency and 100 is the loss incurred during the process discussed.

When dealing with diesel engines a number of efficiencies may be discussed, the principal ones being:

1. Mechanical efficiency.
2. Thermal efficiency.
3. Over-all efficiency.
4. Volumetric or charge efficiency.

Mechanical efficiency. The horsepower computed by expression (6–4) is indicated hp. or power developed by the gases inside of the engine cylinder. This power is transmitted by the piston and connecting rod to the crankshaft. During this transmission several *mechanical losses* occur due to friction between the piston and piston rings and the cylinder walls, friction in the various bearings, power absorbed by the valve and injection mechanisms and by various auxiliaries, such as the lubricating-oil and water-circulating pumps and the scavenge and supercharge blowers. As a result, the power delivered to the crankshaft and available for doing useful work is appreciably (15 to 30 per cent) smaller than the indicated power. The available power is called either shaft horsepower or *brake horsepower* (bhp.). The term brake-hp. is derived from the usual method of its determination: the engine is operated with the power measured and absorbed by a brake. The ratio of the bhp. to ihp. is called mechanical efficiency (mech. eff.):

$$\text{mech. eff.} = \text{bhp.} \div \text{ihp.} \qquad (6\text{–}5)$$

The difference between the ihp. and bhp. is called friction hp. (fhp.):

$$\text{fhp.} = \text{ihp.} - \text{bhp.} \qquad (6\text{–}6)$$

Example 6–4. Find the mech. eff. and fhp. of the engine of Example 6–1 if its load under the given conditions was 110 bhp.:

By expression (6–5)

mech. eff. = 110 ÷ 135.0 = 0.814 or 81.4 per cent

By expression (6–6)

fhp. = 135.0 − 110 = 25.0 hp.

Actually fhp. represents not only work lost due to friction but to other losses too, as explained above. Also it is well to recall, that friction loss is really not a loss of energy, as energy cannot be lost or destroyed, but is only a transformation of work into heat. By the time this heat is dissipated to the atmosphere, it has become useless or unavailable for the purpose for which the engine is operated.

Mechanical efficiency depends upon the construction of the engine, workmanship of the various engine parts, and operating conditions of the engine, such as the amount of load carried, temperature of the cooling water, etc.

Two-stroke engines, generally speaking, have a slightly lower mechanical efficiency, due to the power absorbed by the scavenge pump; an engine having a better designed lubricating system, better workmanship, better alignment of various parts, has a smaller fhp. and a higher mechanical efficiency. An engine operating under part-load has a lower mechanical efficiency because most mechanical losses are almost independent of the load, and therefore, when the load decreases, ihp. decreases relatively less than bhp., which results in a lower mechanical efficiency according to expression (6–5). Mechanical efficiency becomes zero when the engine runs at no load because then bhp. = 0 but ihp. is not zero, in fact, according to expression (6–6) in this case ihp. = fhp.

Example 6–5. Find the mech. eff. at ¾-load of the engine discussed in Examples 6–3 and 6–4.

¾ load means 110 × 0.75 = 82.5 bhp. Assuming that the fhp. does not change with load, fhp. = 25.0 hp. The ihp. is, from expression (6–6):

ihp. = 82.5 + 25.0 = 107.5 hp.

and by expression (6–5):

mech. eff. = 82.5 ÷ 107.5 = 0.767 or 76.7 per cent, appreciably lower than 81.1 per cent at full load.

Brake mean effective pressure, designated as *bmep.*, is a fictitious but useful concept; bmep. is obtained if expression (6–4) is applied to bhp. instead of ihp. and the mean pressure p designated by bmep. is found as:

$$\text{bmep.} = 33{,}000 \times 12 \times \text{bhp.} \div \text{lan}. \qquad (6\text{–}7)$$

From the relations between bmep., bhp., ihp. and mech. eff., designating indicated mean effective pressure in expression (6–4) by imep., one can also write:

$$\text{bmep.} = \text{imep.} \times \text{mech. eff.} \qquad (6\text{–}8)$$

Bmep. gives an indication of what duty an engine carries, and what its output is for its displacement. An engine which develops continuously a higher bmep. carries a heavier duty, and develops a greater hp. per lb. of its weight than an engine with a lower bmep., all other things being equal. For a given engine, bmep. changes in direct proportion with the load.

The following figures will give an idea of bmep. found in present engines: in four-stroke, medium-speed, continuous-operation engines, bmep. = 75 to 85 psi.; in four-stroke, high-output, unsupercharged engines, bmep. = 95 to 100 psi.; in supercharged engines, bmep. of 110 to 125 psi. are found and for intermittent operation may be raised as high as 140 psi. In two-stroke engines, depending upon the method of scavenging, bmep. varies from 30 to 65 psi.; in heavy-duty high-output two-stroke engines, such as those used in the Navy, bmep. may be found as high as 90 to 100 psi.

Example 6–6. Find the bmep. at full and ¾-load for the engine discussed in Examples 6–3 and 6–4.

By expression (6–7):

bmep. = 33,000 × 12 × 110 ÷ (6 × 25.97 × 600) × 6 = 77.7 psi.

or by expression (6–8), using from Example 6–3 p_m = imep. = 95.8 and from Example 6–4 a mech. eff. = 0.811:

bmep. = 95.5 × 81.4 = 77.7 psi.

Since bmep. is proportional to load, therefore, at ¾ load

bmep. = 77.7 × 0.75 = 58.3 psi.

Thermal efficiency is the *ratio* of the *output* or work done by the working substance in the cylinder in a given time *to* the *input* or heat energy of the fuel supplied during the same time. Since the work done by the gases in the cylinder is called *indicated* work, the thermal efficiency thus determined is often called *indicated thermal efficiency*. The expression for computing it is

$$\text{ind. therm. eff.} = \frac{\text{ind. work}}{\text{heat input}} \qquad (6\text{–}9)$$

Example 6–7. Find the thermal efficiency of the engine discussed in Example 6–3 if its fuel consumption determined as an average of three tests was 46.25 lbs. per hr. The heat value of the fuel was 18,200 Btu. per lb.

In any ratio, such as expression (6–9), both the numerator and denominator must be expressed in the same units. In this case it is more natural to use Btu. as used in the denominator. A convenient relation was given in Sec. 2–8: 1 hp.-hr. = 2,544 Btu. Therefore, the work done per hr., when 135.0 ihp. are developed, 135.0 × 2,544 = 343,440 Btu. The heat input for the same time is 46.25 × 18,200 = 842,000 Btu., and by expression (6–9):

ind. therm. eff. = 343,400 ÷ 842,000 = 0.408 or 40.8 per cent.

Over-all efficiency is a ratio similar to ind. therm. eff., only using as an output of the engine its useful or shaft work expressed in brake-hp. Therefore, over-all efficiency is also often called brake thermal efficiency. Thus the expression for computing it is:

$$\text{over-all eff.} = \text{br. therm. eff.} = \frac{\text{shaft work}}{\text{heat input}} \qquad (6\text{–}10)$$

Example 6–8. Find the over-all efficiency of the engine discussed in Examples 6–3, 6–4, and 6–7. The 100-bhp. load as equivalent to a useful work per hr. of 110 × 2,544 = 279,840 Btu. The heat input was found in Example 6–7 as 842,000 Btu.

Therefore, by expression (6–10)

Over-all eff. = 279,840 ÷ 842,000 = 0.332 or 33.2 per cent.

Volumetric efficiency of a four-stroke engine is called the *ratio of* the *volume of* the *air admitted* to the engine cylinder during the suction stroke, referred to normal pressure and temperature conditions, *to* the *suction volume* of the piston. The latter volume is equal to the piston area times the piston stroke and is called piston displacement. Thus:

$$\text{vol. eff.} = \frac{\text{volume admitted}}{\text{piston displacement}} \qquad (6\text{–}11)$$

Volumetric efficiency shows the amount of air admitted as compared with the maximum possible amount of air represented by the piston displacement.

A given volume of gas having a certain pressure and temperature is equivalent to a certain weight as computed from expression (2–7). Therefore, volumetric efficiency is also often formulated as the ratio of the weight of air admitted in the cylinder during the suction stroke to the weight, at normal temperature and pressure, of a volume of air equal to the piston displacement. With this definition, volumetric efficiency is also called *charge efficiency*. The actual amount of air admitted is smaller than the amount corresponding to the piston displacement for two main reasons: (1) the temperature of the air in the cylinder is higher, and (2) the pressure is lower than the normal or standard values. The temperature of the fresh air drawn into the cylinder is raised, first, due to mixing with the hot gases left in the compression space from the previous cycle, and second, due to contact with the hot cylinder walls, cylinder head, and piston crown. The pressure in the cylinder at the end of the suction stroke is lower than the outside, atmospheric air, because of the resistance to air flow in the air intake system, particularly through the intake valve.

The lower the volumetric efficiency, the less the amount of air admitted. Consequently less fuel can be burned, and less power is developed by the engine. Taking air from a point in the engine room where the air is cooler, and having all valves properly timed, will increase the volumetric efficiency and through it the maximum power which the engine can develop. In present diesel engines, volumetric efficiency is on the order of 80 to 87 per cent, decreasing chiefly with an increase of the engine speed.

For two-stroke engines, the concept of volumetric efficiency does not apply. Instead, the term *scavenge efficiency* is used, which shows how thoroughly the burned gases are removed and the cylinder filled with fresh air. Similarly in four-stroke engines it is desirable to take the air to the scavenge blower at a point where the air is cooler. Scavenge efficiency depends greatly upon the arrangement of the exhaust, scavenge air ports, and valves. The best scavenge efficiency is obtained by uniflow scavenging, as explained in Sec. 5–5.

6–4. Combustion and ignition delay. Regardless of how finely atomized the fuel injected into the combustion space of the cylinder filled with hot air is, it takes some time before the relatively cold fuel spray becomes heated and vaporized so as to ignite and start burning. This time element is rather small when expressed as a fraction of a second, but quite appreciable when referred to the number of degrees which the crank travels between the moment when the fuel is introduced into the cylinder and when the first particles of it are ignited. This time element is called ignition delay or ignition lag and amounts to several degrees of crank travel.

After ignition has started, that fuel will burn, which by this time, is already introduced in the cylinder. This combustion usually is accompanied by a rather quick pressure rise. In the meantime, the pump continues to deliver fuel, and during the third period

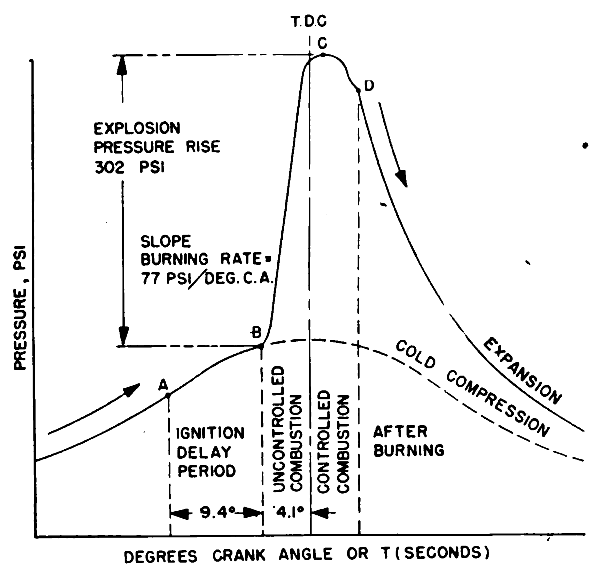


Figure 6–10. Pressure-time diagram of combustion process.

of combustion the fuel burns more or less as it is introduced. However, since the supply of oxygen in the air charge gradually is being used up by the combustion, the fuel particles introduced toward the end of the injection have more difficulty meeting the necessary particles of oxygen; combustion is consequently retarded, and when injection is terminated, some unburned fuel is still present in the cylinder and continues to burn. The piston by this time has moved away from the dead center and its speed increases; therefore, the pressure begins to fall in spite of additional heat being developed by the rest of the fuel.

The whole procedure can be illustrated by the pressure-crank angle diagram, Fig. 6–10. Point *A* is the start of the injection, point *B* is where ignition occurs; and period *A-B* is *ignition delay;* up to point *C* the pressure rises very fast; period *B-C* corresponds to burning of the fuel introduced up to this point, and represents the first or *uncontrolled* combustion; from *C* to *D* the fuel burns more or less as it is introduced, giving the second or *controlled* combustion; from point *D* the fuel burns with the pressure dropping—this is *after-burning*. After-burning may continue through a considerable distance of the expansion stroke.

The highest thermal efficiency is obtained from the fuel which burns at the highest compression ratio, at top center. In practice, burning of the fuel must start before top center and be completed after top center. The shorter the period of combustion, the higher the thermal efficiency, and the lower the fuel consumption. However, an excessively short burning period requires a fast pressure rise and produces high maximum pressures. This is undesirable as far as the quietness of engine operation, pressures, and stresses in various engine parts are concerned. However, afterburning is an undesirable phenomenon and should be reduced to a minimum.

6–5. Turbulence. In order to obtain efficient, smokeless combustion, the fuel injected into the cylinder must be broken up in very fine particles, be well atomized, and the fuel particles must be distributed uniformly through the whole combustion space. In air-injection engines, the distribution of the fuel is accomplished through a thorough mixing of the injection air, carrying the atomized fuel, with the air in the cylinder. In mechanical- or solid-injection engines, distribution is accomplished by using nozzle tips with several holes and by directing the fuel sprays so as to reach the desired portions of the combustion space, or by using pintle-type nozzles with a cone-shaped spray. In larger engines, better fuel distribution is obtained by using two or more separate fuel nozzles, each having several holes or fan-shaped sprays. However, distribution of the fuel by separate sprays usually is not sufficient. Distribution of the fuel in the air charge is improved by stirring up the air in the combustion space, by creating air turbulences, and thus mixing air having too much fuel with air which does not have any fuel.

While theoretically 1 lb. of air is sufficient to burn completely 0.065 lb. of fuel oil, actually, not all of the oxygen of the air will be reached by the fuel particles. Hence, only a smaller amount of fuel, on the average not over 0.052 or even 0.043 lb., can be burned efficiently with 1 lb. of air in the combustion chamber.

Turbulences in the air charge help to reduce the amount of air not reached by the fuel particles, and thus help to increase the power output of the engine.

Turbulences may be created by various means, including special shapes of the piston crown or of the entire combustion chamber. Fig. 6–11 shows examples of different turbulent heads. In Fig. 6–11a turbulences are created by a restriction through which the air has to pass when the piston moves upward; the air velocity at the restriction is several times higher than before and after it, and the change of velocity creates a turbulent flow into which the fuel is injected from the fuel nozzle. Fig. 6–11b shows a Ricardo-Comet head used in Waukesha diesel engines; here turbulences are created not only by the restriction, but also by forcing the air to travel on a circular path. Fig. 6–11c shows a turbulent head used in Hercules diesel engines which is similar to the Ricardo head. However, it has an additional feature: when the piston approaches the dead center, it begins to cover partially the air passage between the cylinder and the turbulence chamber. This increases the air velocity in the passage, and thus makes more turbulent the flow of air into which the fuel is injected from the nozzle *f*.

Turbulence in two-stroke engines is created by making the scavenge-air ports tangential, as shown in Fig. 6–12. It is noteworthy, that a circular movement of the air created during scavenging continues up to the time of fuel injection, in spite of the fact that the air has been displaced from one end of the cylinder to the other, and compressed to a small fraction of its original volume.

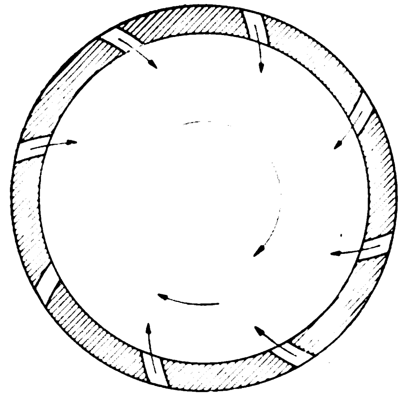
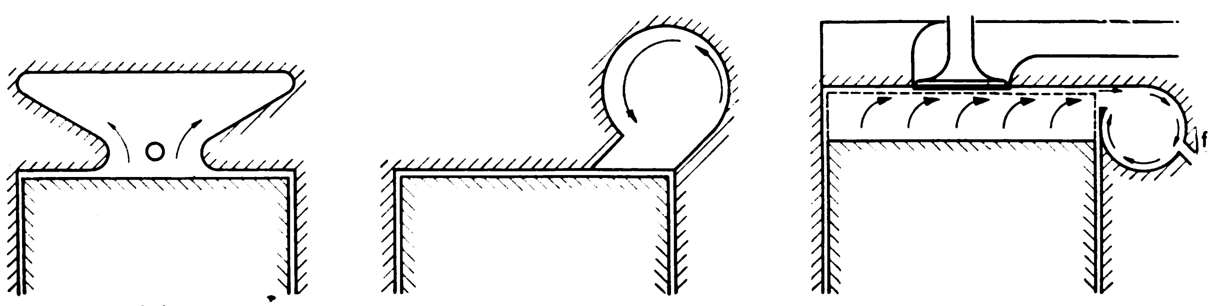


Figure 6–12. Turbulence in a two-stroke engine.

Another method of creating turbulence is used in the so-called Lanova *energy cell* Fig. 6–13. The fuel is injected from the nozzle; a part of the fuel is atomized as it leaves the nozzle, ignites, and burns in the main combustion chamber, while the rest is injected in a more or less solid stream in the so-called energy cell or minor air cell. Here it is atomized or broken up in a fine mist form and ignited; the resulting combustion raises the pressure in the minor air cell over the pressure in the main combustion chamber, and throws the burned and unburned fuel back into the main chamber, creating a strong turbulence (tentatively indicated by the arrows), and helping to burn the rest of the unburned fuel.



(a) (b) (c)

Figure 6–11. Turbulent heads.

6–6. Timing and injection. As already explained in Sec. 6–4, there exists a certain time lag between the time that the fuel is injected into the cylinder and the time that it ignites and begins to burn and raise the cylinder pressure. This time lag, called *ignition delay*, requires an advance of the fuel injection several crank-angle degrees before top center. In addition, it should be noted that there exists another lag in the fuel injection. The beginning of the delivery stroke of the injection pump is set to correspond to a certain position of the engine crankshaft. The injection timing is checked by slowly turning the engine over. The actual admission of the fuel into the engine will start several crank-angle degrees later. The reason for this time lag is the mechanical flexibility of the injection mechanism, taking up of clearances between the various rollers, pins, levers, etc., and the compressibility of the fuel oil, especially noticeable with a long fuel line. This lag is called *injection lag*, and amounts also to several c.a. degrees.

Both the ignition lag and injection lag depend upon a number of various factors and may vary considerably from engine to engine. The following data obtained from actual tests may serve as an illustration. In an engine, operating at 900 rpm., the injection was set to begin at 22° b.t.c.; actual injection started only about 17° b.t.c., which gives an injection lag of 5°; ignition started 8° b.t.c. which gives an ignition delay of 9°, or a total lag of 14° behind the nominal fuel timing. On the other hand, the pump delivery stroke was cut off 3° b.t.c., but due to the expansion of the fuel compressed in the fuel line the actual end of injection occurred slightly after t.c. In other engines the lag may be greater or smaller.

The only way to determine the correct fuel timing is by operating the engine, changing the timing, and finding the timing with which the engine operates best, i.e., has the lowest fuel consumption, carries the highest load without smoking, and runs the smoothest. Such timing is worked out at the engine factory and is given in the instruction book. The operator should maintain this timing, unless the engine operating conditions are radically changed, such as running the engine at a different speed, or using a very different fuel. In this case the proper timing must be found again as mentioned before.

Valve timing. As mentioned in Sec. 5–1, the opening of both the exhaust and intake valves occurs *before* the corresponding dead center, and their closing *after* it. The causes are partly in the mechanical lag of action, due to clearances which must be taken up, and the flexibility of the long push rods, rocker arms, etc., but chiefly in the necessity of a gradual opening and closing of the valves. Thus, there elapses an appreciable time element between the moment when the valve begins to leave the seat, and the moment when it has sufficiently moved away from the seat to allow exhaust gases to pass from the cylinder, in the case of an exhaust valve, and air into the cylinder, in the case of an intake valve. The same holds true for the closing of the valves, but in reverse: several crank angles before a valve touches its seat, the passage becomes so restricted that the flow of gases practically stops. The gradual opening and closing is necessary to overcome the forces of inertia of the parts of the valve actuating mechanism, without exerting undue pressures between the cams, cam followers, and various pins and bearings during the opening of a valve and the pounding of the valve against its seat when the valve is being closed.

The best timing depends upon a number of factors such as valve lift, shape of the cam, speed of the engine, restrictions in the cylinder head passages, etc. The proper timing is found and set when the engine is tested at the factory and is given in the manufacturer's instruction book. It should be maintained and

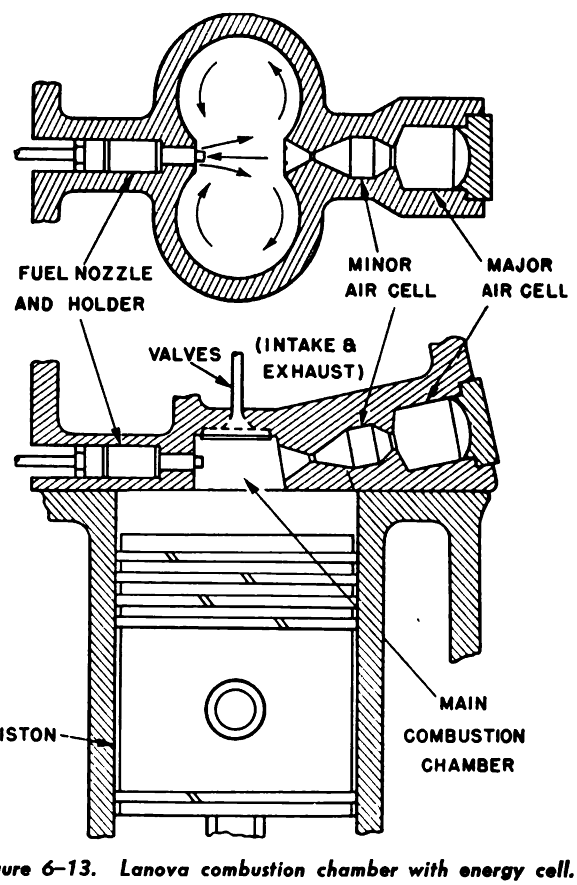


Figure 6–13. Lanova combustion chamber with energy cell.

checked in operation. Even a slight change of the clearance between the cam and the cam followers, which can occur when a valve is ground or when the valve actuating mechanism is disassembled and put together without careful checking, will affect the timing. An increase of the clearance will retard the opening of a valve and advance its closing; a decrease of the clearance will act in the opposite direction. An excessive decrease of the clearance may prevent the valve from seating properly with all the related consequences, such as loss of power, burning of the valve seat, etc.

The same remarks apply to the timing of the exhaust valves of two-stroke engines.

6–7. Supercharging. Supercharging has as its object an increase in the power which an engine of given piston displacement and speed can develop. Since in a diesel engine the power is developed by the burning of fuel, an increase of power requires more fuel to be burned, and therefore, more air must be available since each pound of fuel requires a certain amount of air. According to expression (2–7), other conditions being the same, a given volume will hold a greater weight of any gas, including air, if the gas pressure is increased. Thus, supercharging means a higher pressure of the air charge in the cylinder at the beginning of the compression stroke.

In order to increase the air pressure, in four-stroke engines, the air charge is *not sucked* into the cylinder, or as it is called, is not admitted due to *natural aspiration* by the receding piston, but is *pushed in* by a higher pressure created by a separate air pump or blower. There are three types of blowers used: (1) reciprocating piston pumps, similar to an air compressor; (2) rotating positive-displacement blowers, of the Root-blower type; and (3) centrifugal high-speed blowers, similar to centrifugal pumps.

When a supercharger is applied to a four-stroke engine, the main change required in the engine design is a change in timing of the intake and exhaust valves. The intake-valve opening time is advanced and the exhaust-valve closing is retarded. The two valves are designed to stay open simultaneously for about 80 to 160 degrees, the selection depending upon the normal engine speed. This simultaneous opening is called *overlapping*. Tests have shown that an overlap of 40 to 50 degrees increases the power output of an engine about 5 per cent if the supercharging is very small, sufficient only to eliminate the vacuum in the cylinder during the suction stroke, and up to 8 per cent with a supercharger pressure of 12 in. mercury, as compared with an overlap of 10 to 20 degrees commonly used in unsupercharged engines. The total power gain due to supercharging varies from 20 to 50 per cent depending upon the supercharging pressure, which in present naval engines varies from 5.0 in. to about 12 in. mercury.

It should be noted, that simultaneously with an increase of the mean effective pressure, supercharging increases also the maximum or firing pressure and the maximum temperatures. On the other hand, the fuel consumption per hp.-hr. usually decreases with supercharging, due to an increase of turbulence and hence better mixing of the fuel with the air charge.

Two-stroke engines usually have a blower to obtain scavenge air and their supercharging is obtained simply by increasing the amount and pressure of scavenge air. In addition, a slight change of the exhaust and scavenge timing is made in order to retain more scavenge air at the beginning of the compression stroke.

The supercharging equipment will be discussed in greater detail in Chapter 12.

6–8. Problems. 1. Determine the indicated horsepower developed by a $6 \times 12\frac{3}{4} \times 15\frac{1}{2} \times 400$ four-stroke diesel engine. The indicator card taken with a 320-psi. in. spring has an area of 0.546 sq. in., its length is 2 in.; assume that all cylinders give the same indicator card. *Ans.* 523 ihp.

2. Find the brake hp. of the engine of problem 1, if the load which it was carrying when the indicator card was taken was 300 kw.; assume the efficiency of the D.C. generator as 95 per cent. *Ans.* 423 bhp.

3. Find the friction hp. and mech. eff. of the engine of problems 1 and 2. *Ans.* 100 fhp.; 80.9 per cent.

4. Find the bmep. of the engine of problems 1 and 2. *Ans.* 71.4 psi.

5. The engine of problems 1 and 2 was using 0.559 lb. per kw.-hr. fuel oil. The heat value of the fuel is 18,720 Btu. per lb. Determine the indicated thermal efficiency of the engine under these conditions. *Ans.* 42.4 per cent.

6. Find the over-all efficiency of the engine of problems 1, 2, and 5. *Ans.* 34.3 per cent.

7. Find the injection lag in seconds for an engine running at 2,200 rpm., if the injection is set for 23° b.t.c., and the actual delivery of the fuel into the cylinder occurs at 15.5° b.t.c. *Ans.* 0.00057 sec.

6–9. Questions. 1. What is the purpose of an indicator card?

2. What is indicated horsepower?
3. What is a pressure-time diagram?
4. Enumerate the principal efficiencies encountered when dealing with a diesel engine.
5. What is brake horsepower?
6. What is mechanical efficiency of a diesel engine?
7. What is brake mean effective pressure?
8. What is ignition delay?
9. What is the object of turbulence?
10. Why do the valves in four-stroke engines not open and close on the corresponding dead center?
11. What is the object of supercharging?

CHAPTER 7

STRUCTURAL ENGINE PARTS

A. MAIN STATIONARY PARTS

7–1. General. The main stationary parts of a diesel engine are designed to maintain the moving or working parts in their proper relative positions, so that the gas pressure produced by the combustion is used to push the piston and rotate the crankshaft. The main requirement is strength, next comes low weight, and finally simplicity of design. In naval service, the size of an engine must be the smallest possible because of space limitations. Diesel engines of a few years ago were several times as heavy per horsepower output as the more modern engines. The reduction in weight in recent years has been possible, partly to improvements in materials which provide greater strength per unit area, but mostly due to improved design and methods of calculation and manufacture which permit the use of lighter sections.

7–2. Engine frame. The frame connects the top of the cylinder to the supports for the crankshaft. In the earlier designs, at present used only in large, low-speed engines, the frame consists of a separate cylinder block, crankcase, and bed plate with an oil pan or sump, Fig. 7–1. The main bearings supporting the crankshaft were held in the crankcase, while the pistons operated in the cylinder block above it. The gas-pressure load was taken up by tie bolts running from top to bottom. Cylinder block, crankcase, and bed plate were made of gray-iron castings.

Modern designs of high-power output engines have frames welded of steel with plates located at places where the loads occur, Fig. 7–2. The customary arrangement combines the cylinder block and oil pan with the main bearing supports, although a separate crankcase section is sometimes used. Cylinder blocks and crankcases of small high-speed engines are still made of cast iron.

Crankcase. The crankcase is often integral with the cylinder block. In the models where it is a separate section, it generally consists of a plain rectangular frame with cross-ribbing to provide rigidity. Occasionally, the main bearings are held by a cross-ribbing in the crankcase, but more often they are hung from the bottom of the cylinder block. Access doors are provided at every cylinder to permit assembly and observation of the bearings.

7–3. Cylinders. The cylinders were separate units on some older models, but in modern naval engines they are secured within the block which also contains

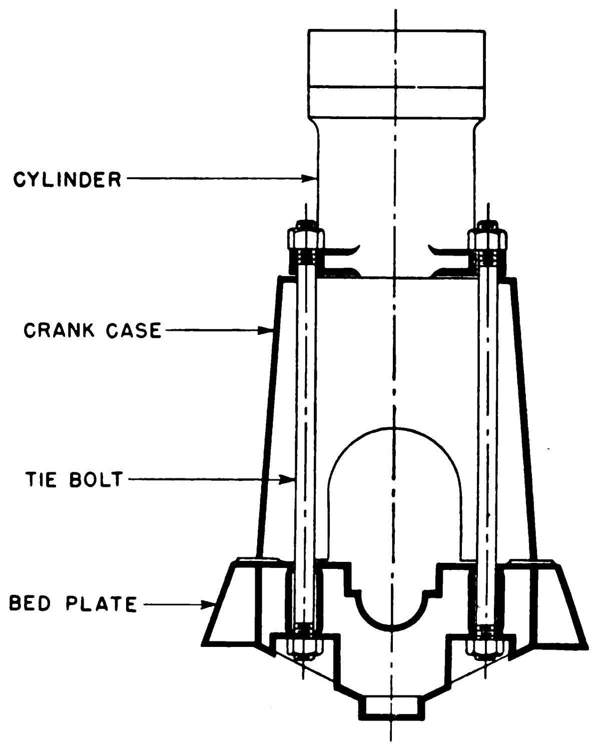


Figure 7–1. Engine frame with tie bolts.

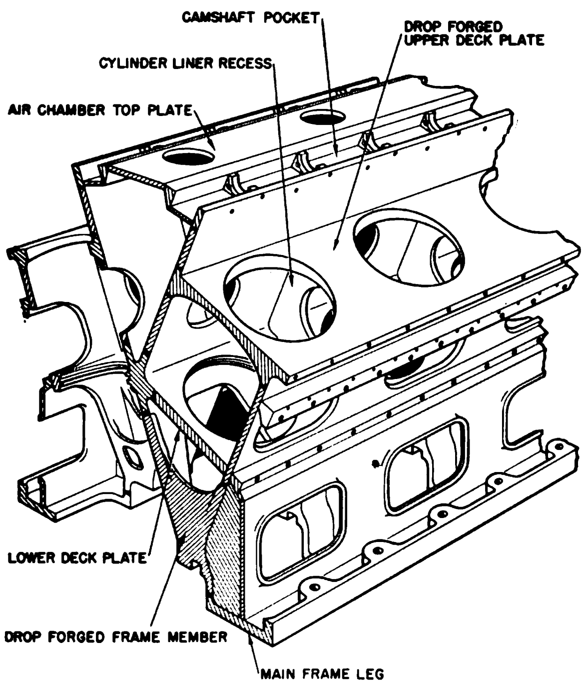


Figure 7-2. Sectional sketch of Winton welded frame.

passages for cooling water, lubricating oil to the bearings, and intake air. Each cylinder is secured in a separate compartment with cross-bracing between the compartments.

Cylinder liners. There are three types of cylinder liners in use:

a. *Dry type:* a simple sleeve with thin walls inserted in the cylinder which is part of the cylinder block, Fig. 7-3. The cooling water moves about the outer cylinder and does not contact the liner. The liner is inserted in the cylinder with a light press fit. When worn or scored it can easily be removed and replaced by a new liner.

b. *Wet liner:* a sleeve whose outer surface comes into direct contact with the cooling water, Fig. 7-4. The liner is normally sealed against water leakage at the top end by a gasket under the flange or a machined fit, and by rubber or neoprene rings around the lower end. The liner is free to expand or contract lengthwise. The thickness of the wall is such as to take the full working pressure of the gases.

c. *Water jacketed type:* the liner has its own cast-on or permanently shrunk-on jacket around the outside for the circulating cooling water, Fig. 7-5. The water is admitted into the bottom of the jacket and leaves through the top. This type is used mostly in two-stroke engines where it is difficult to obtain a watertight seal around the ports when using a wet liner, because of the expansion of the liner due to heat during engine operation.

The cylinder liner must be made of material which will enable the piston and rings to move up and down with the minimum friction, and will give the least wear to both the liner and the piston parts. Cast iron is the usual material, although steel sleeves are sometimes used. A recent development has been a coating of 0.003- to 0.006-in. electro-deposited porous chromium. The chromium resists wear, while the pores in the plating hold lubricating oil and maintain a lubrication film necessary to reduce friction and scuffing.

Cylinder heads. The cylinder head seals the combustion chamber, and in most engines, contains the valves and passages for intake and exhaust gases, the fuel injection nozzle, the air starting and relief valves, and passages for the cooling water from the cylinder jacket as shown in Fig. 7-5. It is a casting of alloy iron, seldom of aluminum. Because of the heat passing through it from the combustion chamber and the exhaust passages, it has to be water-cooled. Such cooling prevents excessive temperatures which might crack it and which would interfere with the operation of the fuel injection nozzle and all other valves. The larger-bore engines have individual heads for each cylinder, while small-bore engines may have a single head covering all cylinders, or pairs of cylinders.

7-4. Other parts. *Crankshaft bearings.* In older designs and in very large, low-speed, diesel engines, the main and crankpin bearings consist of heavy,

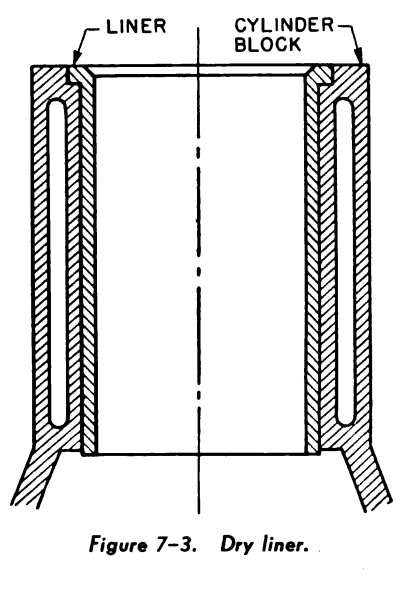


Figure 7-3. Dry liner.

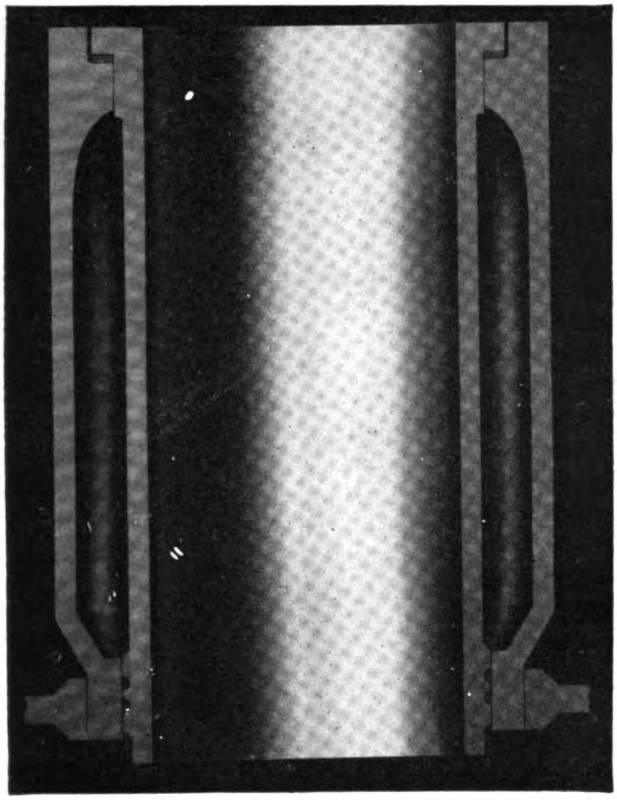


Figure 7–4. Wet liner.

B. MAIN MOVING PARTS

7–5. Crankshaft. The forces acting on a diesel engine crankshaft are high because of the high peak gas loads and the inertia forces of the moving parts. The main requirements are mechanical strength, and lengthwise and torsional rigidity. The shafts have relatively large diameters and are supported by main bearings between each pair of cranks, as shown in Fig. 7–6. At present, most crankshafts are forged, some from open-hearth, but the majority from high-strength alloy steels. However, advances in the understanding of the stress distributions in diesel crankshafts are changing the situation. It has been found that the removal of metal at certain sections will redistribute the stresses and result in a stronger shaft. Casting permits easier application of such improved shapes, and as a result high-strength cast-iron, and in some cases cast-steel, crankshafts are

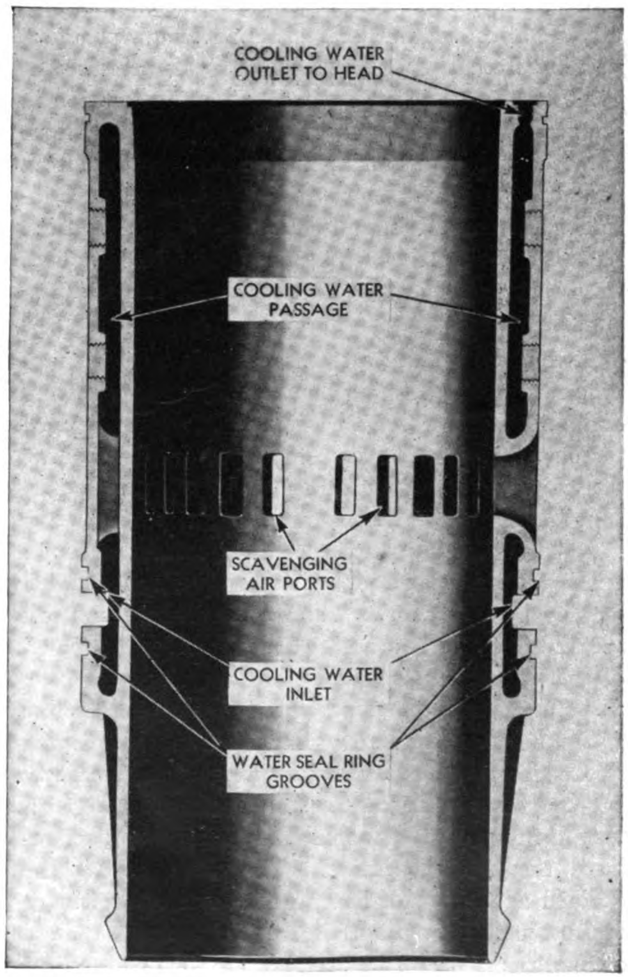


Figure 7–5. Water-jacketed liner.

cast-iron or cast-steel boxes with a thick, up to one-half inch, babbitt lining. Each bearing must be hand-scraped to a running fit with its journal. All modern diesel engines, regardless of size and speed, have precision bearings. Precision bearings are separate from the saddles and connecting rods. They consist of relatively thin steel, bronze, or brass shells, with a lining of bearing metal, which is generally 1/32-inch or less in thickness. The bearing metal may be one of several types which have proved satisfactory: lead-base babbitt, copper-lead, cadmium-silver, etc. Grooving is kept to the minimum, and wedge-type lubrication is used to the fullest extent, as explained in Sec. 10–6.

Crosshead guide. Only double-acting diesel engines require a crosshead and crosshead guide; however, a few of the largest single-acting engines also use them. The purpose of the crosshead guide is to take the side thrust coming from the angularity of the connecting rod, which otherwise would be taken by the cylinder liner. The bearing surface of the crosshead guide is a flat slipper. Bearing loads are low, and, with proper lubrication, ordinary babbitt usually suffices as a bearing surface.

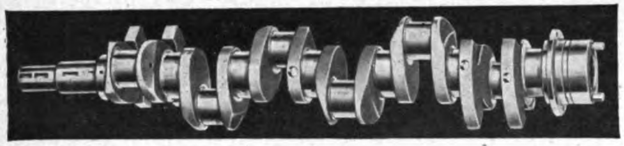


Figure 7–6. Crankshaft of a six-cylinder diesel engine.

being used in several modern diesel engines. Many forged steel shafts are surface-hardened by electrically heating the surface alone and quenching it with sprays of water. Surfacing of the journals with a harder metal, such as electro-deposited chromium which will take a smooth dense finish, is being considered to reduce shaft wear and improve bearing conditions. The shafts are drilled through the crank webs to admit pressure lubrication from the main bearings to the crankpins and wrist pins. In some cases the oil is carried further and is used for cooling the piston crowns.

7–6. Pistons. Pistons of single-acting engines usually are of the trunk type. They must carry pressures varying from a slight vacuum to peak pressures of 1,000 to 1,200 psi., with the resulting fluctuation in temperature setting up expansion stresses. They have to withstand high bearing and side-thrust loads from contact with the cylinder liner. Often they have insufficient lubrication, and have to resist wear on the outside cylindrical surface and in the ring grooves from the pressure and sliding action of the compression rings.

Pistons are usually cast because it is easier to provide satisfactory ribbing on the interior in a casting and still keep the weight low. However, forged pistons are used with some engines. Pistons of slow, heavy-duty engines, where strength is a more important factor than weight, are made of cast-iron. They may be cooled by water or oil circulated through baffles within the piston, but generally they are cooled merely by air contact, and with lubricating oil sprayed from the connecting-rod cap under the crown of the piston. Pistons of higher speed engines have to be made as light as possible because of the effect of the mass on the bearing loads. For several years the tendency was to use aluminum pistons. Recently the trend has changed to pistons of cast-iron with very thin walls and cooled by lubricating oil circulated through the inside. The cast-iron has the decided advantage of having the same coefficient of expansion as the cylinder liner. This permits the use of smaller piston-to-liner clearances when cold, without the danger of seizure when operating under a heavy load. Fig. 7–7 shows a piston of a single-acting two-stroke engine, and Fig. 7–8 the piston of an opposed-piston two-stroke engine.

Pistons for use in double-acting engines are built up of several sections, and are closed at both ends because both ends are used for combustion chambers. They are cooled by oil entering and leaving through the piston rod.

Wrist pins. All of the load developed in the cylinder passes through the wrist or piston pin. It is the only connecting link between the piston and the connecting rod. Most wrist pins are supported at both ends by bronze bushings in the piston bosses, and are maintained in place by caps which are fitted to each side of the piston. The connecting rod swings on a bronze bushing or needle bearing at the center of the pin. Such wrist pins are known as *full-floating* pins. In some engines the wrist pins are locked in the piston at the ends and are known as *stationary* pins. The disadvantage of this design is that all swinging movement is confined to the connecting rod bearing and there is some danger of less uniform wear. A third type, a *semi-floating* pin, is supported at both ends by bronze bushings in the piston bosses, and clamped in the middle in the connecting-rod end.

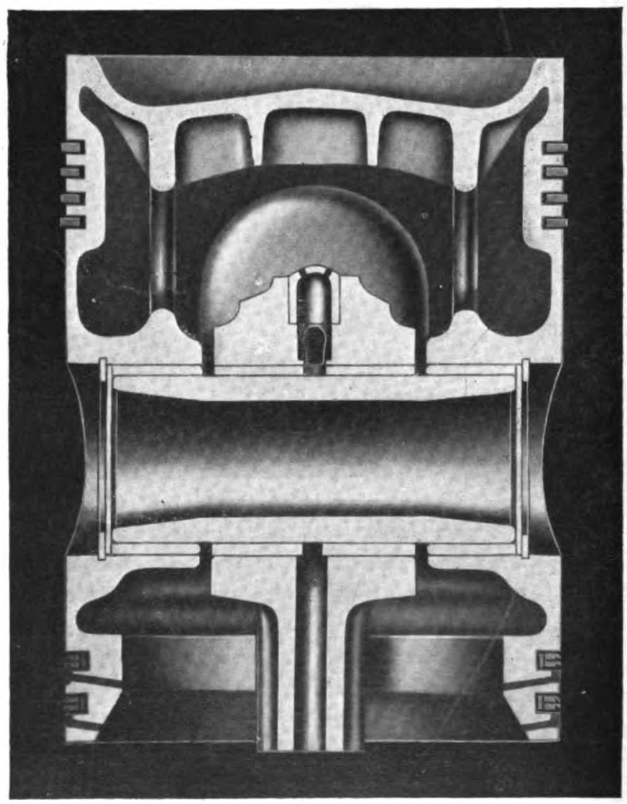


Figure 7–7. Piston of a single-acting two-stroke engine.

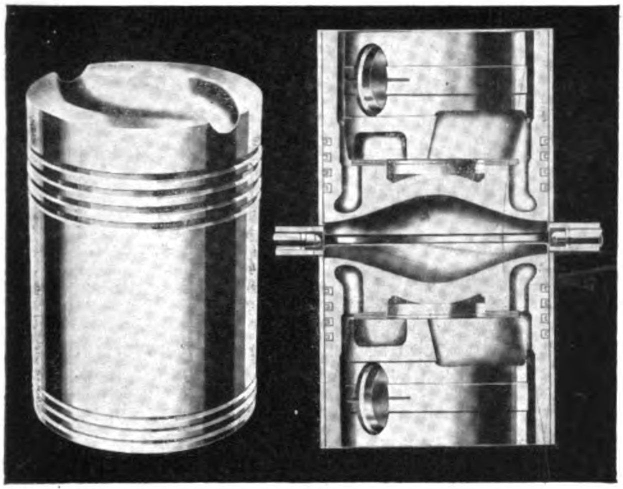


Figure 7–8. Piston of an opposed-piston two-stroke engine.

Wrist-pin bearings operate under rather severe conditions. In addition to the great load from the piston pressures, there is the handicap of less efficient lubrication because the swinging motion does not help to form an oil film as much as the rotary motion of a journal. The wrist pin is made of a steel alloy of sufficient strength to carry the load and must have a fine-finish hardened surface to obtain good bearing action.

7–7. Piston rings. At the top of the piston are several *compression rings* which serve three purposes:

1. They seal the space between the piston and the liner, thus preventing the high-pressure combustion gases, or the air charge during the compression stroke, from escaping down the liner.

2. They transmit heat from the piston to the water-cooled cylinder liner.

3. They damp out part of the fluctuations of the piston side thrust.

The *oil-scraper* or *oil-control rings* usually are located at the bottom end of the piston; in some engines they are placed above the wrist pin. Small engines use one, larger engines use two or three oil-control rings to a piston. Their purpose is to scrape off most of the lubricating oil splashed upward by the crankshaft and connecting rod, thus reducing the amount of oil carried upward and burned in the combustion chamber. At the same time they must allow sufficient lubricating oil to be carried to the upper part of the liner during the upstroke, so that there will be proper lubrication for the piston and the compression rings. Double-acting pistons have no oil-scraper rings as no oil is splashed on the liner.

Compression rings are made of gray cast-iron. Some types have special facings, such as a bronze insert, Fig. 7–10a, or a treated surface, to facilitate seating-in to the liner. To expedite the wearing in, or seating of the ring face, some rings have a slight angle, ½ to 1 degree to the face, so that at first the contact area is very small, wear is rather fast, and later decreases.

The type of compression ring most widely used has a rectangular cross-section. The diameter of the ring is slightly larger than the cylinder bore, and part of the ring is cut away to permit it to go into the cylinder. The difference in diameters produces a pressure against the liner wall. The pressure of the upper rings is increased by the additional action of the gases. The combustion gases or compression air enter behind the ring through the vertical clearance which always exists between a ring and its groove and force the ring against the cylinder liner.

Some engines have compression rings with the bottom wall or both bottom and top walls beveled, making the ring thinner at the inside than on the outside

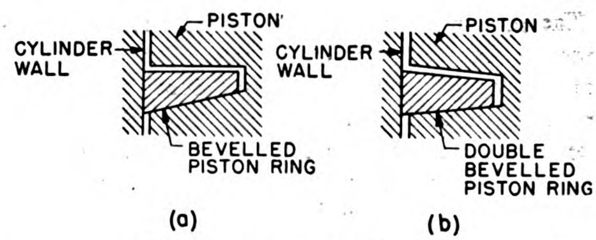


Figure 7–9. Bevelled piston rings.

diameter, Fig. 7–9. The groove in the piston is machined to the same shape. The gas pressure acting on the top wall, due to the beveled bottom surface, produces an additional force pressing the ring against the cylinder wall and helps to seal it. On the other hand, at each reversal of the side thrust of the piston, the ring slides slightly into the groove, is pressed against the upper groove wall, crushes the carbon which is deposited on it, and keeps the ring from sticking. Some engines use bi-metal rings, Fig. 7–10b,

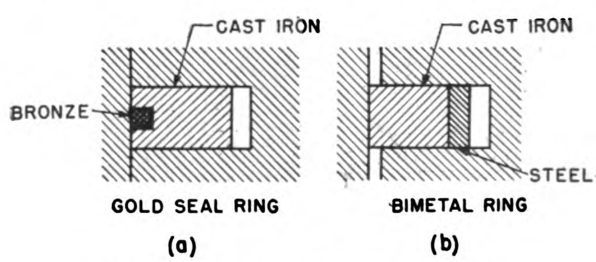


Figure 7–10. Compression piston ring.

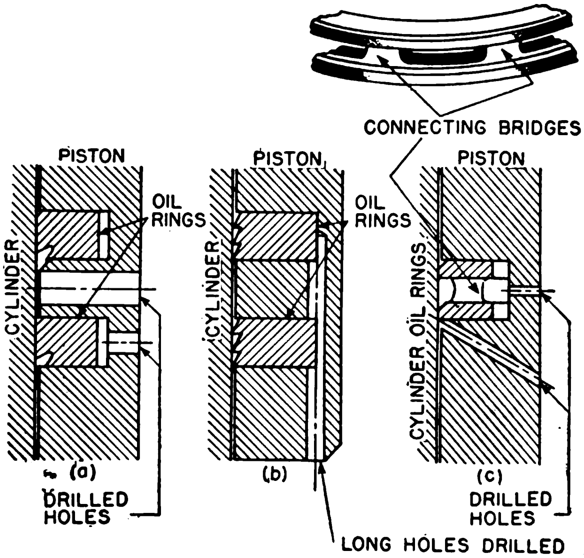


Figure 7–11. Oil-control rings.

in which the cast-iron wearing face is brazed to a steel inner ring to obtain increased strength and to reduce the probability of ring fracture.

The oil-control rings have a narrow face so as to obtain a higher unit wall pressure, and are often undercut to give a scraping edge. Some engines use flexible rings which follow deviations in the cylinder liner bore. In some designs the ring has one, in others two narrow scraping edges, and the piston has rows of holes drilled in it for draining the oil through the bottom of the ring grooves or through the lands between the grooves or both, Figs. 7–11a, b, and c.

The oil scraped by the ring must be drained off immediately, otherwise it will build up a pressure which will force the ring back into its groove and stop the scraping action. It is important that the drainage from the piston grooves be complete. Inadequate drainage means faulty scraping, higher lubricating oil consumption, and a darker color of the engine exhaust gases. Spring steel expanders are sometimes used behind the rings to increase the wall pressure and improve the scraping action.

The gap between the ends of the compression rings when inserted cold in the cylinder must be sufficiently large so that when the ring expands with the full piston temperature the ends will not be pressed together and buckle the ring. The way in which the ends are cut varies. Most rings have the ends cut

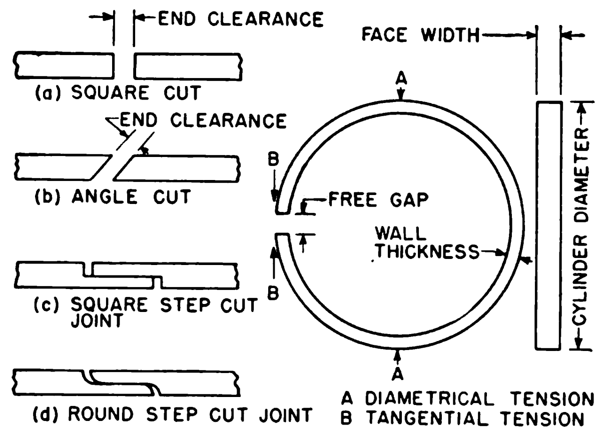
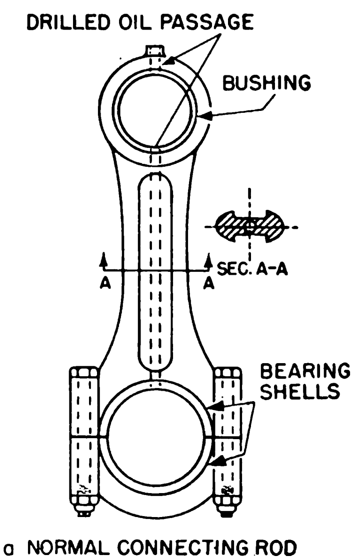


Figure 7–12. Piston rings.

square, Fig. 7–12a. A design which makes gas blow-by more difficult has the ends cut at a 45° angle, Fig. 7–12b. There are several designs of step-seal rings, Figs. 7–12c and 12d. However, there is little gained



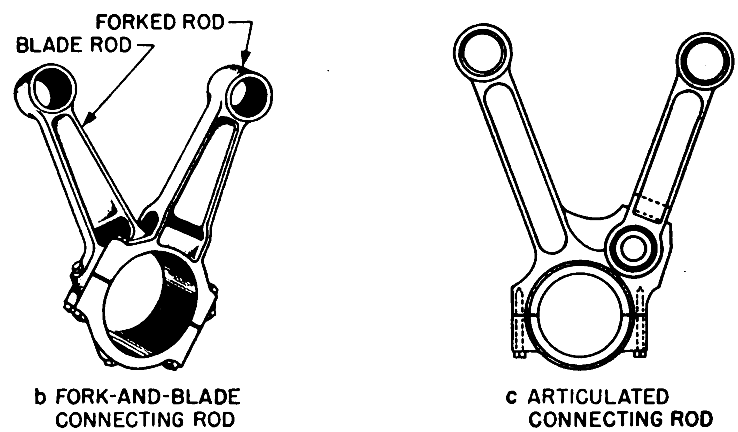


Figure 7–13. Connecting rods.

by this more complicated shape. Oil control rings in two-stroke engines are likely to catch the ring ends in the ports over which the rings slide because of the ring flexibility. To prevent this, the ends are notched and a pin is installed in the piston groove to hold the ring ends always in line with a bridge between the ports.

7-8. Connecting rods. Connecting rods used in Navy engines are mostly of the type used in automobiles, Fig. 7-13a. They have an eye at the small end for the piston pin bearing, a long shank, and a big-end opening which is split to take the crankpin precision bearing shells. The rods are forged of a high-strength alloy steel. Most connecting rods are rifle-drilled from the big end to the eye for oil flow from the crankshaft to the piston pin, and in some engines to the piston crown to cool it. Very often the rod shank is H-shaped for maximum strength with minimum weight. Types of rods include: (1) the *normal* shape, Fig. 7-13a, used with only one cylinder to a crankpin of two cylinders, offset so the rods can operate side-by-side; (2) *fork-and-blade* rods in V-type engines, Fig. 7-13b, in which the big end of one rod has the normal shape while the rod of the piston in the opposite bank is widened and split into a fork shape straddling the first rod; (3) *articulated* connecting rods, Fig. 7-13c, in which one rod, the *master rod* is of the conventional shape except that it has a projection off the shank with an eye to which the rod for the piston in the opposite bank, called the *articulated* or *link* rod, is attached. Finally, there is the rod type used on the pancake engine, in which the big end of the rod consists of a short pad with the bearing

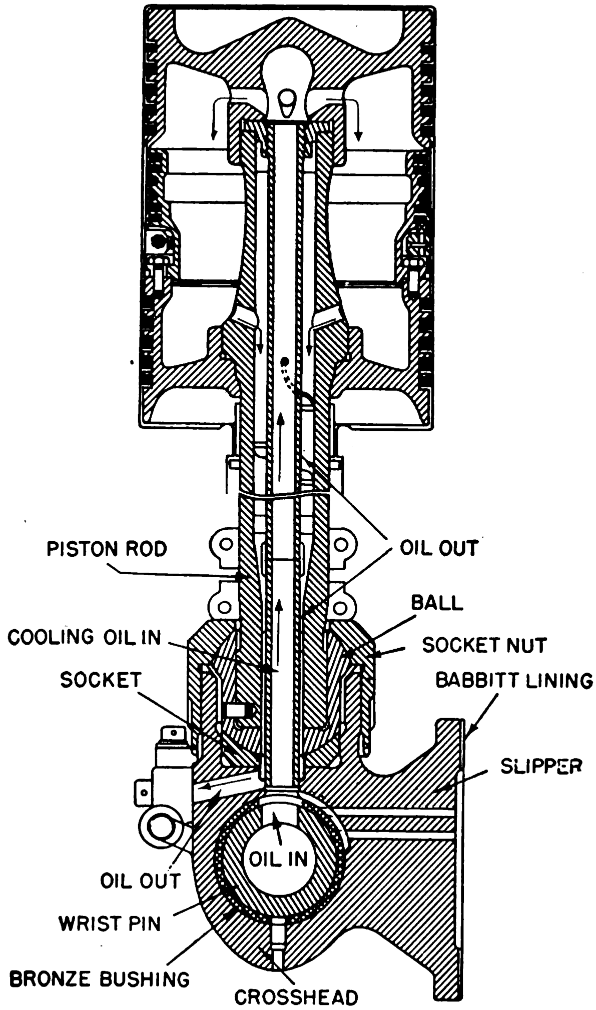


Figure 7-15. Crosshead.

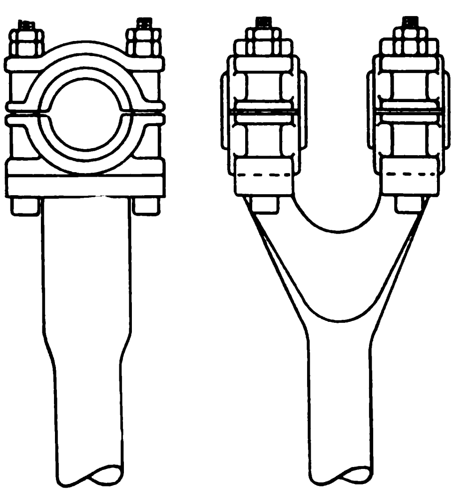


Figure 7-14. Forked connecting rod for crosshead piston.

metal directly on the rod. Four such rods are located radially around a crankpin with retainer rings around the outside of the pads to hold all four against the crankpin. Connecting rods of double-acting engines have a forked small end with the wrist pin fastened in it and swinging in the bearing fastened to the crosshead, Fig. 7-14.

7-9. Piston rods. The piston rod is a hollow, straight rod, used to connect the piston to the crosshead in engines with crossheads. Its motion is straight reciprocation. Within the rod are contained the passages for the cooling oil entering and leaving the piston. If the rod is used with a double-acting engine, the diameter of one end of the rod must not be larger than the diameter of the main rod length so that it can be inserted through the stuffing box. The rod is generally assembled together with the piston, often with the piston parts clamping the end of the rod. It is installed through the stuffing box in the lower head and fastened to the crosshead, usually by a screwed connection, as shown in Fig. 7–15.

7–10. Crosshead. The crosshead, Fig. 7–15, transmits the load from the piston rod to the connecting rod and takes all the side thrust usually taken by the piston. The design now generally used includes a ball-and-socket joint with the piston rod end, to take care of any unavoidable slight misalignment, and a bearing for the wrist pin fastened in the forked end of the connecting rod.

The crosshead, long oil-cooled piston, and long piston rod, increase considerably the weight of the reciprocating masses of double-acting engines. At a speed of 720 rpm., used in Navy engines of this type, the loads on all bearings, main, crankpin, and wrist pin, become rather large due to the forces of inertia and make more important good lubrication of these bearings.

7–11. Questions. 1. What is the purpose of the main stationary diesel engine parts?

2. State the material and procedures used in making the engine frames of naval high-power output diesel engines.

3. What types of cylinder liners are in use in naval diesel engines?

4. What type of crankshaft bearings are used in present naval diesel engines?

5. What methods are used in making crankshafts for naval diesel engines?

6. What material is used in present engine pistons?

7. What are the main purposes of piston rings?

8. What is meant by an oil-control piston ring?

9. What types of connecting rods are used in naval diesel engines?

10. In what engines is a piston rod necessary?

11. What is the purpose of a crosshead?

CHAPTER 8

VALVE GEAR

8–1. General. The combination of all parts, including the various valves, which control the admission of the air charge in four-stroke engines, the discharge of exhaust gases in all four-stroke and many two-stroke engines, the admission of fuel in some diesel engines, and the admission of compressed air for starting all larger engines, is called the *valve gear*. The combination of those parts which only *operate* the various intake, exhaust, fuel, and air starter valves, open and close them at the proper moment in respect to the position of the piston, and hold them open during the required time, is called the *valve actuating gear*, the word actuating, meaning producing action or moving.

In Fig. 4–1 the various parts of a valve gear were indicated. Their action is as follows: the rotating camshaft 19 with the cam 20 pushes the roller cam follower 21 and push rod 22 upward, thus transmitting the cam action to the rocker arm 23; the latter changes the upward motion of the push rod to a downward motion of the valve, and the valve spring 24 returns the valve to its seat, and closes it; the spring action is controlled by the closing side of the cam again transmitted through the same intermediate members. In some engines the camshaft is raised to the cylinder-head level which eliminates the need for push rods, and the roller-type cam follower is made part of the rocker arm.

8–2. Cams and camshaft. A cam is an eccentric projection on a revolving disk which controls the operation of a valve, usually through various intermediate parts as described above. Originally cams were made as separate pieces and fastened to the camshaft. However, most modern diesel engines, even in larger sizes, now have cams forged or cast integral with, i.e., a part of the camshaft, and then machined, the camshaft being similar to automobile-engine camshafts. Therefore, if *one* cylinder is timed correctly, *all* are, and any change in timing will affect all cylinders.

In operation, cams are subjected to shock action, and in order to reduce wear they must be hardened. The shape of the cam determines the points of opening and closing of the valve, the speed of opening and closing, and the amount of the valve lift. The required cam shape or profile surface is obtained by accurate grinding. The sides of a cam are called *flanks*, the highest part is called the *nose*. The cam may have flanks which are curved outward and are called *convex-curve flanks*. Fig. 8–1a, 8–1b and 8–1c, show cams with straight sides, called *tangential flanks*. Figs. 8–1a and 8–1b show typical *intake* and *exhaust* cam shapes for four-stroke engines; Fig. 8–1c shows an *exhaust* cam for a two-stroke engine.

Fig. 8–2a shows an adjustable *fuel-injection* cam as used on four-stroke engines with a common-rail or similar injection system (explained in Chapter 9), in which the nose may have to be exchanged and the injection accurately timed. The nose is made in the form of a hardened steel insert *i*, held by set screws *s*, and blocks or keys *k*, which can be filed to obtain the exact location of the nose. Fig. 8–2b shows a fuel-injection cam as used on a two-stroke engine with a unit injector (Chapter 9).

Fig. 8–3a shows a typical shape of the cam for the *air-starter* valve of a four-stroke engine; it has a quick

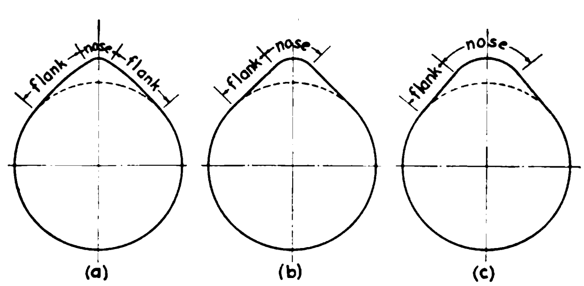


Figure 8–1. Profiles of intake and exhaust cams.

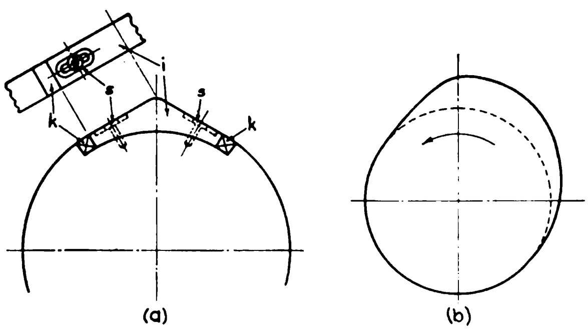


Figure 8–2. Fuel-injection cams.

and abrupt lift which gives full valve opening almost immediately, and prevents throttling of the compressed air through the starting valve. A gradual seating of the valve is obtained by the smoother shape on the closing side. Fig. 8–3b shows an air-starting cam for a high-speed two-stroke diesel engine.

In a four-stroke engine there are at least two cams for each cylinder, one for the inlet and the other for the exhaust valve. Depending upon the fuel-injection system used, there may be another cam to operate the fuel injector, and if air-starting is used, still another cam for the air-starting valve. If the engine is reversible, there will be another set of cams for the other direction of rotation of the engine, resulting in as many as eight cams per cylinder. In two-stroke engines there is no cam for the intake, but if exhaust valves are used, at least two exhaust valves per cylinder are present which may be operated either by common or two separate cams, so that the number of cams is about the same as in a four-stroke engine.

Camshaft. As already mentioned, in some engines the camshaft is a straight round shaft and the cams are separate pieces, machined and keyed to the shaft. However, in most modern diesel engines the cams and the shaft are forged or cast in one piece. In some larger engines the integral camshafts are made up of two or more sections to facilitate replacement in a restricted space. The sections are bolted together by flanges with fitted reamed holes to assure accurate timing. Most camshafts are made of forged steel, usually of nickel-chromium alloy steel, and the larger ones are usually bored hollow. They are heat treated and in some cases the cams are surface hardened. The camshafts are carried in plain bearings.

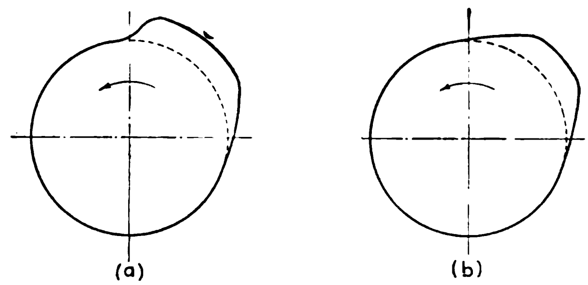


Figure 8–3. Air-starting cams.

Camshaft drives. Camshafts are driven from the engine crankshaft by various means. Fig. 8–4a shows a drive by a train of straight spur or helical spur gears used in most Navy diesel engines. Fig. 8–4b shows a drive by two pairs of helical or screw gears and an intermediate vertical shaft; and Fig. 8–4c a similar drive but using two pairs of bevel gears. Fig. 8–4d shows a chain drive, *d* is the crankshaft, *p*, the sprocket keyed to it, *g*, the camshaft sprocket and *m* the camshaft.

In two-stroke engines the camshaft rotates at the same speed as the crankshaft, while in four-stroke engines it rotates at half crankshaft speed.

8–3. Cam followers. Modern diesel engines use several types of cam followers:

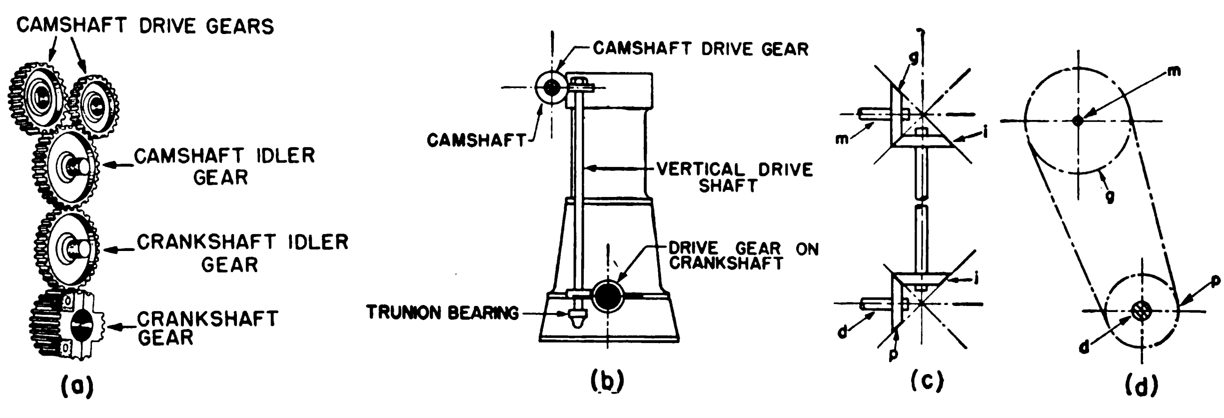


Figure 8–4. Types of camshaft drives.

1. The flat or mushroom follower, Fig. 8–5a, operated by a convex cam.

2. The roller-type follower, Fig. 8–5b, with a tangential-flank cam.

3. The pivoted follower, Fig. 8–5c, which may be used with a cam of any shape.

The combination of the convex-flank cam with the flat mushroom follower provides a considerably faster valve opening and closing and is used on smaller engines, running at higher speeds. The disadvantage of this type of follower is a faster acceleration which results in higher inertia forces. On the other hand, this combination gives a lower deceleration of the valve assembly as the latter approaches the top or nose of the cam, and consequently requires somewhat smaller spring forces to prevent *bouncing* of the valve gear. Bouncing of the valve gear occurs after it has reached the maximum acceleration point during the lifting of the valve, and from then on it must be decelerated as the follower nears the top of the lift at the nose of the cam. The inertia of the valve gear at this point tends to lift the valve faster than the cam action. If the valve spring does not have enough force to decelerate the moving parts of the valve gear at the same rate as the contour of the cam, then the follower will leave the surface of the cam, if only for an instant. When it makes contact again, it does so with considerable shock, causing pounding and excessive stresses and wear in various parts of the valve gear. This is called *bouncing* and is very undesirable. The sliding part *t* of a follower, Fig. 8–5, which moves up and down in a bored hole above the cam and takes the side thrust exerted by the cam, is called a *valve lifter* or *tappet*.

The pivoted-type follower, Fig. 8–5c, in its action resembles the roller follower. Its main advantage is that the side thrust from the cam is taken by the pivot of the lever arm instead of by the sliding tappet as with the other types, Figs. 8–5a and 8–5b.

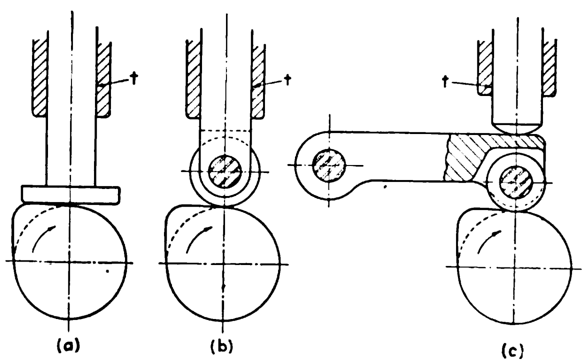


Figure 8–5. Cam followers.

The *rollers* of the followers are made of steel, hardened and accurately ground to size and to a true circle. It is important that there be little or no play in the rollers and that they remain concentric, meaning that the center of the outside circle and of the hole in one roller are at the same point.

8–4. Rocker arms. Push rods. The rocker arm has one end in contact with the top of the valve stem, and the other end, by means of a hardened-steel roller, is in contact with the rotating cam profile as it rotates, if the camshaft is close, near the cylinder head, Fig. 8–4c. If the camshaft is located below, as in Fig. 4–1, the other end of the rocker arm is in contact with the upper end of the push rod; the lower end of the push rod is in contact with the rotating cam through a cam follower. In most designs, the rocker arm is pivoted, not always at or near its middle, and the pivot pin is held in brackets secured to the cylinder head.

When two or more valves of a cylinder must be opened and closed at the same time, as in the case of the exhaust valves of some General Motors Company two-stroke engines, one rocker arm may be used to operate two valves by acting upon a so-called *bridge* between the valve stems, Fig. 8–6. The bridge has special cams *e* in contact with the valve stems, and a lower extension acting as a guide. An auxiliary spring *s* is used under the bridge to offset its inertia and that of the rocker arm, and to assist the valve springs in maintaining the cam follower in contact with the cam at all times. Fig. 8–6 also shows the arrangement of the push rod designed to take care of the slight swinging of its upper end. The lower end of the push rod has a hardened spherical insert which rests in the bored tappet that is accurately lapped into the bored bushings; these are pressed into special cast-iron supports *c*. The upper end of the push rod is connected to the rocker arm by a ball-and-socket joint. The entire assembly is lubricated from the force-feed lubrication system of the engine.

8–5. Valves. *Valve requirements.* Getting the fresh air into, and the exhaust gases out of the engine cylinders requires power, referred to as the *pumping loss*. In order to reduce the back pressure during the exhaust process, the exhaust valve openings are made as large as practical. This is particularly important in the case of two-stroke engines, since the entire exhaust process occurs in a small fraction of the piston stroke, and scavenging must be accomplished entirely by the pressure of the fresh-air charge. For these

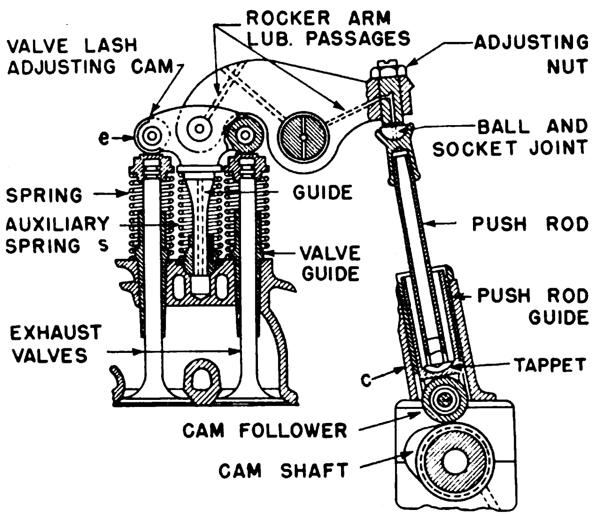


Figure 8–6. Winton valve actuating gear.

reasons, two-stroke diesel engines generally use two to four exhaust valves per cylinder.

In the case of four-stroke engines, the exhaust valve opening is not so critical because the exhaust gases are forced out by the piston during an entire stroke. The inlet-valve opening is more important since all the intake air enters the cylinder through it. Any restrictions to the air flow not only increase the pumping loss but also reduces the density of the air charge. Reduction in the density of the air charge means that less weight of oxygen is available per intake stroke; consequently, less fuel can be burned and the maximum power which can be developed will be smaller.

These conditions become more pronounced as engine speeds are increased; the pumping losses increase rapidly due to the higher velocity of gas flow, and the density of the air charge decreases. For these reasons, the power of every high-speed engine reaches a maximum or *peak* of its brake-horse-power curve at a certain speed beyond which an increase in speed results in a drop in power output. Because of these conditions, intake valves are made as large as possible, and in some engines are made larger than the exhaust valves.

Poppet valves. Intake and exhaust valves used in diesel engines are all of the so-called poppet type, consisting of a disk or *head* at the end of a rod or *stem.*

Poppet valves have heads with conically shaped or beveled edges and bevel seats, which give them a self-centering action. Most poppet valves have bevel seats ground at an angle of 45° with the plane of the valve head, as this has proved to give the best service, especially under the severe operating conditions usually encountered by exhaust valves. In order to obtain a greater opening area for a limited valve lift, some inlet valves are made with a flatter angle, usually 30°. For exhaust valves, an angle smaller than 45° makes the edge of the heads too thin, and more subject to the corrosive action of the hot exhaust gases.

The maximum diameter of a poppet valve is limited, chiefly by the problems of valve weight and cooling, to approximately one-half the diameter of the cylinder bore.

Valve construction. The conditions of operation of poppet valves in high-speed engines are most severe. The valves must be opened as quickly and as widely as possible; they must remain open as long as possible and then be closed as quickly as possible. In order to reduce the stresses in the valve gear due to the inertia forces required to accelerate the valve during opening and closing, the valves themselves should be as light in weight as possible. On the other hand, the valves must be made sufficiently sturdy to take the constant pounding due to continuous opening and closing. They must be able to withstand the extreme temperature and pressure occurring in the engine cylinder and maintain a gastight seal.

The construction of diesel engine valves, both intake and exhaust, follows the same general practice as that developed for heavy-duty automotive engines. Exhaust valves are usually made of silicon-chromium steel (silchrome) or steel alloys containing a high content of nickel and chromium to resist the corrosion of high-temperature gases. Often a hard alloy such as stellite is welded to the seating surface of the valve head, as well as to the tip of the valve stem, to increase the hardness of the surfaces subjected to the constant pounding of closing and opening. To improve the wearing qualities of the valve stem, which usually operates in a bushing of softer metal, the stems of many exhaust valves are made of a steel different from that used for the heads, to which they are welded to form so-called built-up valves. Inlet valves are not subjected to the corrosive action of the hot exhaust gases, and are usually made of cheaper, low-alloy steels.

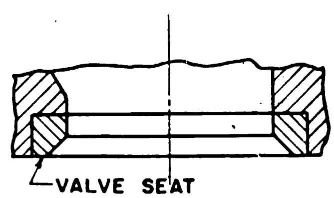


Figure 8–7. Valve-seat insert.

8–6. Replaceable parts. *Valve-seat inserts.* All poppet valves formerly were seated directly on the surface of the cast-iron cylinder head. This practice is satisfactory for most inlet valves, but usually will not give good service for exhaust valves of high-output engines. At the high temperatures resulting from heat transfer from the exhaust gases and contact with the hot exhaust valves, cast-iron seating surfaces tend to soften and erode due to the constant pounding of the valves. In order to increase the service life of exhaust-valve seats between reseatings, it is now common practice to employ valve-seat inserts made of alloys which resist these high-temperature operating conditions. In some high-output engines, valve-seat inserts are used on both inlet and exhaust valves.

Valve-seat inserts consist of rings of a heat-resisting alloy which fit into counterbored recesses in the valve port opening, Fig. 8–7, and are generally held in place by a shrink fit. To obtain a shrink fit, the diameter of the insert is made slightly larger than the counterbore into which it goes. To install the insert, it is shrunk by cooling with dry ice, which has a temperature of −110° F, or the cylinder head is expanded by heating it in boiling water or oil, or both methods are used, after which the insert will go readily into the counterbore. As soon as the insert and cylinder head assume the same temperature, the expansion of the valve insert or contraction of the cylinder head will hold the insert firmly under slight compression. Valve-seat inserts are made of alloys having approximately the same rate of heat expansion as the cylinder head. The inserts, therefore, will be held in place firmly at all operating temperatures so long as their temperature is not lower than the temperature of the cylinder head.

Valve guides. The guide holes for the valve stems wear, due to the continuous up-and-down motion of the stems, and in modern engines, separate bushings are inserted in these holes to act as valve guides. These bushings not only can be renewed easily when worn, but they can be made of material which wears less than the cast-iron of the cylinder head. Valve-guide bushings are commonly made of a cast-iron which has good wearing qualities when used with the steel of the valve stem employed. In order to obtain improved heat conductivity from the valve stem, some engines use bronze bushings for valve guides.

Valve guides for exhaust valves are sometimes made longer, i.e., extend further into the valve passage, than guides for inlet valves. The longer guide more effectively protects the valve stem from the hot exhaust gases and also provides more area to assist in cooling the exhaust valve.

Valve cages. To facilitate grinding and reconditioning of valves and their seats, some large, low-speed engines have their entire valve assemblies mounted in cages. The valve cage consists of a hollow cylindrical casting containing the guide for the valve stem. The valve is seated on the lower edge of the cage and the valve spring is mounted on the top. The valve cage fits into a hole bored through an inlet or exhaust passage in the cylinder head, and is mounted with the head of the valve flush with the bottom surface of the head. Openings are provided in the walls of the cage which communicate with the corresponding passages in the cylinder head.

8–7. Valve springs. A valve spring serves to close the valve. Valve springs used on diesel engines are made of round steel wire, wound in a helical coil of cylindrical shape. Springs of this type have a force which is directly proportional to the amount they are compressed. Only a small portion of the maximum valve spring force is necessary to keep the valve tight on its seat. The principal duty of the valve spring, as mentioned before, is to provide sufficient force during the valve lifting process to overcome the inertia forces of the valve gear and keep it in contact with the cam without bouncing; this is obtained by putting the spring under compression when the valve is installed. When the valve is opened, this force is increased by the additional compression of the spring.

The space available for the valve spring is limited and therefore it is not easy to design and construct a spring which will exert the necessary force, and still will not break under the constantly repeated change of stresses. A spring made of wire of a small diameter is subjected to less stress than one made of wire of a larger diameter, but it also has a smaller force. In some engines, therefore, two or more valve springs of small wire diameter are used to operate the valve gear.

Spring surge. Another factor which influences the operation of valve springs is the vibration of the spring. This is discussed in detail in Sec. 17–6.

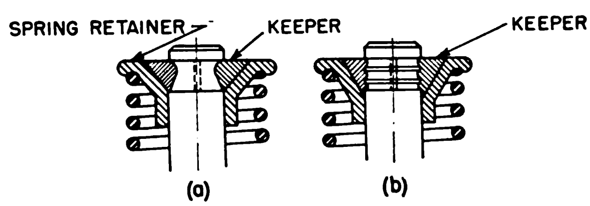


Figure 8–8. Valve-spring retainers.

Spring retainers. Valve springs are mounted between supports located at their ends and are known as *spring seats.* The lower spring seat may be simply a recess in the top of the cylinder head or valve cage, or it may be a steel washer which rests on top of the cylinder head and is shaped to fit the bottom coil of the spring. The upper spring seat, called the *spring retainer*, is a steel washer shaped to fit the top of the spring and attached to the top of the valve stem by some type of removable fastening.

The most widely used type of spring retainer is provided with a conical recess in the upper seat in which the valve stem is locked by means of a conical split collar, called a *lock* or *keeper*. This collar fits around the stem and into one, Fig. 8–8a, or more grooves, Fig. 8–8b, turned in the valve stem.

The pressure of the spring on this retainer tends to hold the locks tightly in place; yet they may be removed easily by depressing the spring while holding the valve in its closed position. There are two advantages to this type of retainer and keeper: first, they will not loosen in service, and second, they give a strong fastening which does not result in weakening of the valve stem, which frequently occurs when other types of fastenings are used.

Some large diesel engines use valve spring retainers held in place by a lock nut which screws into threads provided on the upper end of the valve stem.

8–8. Valve lash and adjustment. The expansion of the valve stem and of other parts of the valve gear when the engine heats up, has a tendency to hold the valves off their seats. Some provision, therefore, must be made in the valve gear to take care of this condition. The most common method used to permit this expansion is to provide a clearance or *lash* between the top of the valve stem and the valve-lifting mechanism. The proper lash is determined at the factory and is given in the engine instruction book. It is important that the valve lash specified for the valve be maintained. Too much lash will cause noisy operation and excessive wear. It will also result in improper valve timing, since the valve will open later and close sooner than it does with the proper lash. Too little lash is even more serious since it may prevent the valve from seating properly. This will result in valve leakage and burning of the valve-seating surfaces, and may even prevent combustion, due to loss of compression. All engines are provided with means for adjusting this lash or clearance in the valve gear somewhere between the cam follower and the valve stem. In most engines this adjustment consists of an adjustable screw and lock nut located at one end of the valve rocker arm. The clearance is measured directly by means of a feeler gauge inserted between the tip of the valve stem and the rocker-arm roller.

Automatic valve-lash adjusters are used to avoid the necessity of a clearance between the cam and the follower regardless of whether the engine is cold or warm and, by insuring a constant contact between the cam and the follower, to eliminate shock action at the beginning of the valve opening. They also eliminate the need for manual adjustment in order to take care of the wear at various points of the valve gear. At present two types of automatic adjusters, mechanical and hydraulic, are in use.

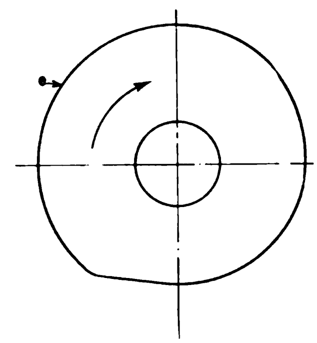


Figure 8–9. Cam of a mechanical lash adjuster.

Mechanical adjuster. A mechanical valve-lash adjuster is incorporated in the valve bridge, Fig. 8–6. A spiral spring has a tendency to turn the cam as shown by the arrow in Fig. 8–9, when the cylindrical spring *s*, Fig. 8–6, pushes the bridge to its highest position. Spring *s* takes up any clearance between the various parts of the valve-actuating gear, and the turning of the cams *e* takes up the clearance between

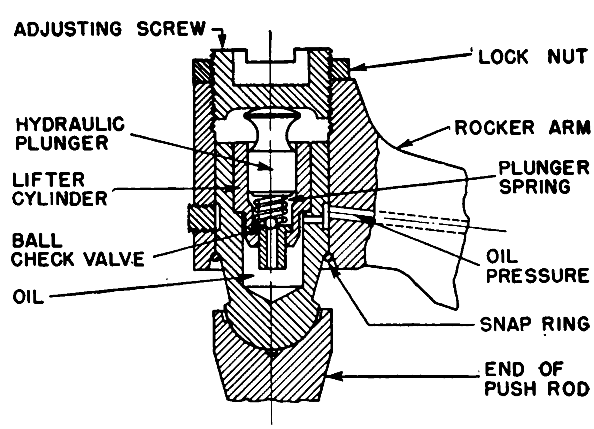


Figure 8–10. Hydraulic lash adjuster.

the ends of the bridge and the upper spring seats which are solidly connected to the valve stem.

Hydraulic adjuster. Hydraulic lash adjusters may be built into the valve tappets, but with valve-in-head engines they are generally built into the end of the rocker arms or the valve bridges which operate directly on the ends of the valve stems. Such an adjuster is shown in Fig. 8–10. It consists essentially of a small cylinder, referred to as the lifter cylinder, containing a piston or plunger, a spring, and a ball check valve. It is interposed between the push rod and the rocker arm end. In operation, oil under pressure from the lubricating oil system enters the lifter cylinder past the ball check valve and is trapped under the plunger. Any force exerted against the outer end of the plunger will be transmitted to the cylinder, mounted in the valve gear, by the entrapped oil. The valve is thus actuated just as if the lash were taken up mechanically. Since the spring inside of the cylinder acts to force the plunger outward, any clearance between the valve and its lifter will be taken up, and the oil pressure will immediately fill up the lifter cylinder through the check valve. If the valve stem expands, there is sufficient leakage of oil past the plunger to permit it to move in slowly so there is no danger of holding the valve open.

8–9. Questions. 1. What is meant by valve gear in a diesel engine?

2. Enumerate the kinds of diesel engine valves operated by cams.

3. Enumerate the types of camshaft drives used in Navy diesel engines.

4. Enumerate the types of cam followers used in Navy diesel engines.

5. What is the purpose of a rocker arm in valve gear?

6. Explain why two-stroke diesel engines with exhaust valves use two or more such valves.

7. Of what material are exhaust valves made?

8. What is the purpose of valve-seat inserts?

9. What is the purpose of valve guides?

10. What is the purpose of valve springs?

11. What is valve lash?

12. What types of automatic valve-lash adjusters are in use?

CHAPTER 9

FUEL INJECTION

9–1. Requirements. The fuel injection system, in delivering the fuel to the combustion chamber, must fulfill the following main requirements:

1. Meter or measure the correct quantity of fuel injected.
2. Time the fuel injection.
3. Control the rate of fuel injection.
4. Atomize or break up the fuel into fine particles according to the type of combustion chamber.
5. Properly distribute the fuel in the combustion chamber.

There are two different methods of fuel injection: *air injection* and *mechanical* or *solid injection*. Solid injection can be subdivided into three distinctly different groups: (1) *common-rail*, (2) *individual-pump*, and (3) *distributor* systems. The means of fulfilling the five requirements listed above vary according to the system used, and are shown in the following table:

Requirement	*Air-Injection System*	*Mechanical Injection Systems*		
		Common-Rail	*Individual Pump*	*Distributor*
Metering	Pump	Injection valve	Pump	Pump.
Timing	Fuel cam	Fuel cam	Pump cam	Fuel cam.
Injection rate	Spray valve	Injection valve	Pump cam	Fuel cam.
Atomization	Spray valve	Injector tip	Injector	Injector.
Distribution	Spray valve	Injector tip	Injector	Injector.

Metering. Accurate metering or measuring of the fuel means that, for the same fuel control setting, exactly the same quantity of fuel must be delivered to each cylinder for each power stroke of the engine. Only in this manner can the engine operate at uniform speed with a uniform power output.

Timing. Proper timing means beginning the fuel injection at the required moment, and is essential in order to obtain the maximum power from the fuel air mixture, and thus insure fuel economy and clean burning. When the fuel is injected too early in the cycle, ignition may be delayed because the temperature of the air at this point is not high enough. An excessive delay, on the other hand, gives rough and noisy operation of the engine and also permits some fuel to be lost due to wetting of the cylinder walls and piston head. This in turn results in poor fuel economy, high exhaust gas temperature, and smoke in the exhaust. When fuel is injected too late in the cycle, all the fuel will not be burned until the piston has traveled well past top center. When this happens, the engine will not develop its maximum power, the exhaust will be smoky, and the fuel consumption will be high.

Rate of fuel injection is important for the same reason that correct timing is important. If the start of the injection is correct, but the rate of injection is too high, the results will be similar to an excessively early injection; if the rate is too low, the results will be similar to an excessively late injection.

Atomization of the fuel must be according to the type of combustion chamber in use. Some chambers require very fine atomization, others can operate with coarser atomization. Proper atomization facilitates the starting of the burning process, and insures that each minute particle of the fuel will be surrounded by particles of oxygen with which it can combine. A more detailed discussion of atomization is given in Sec. 9-3.

Distribution. Proper distribution of the fuel must be obtained in order that the fuel will penetrate to all parts of the combustion chamber where oxygen is available for combustion. If the fuel is not properly distributed, some of the available oxygen will not be utilized, and the power output of the engine will be low.

Additional requirements. In order to be practical, the fuel injection system, and especially the high-pressure pump, must have additional features; it must:

1. Maintain its adjustment for a reasonable period of time, not lose it due to the vibrations connected with the high engine speed.

2. Be economical of power.
3. Be light and not too bulky, especially in small engines.
4. Be quiet in operation.

9-2. Air injection. Air injection was used in early diesel engines; at present it is seldom used and only for large engines operating on heavy viscous fuels.

In air-injection engines, the potential energy of the compressed air is converted into kinetic energy, folowing the principle explained in Sec. 2-6. This kinetic energy of the expanding air is used to feed the fuel into the cylinder from the spray valve, to atomize the fuel, and to create turbulence in the combustion chamber for mixing the fuel and air.

The air-injection system consists of three main elements:

1. The fuel pump for metering the fuel.
2. The compressor for supplying the injection air.
3. The spray valve.

Fuel pump. The type generally used has a plunger for each cylinder of the engine, and the quantity of fuel is controlled by varying the effective length of the plunger stroke. The only function of the pump is to meter accurately the required quantity of fuel and to deliver it to the spray valve.

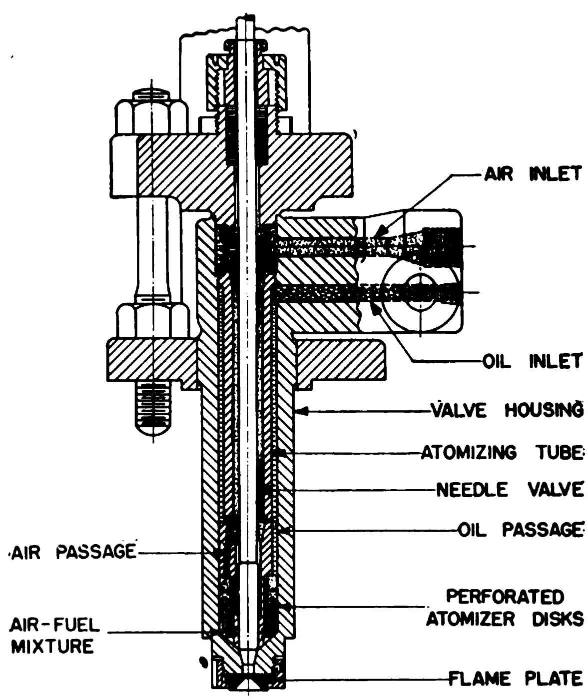


Figure 9-1. Nordberg air-injection fuel nozzle.

The spray valve, Fig. 9-1, consists of a *needle valve* held on its seat by a heavy spring, *atomizer disks* with holes to break up the fuel and mix it with the injection air as both flow through the valve, and a *flame plate* with an orifice through which the fuel-air mixture is admitted to the combustion chamber. The needle valve is lifted mechanically by a lever actuated by a cam on the camshaft.

The timing and duration of injection are controlled by the fuel cam. The rate of injection, the atomization of the fuel, and the distribution in the combustion chamber are all controlled by the number and size of the orifices in the atomizer disks and in the flame plate, and by the injection-air pressure. Injection-air pressures vary from 750 to 1,200 psi., the usual pressure being about 1,000 psi. Air injection fulfills the five main requirements adequately, but has the following disadvantages: the compressor absorbs a comparatively large amount of power, and is rather heavy and bulky, not suitable for high-speed, small-bore engines.

9-3. Mechanical injection systems. *Classification.* As already mentioned, all mechanical-injection systems may be subdivided into three main groups: *common-rail*, *individual-pump*, and *distributor* systems. The differences in construction in engines of different manufacturers are such that it may help in their discussion to subdivide further the main groups. The common-rail system may be divided into the basic system and a modification, such as that used in Cooper-Bessemer engines. The individual-pump system may be divided into the original system, with a separate pump and fuel injector for each cylinder, and a modification, in which the pump and injector are combined in one unit.

Atomization is a term applied to the breaking up of the fuel stream into mist-like sprays. With mechanical injection, atomization is obtained as follows: the liquid fuel, subjected to a high pressure, passes through a small opening into the combustion space filled with air, the pressure of which is considerably lower. As a result, the fuel streams develop high velocity and this creates great friction between the liquid stream and the air in the combustion space. Due to this friction, minute particles of fuel are broken off the surface of the stream, then the freshly exposed particles are again broken off, and so on, until the whole liquid stream is broken up into very small particles or globules. Atomization literally means breaking up into atoms. Actually, each globule is not one atom, but consists of many atoms. Therefore, the term atomization, while an exaggeration, gives a general idea of the aim of the process.

Penetration. In mechanical injection the distribution of the fuel in the combustion chamber, generally speaking, is obtained by two means: *penetration* and *air turbulence.* Penetration is the distance through which fuel particles are carried by the kinetic energy imparted to them when they leave the fuel nozzle. Friction between the fuel and the air in the combustion space gradually absorbs this energy. Penetration depends upon various characteristics of the fuel-injection system, chiefly upon the injection pressure and the size of the nozzle hole, and is reduced by finer atomization. Thus, the best conditions are obtained by a compromise between a minimum necessary penetration and the desirable fineness of atomization.

Air turbulence, discussed in Sec. 6-5, is independent of the fuel injection system and is only an additional tool in obtaining good combustion.

9–4. Common-rail system. This system consists of a high-pressure, constant-stroke and constant-delivery pump which discharges into a common rail, or header, to which each fuel injector is connected by tubing. A spring-loaded by-pass valve on the header maintains a constant pressure in the system, returning all excess oil to the fuel supply tank. The fuel injectors are operated mechanically, and the amount of oil injected into the cylinder at each power stroke is controlled by the lift of the fuel-admission valve. The operation of the injection system is shown diagrammatically in Fig. 9-2: the fuel cam gives an upward motion to the push rod; through the rocker arm and intermediate lever this motion is transmitted to the needle valve; the space above the needle valve

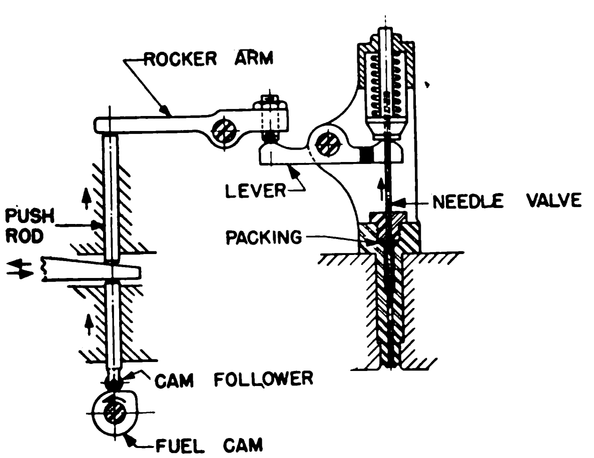


Figure 9–2. Control of fuel admission in a common-rail system.

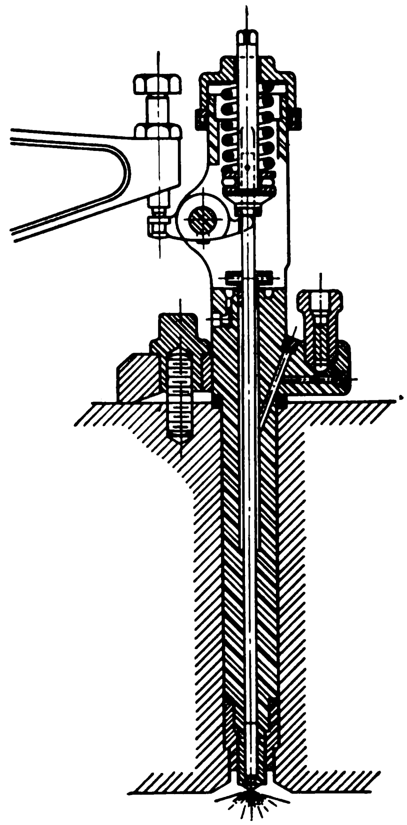


Figure 9-3.
Winton injection nozzle for common-rail fuel system.

seat is connected at all times with the fuel header and sealed from the top by a packing gland, Fig. 9-3. When the needle valve is lifted from its seat, the fuel is admitted to the combustion space through the small holes drilled in the injector tip, below the valve seat. Passing through these tiny holes, the fuel is divided into small streams which are broken up or atomized as previously explained. The amount of fuel injected is controlled in accordance with the power requirement by means of a wedge, Fig. 9-2, which changes the lash, Sec. 8-8, of the fuel valve. When the wedge is pushed to the right, the valve lash is decreased. The motion of the cam follower will then be transmitted earlier to the push rod, the fuel needle will be opened earlier, closed later, and its lift will be slightly greater. Therefore, more fuel will be admitted per cycle. When the wedge is pulled out, to the left, the valve lash is increased. The needle valve will be lifted later and closed earlier, and less fuel will be admitted. The position of the control wedge is changed either by the governor or, in variable-speed engines, by hand. The fuel-injection pressure is adjusted to suit the operating conditions by changing the spring pressure in the by-pass valve, called *fuel-pressure regulator*. Fuel-injection pressures of from 3,200 psi. to about 5,000 psi. are used at rated load and speed, depending upon the type of engine. In order to reduce the pressure fluctuations in the system, due to the intermittent fuel discharge from the pumps and withdrawals by the fuel valves, the volume of the fuel in the system is increased by attaching to the fuel header an additional fuel container called the *accumulator* which has a relatively large capacity. The area past the needle-valve seat and through the passage between the valve seat and the valve tip containing the orifices, is several times as large as the area through the orifices on the tip. The control of the fuel jets is largely by the orifices. The valve tip commonly used with this system has several holes or orifices.

The common-rail system is not suitable for high-speed, small-bore engines, because it is difficult to control accurately the small quantities of fuel injected into each cylinder at each power stroke.

Cooper-Bessemer system. This system differs from the original common-rail system chiefly in that the functions of the fuel needle valve are divided between two separate pieces of equipment, the fuel *injector* and the injection *nozzle*, as shown schematically in Fig. 9-4. Another difference lies in the fact that the pressure regulation is accomplished by the high-pressure pump itself.

The pump plunger, on its downward stroke, first closes small holes that connect the pump barrel with the fuel admission line. A further downward motion increases the oil pressure in the pump until it opens the spring-loaded discharge valve and delivers the oil into the injection system. During the return stroke the spring moves the plunger upward; this creates a vacuum and when the plunger uncovers the holes on top, oil from the suction side enters into the pump. The oil from the fuel-oil pressure tank on its way to the suction side of the pump is admitted first to the inner side of a sleeve. This sleeve and a second sleeve surrounding it have two mating holes. By turning the sleeves, one relative to the other and to the housing, the amount of fuel admitted to the pump is adjusted to meet the load and speed requirements. The outer sleeve is set and turned by the governor so as to admit the amount of fuel corresponding to the load carried by the engine. The inner sleeve is turned by a mechanism set to maintain a prescribed constant pressure in the system. If the pressure goes up, the sleeve is turned to decrease the effective area of the opening between the two sleeves. The amount of fuel taken by the pump is thus reduced, and as a result the pressure in the system goes down. On the other hand, when the pressure begins to drop, the sleeve is turned in the opposite direction, the effective opening area is increased, more fuel goes into the pump, and the pressure goes up. The injection nozzle consists of a spring-loaded plunger with a conical end which acts as a valve. It is raised from its seat by the oil pressure when the valve in the fuel injector is opened, and is returned to its seat by the spring in the upper part of the nozzle when the injector valve is closed and the oil pressure between this valve and the nozzle begins to drop. A quick closing of the injection nozzle and elimination of after-dribbling of the fuel into the combustion space is obtained as follows: the lifter plunger is drilled lengthwise at its center from the valve and to a point in line with the recess in the injector body, Fig. 9-4. Another hole, drilled at a right angle to the central hole, connects with it, forming a passage from the lifter end to the recess and through it to the drain tank. The bottom of the injector valve is lapped to a seat with the end of the lifter plunger so that when the two are brought in contact during injection, the passage through the plunger is sealed. As soon as the fuel cam releases the lifter plunger, the valve is closed by its spring, Fig. 9-4. The oil pressure on the end of the lifter plunger will move it downward, and a small amount of fuel oil is spilled to the drain tank, relieving the oil pressure in the nozzle. The lifter spring will then return the lifter plunger to a contact with the valve. This arrangement also acts as a safety feature which prevents passage of the fuel oil into the engine cylinder except when necessary, even if the injector valve should leak at its seat. The system gives good results but, like any common-rail system, is not suitable for small-bore engines.

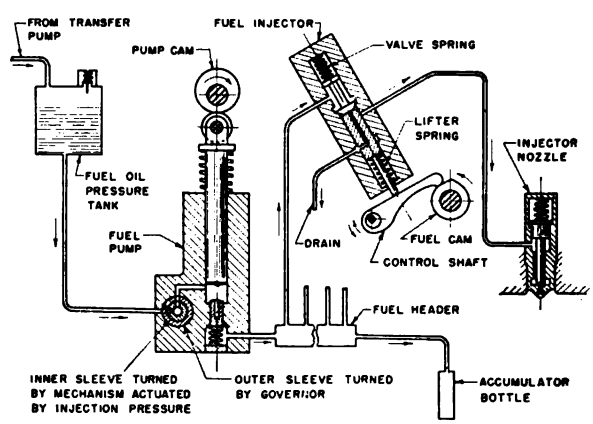


Figure 9–4. Cooper-Bessemer fuel-injection system.

9–5. Features of pump-injection system. This system, also known as the jerk-pump system, has two essential parts to each cylinder, the *injection pump* and the *fuel nozzle*. The requirements which a pump in this system must fulfill, both in respect to metering and timing, are such that they can be met only by a precision piece of equipment.

Accuracy. In operation on a high-speed diesel engine, these injection pumps must measure and deliver, under high pressure and at exactly the required time, an exceedingly small quantity of fuel.

Metering. The volume of the fuel injected is extremely small compared with the piston displacement. At full load, the volume of the fuel injected is about 1/20,000 of the piston displacment, and when the engine is idling, the volume of fuel required is only about 1/5 of that at full load; this gives a volume ratio of about 1/100,000.

> *Example 9–1*. Find the volume of fuel injected in a 3⅝ in. × 4½ in. engine. Its displacement is $\pi \times (3.625)^2 \times 4.5 = 46.5$ cu. in. The amount of fuel necessary per injection varies from $46.5 \div 20{,}000 = 0.00233$ cu. in. down to 0.0047 cu. in. These quantities are equal to a spherical drop from ⅛ in. diameter and down to 1/14 in.

The difficulty of accurately measuring such small quantities of fuel, so that all cylinders receive exactly the same amount, is quite evident.

Timing. An idea of the accuracy of timing required may be obtained from the following considerations: Ordinarily the period during which the fuel is injected does not exceed 20° of crank travel; if the engine speed is 2,000 rpm. the whole injection period corresponds approximately to

$$60 \times (20/360) \div 2{,}000 = 1/600 \text{ sec.}$$

The start and end of injection cannot vary more than approximately 1° of crankshaft rotation, or must occur within limits of about 1/12,000 sec.

Injection pressures. Such minute quantities must be injected within such close time limits under pressures which may run as high as 6,000 psi., and in some injector types even up to 30,000 psi.; most high-speed engines, however, use injection pressures of 2,400 to 3,000 psi.

Precision workmanship. All injection fuel pumps of the jerk-pump type have plungers fitted closely to the pump barrels by lapping. Lapping means finishing hardened surfaces by working them against the surfaces of laps with an exceedingly fine abrasive material. From a number of plungers and barrels which have been lapped truly round and cylindrical but differ slightly in diameter, the plungers are fitted to their barrels by selective assembly. In this way a fit is obtained with a clearance less than 0.0001 in. Such a fit gives very little leakage, even with the high pressures created, and no packing of any kind is necessary. Due to the method of fitting a plunger to its barrel, these parts are not interchangeable. If a plunger or barrel is worn or damaged, *both* pieces must be replaced.

Pressure waves. Fuel oil, like all liquids, is compressible. Therefore, when the pump plunger at the beginning of the actual delivery strikes the oil in the pump barrel, the oil is not accelerated at once in the whole fuel line. The motion of the plunger increases the pressure first in the particles of the oil nearest to the plunger. This pressure increase is transmitted gradually through the line until it reaches the nozzle. On the other hand, due to its inertia, the liquid column in the line has a tendency to move away from the plunger. Thus, the initial blow of the plunger sets up a compressive wave in the fuel line. When this wave reaches the nozzle which presents a certain resistance, it is reflected and travels back to the plunger, increasing the pressure created by the plunger; after it reaches the plunger, it returns to the nozzle, etc. This fluctuation of the pressure at the discharge end of the fuel line disturbs the fuel discharge through the nozzle. During the moment when the pressure at the nozzle is low, the fuel discharge is decreased and the atomization becomes poorer. The disturbances are particularly noticeable in engines operating at variable speed. In addition, the pressure waves may produce vibration of the tubing connecting the pump and the nozzle, and cause its breakage. The building up of pressure waves is affected by many factors, the chief ones being the inside diameter and length of the fuel-line tubing. The proper length and diameter are determined by the engine builders and should never be changed as a change may cause serious trouble.

9–6. Jerk pump. In order to obtain proper atomization of the fuel spray, the injection pressure in the fuel line must consistently be maintained sufficiently high, from the very start to the end of the injection. Also, since this pressure is proportional to the plunger speed, the latter, too, must be reasonably high during the whole injection period. This high pressure is obtained by using for the fuel delivery only part of the plunger stroke, after the plunger has acquired a certain speed, and discarding the initial part of the stroke. This method produces a sudden acceleration of the fuel in the line, causing a *jerk*, hence the name for the pump.

Metering. In most jerk pumps, the total plunger stroke is constant and the metering is controlled by varying the length of the effective part of the plunger stroke by one of the following methods:

1. The fuel is admitted into the pump barrel through *ports* in the barrel controlled by a spiral groove or scroll, also called the helix, on the *plunger.* The plunger can be turned in the barrel while moving back and forth, and this changes the portion of the plunger stroke during which the ports are covered and the fuel is delivered to the nozzle.

2. The fuel is admitted into the pump barrel through *ports* in the barrel which are *controlled by a separate valve.*

Plunger-controlled pumps. The principle of this type of pump can be better understood by referring to Fig. 9–6. At the bottom of the plunger stroke, Fig. 9–6a, the suction and pressure release ports are both in communication with the inner pump space. When the plunger has moved a certain distance and covered both ports, fuel delivery begins with a jerk, and lasts until the lower edge of the spiral begins to uncover the release port, Fig. 9–6b. At that point the pressure drops and fuel delivery stops. The plunger continues to travel a short distance to the top of its stroke and then begins to move downward. If the plunger is turned about 60°, Fig. 9–6c, the distance between the top edge of the plunger and the edge of the spiral sliding over the release port is shorter, and the fuel delivery stops earlier. Finally, if the plunger is turned 90° more, the release port stays uncovered, and no fuel is delivered.

Timing. In the operation as shown in Fig. 9–6, the beginning of the effective stroke always occurs at the same moment, but the end of the injection changes with the engine load. It is later with a higher load, greater fuel delivery, and earlier with a lower load, smaller fuel delivery. The middle of the injection stroke is advanced with a load increase.

This condition can be reversed by making a spiral edge on the top of the plunger and a square groove at its bottom. In this case the beginning of the injection

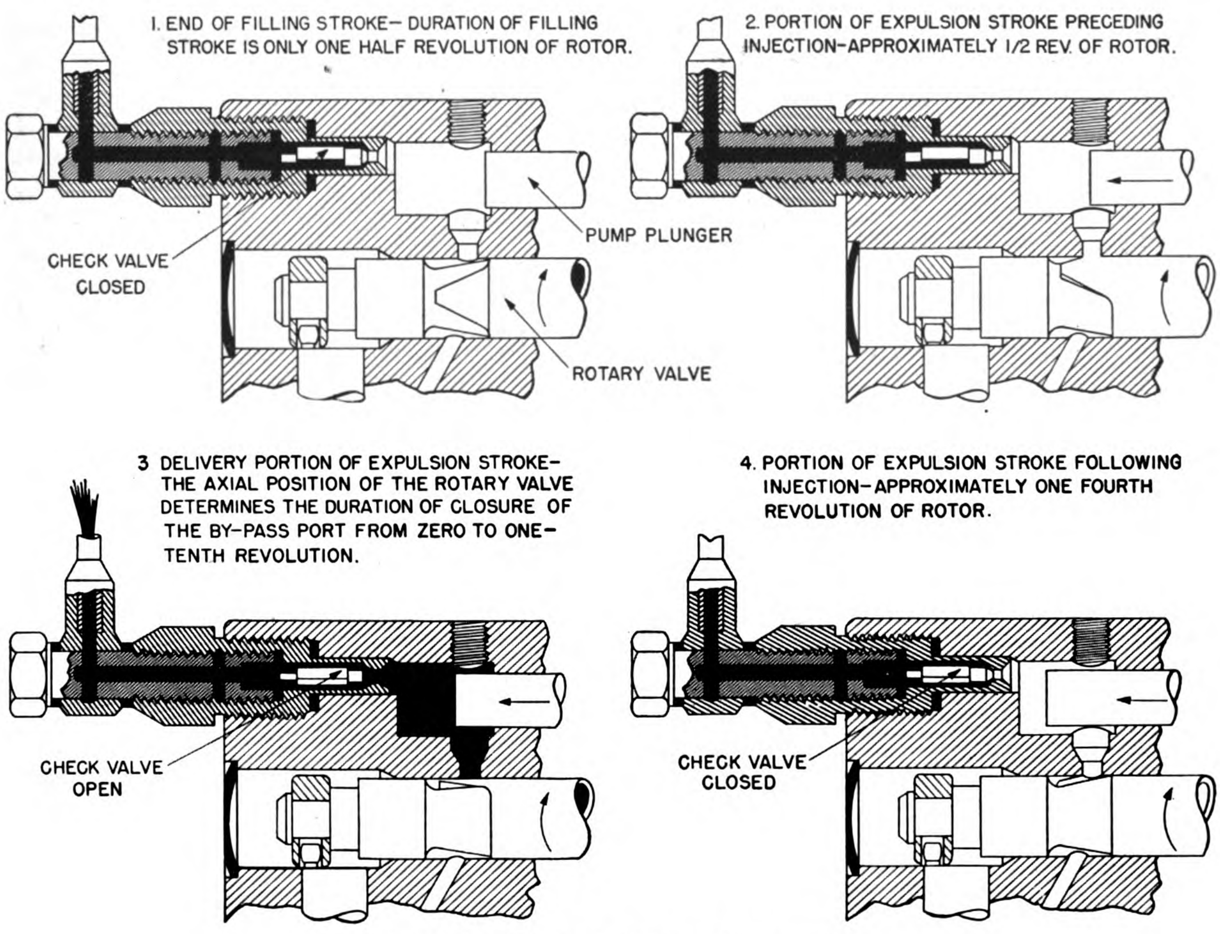


Figure 9–5. Jerk pump with rotary by-pass valve.

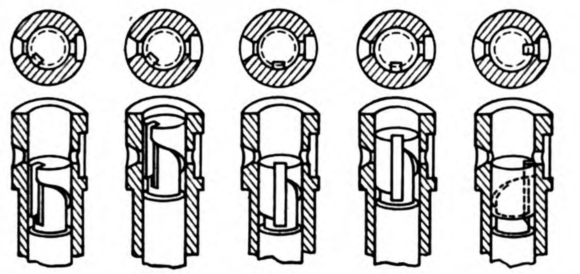


Figure 9–6. Barrel with various plunger positions.

will be earlier with an increase in the load and later with a decrease in it, while the end will be always constant. The middle of the injection will be retarded with an increase of the load, a condition, which however, is not desirable. By making both edges spiral, inclined in opposite directions, the middle of the injection stroke can be kept constant for all loads.

Naturally, the compressibility of the fuel and the mechanical flexibility in the pump mechanism will retard the actual start of injection into the cylinder, producing an *injection lag*, which was described in Sec. 6–5.

Valve-controlled pumps. Fig. 9–5 may serve to explain the principle of such a pump. The pump plunger has a constant stroke and is a plain cylinder. The fuel delivery is controlled by the rotary valve with a wedge-shaped end. When the plunger, travelling to the left, approaches its middle position, the rotary valve closes the admission port, having moved the slanting fuel-cut-in edge across it. When, due to rotation of the valve, the other slanting edge reaches the port, the inside of the pump comes in communication with the low-pressure space, the pressure is released, and fuel delivery is cut off. The rotary valve can be moved axially, by the governor or by hand, and this changes the distance from the cut-in to the cut-off slanting edge and with it the duration of the fuel-injection period.

Pump drives. The plunger of plunger-controlled pumps is pushed by a cam with a cam follower during the delivery stroke and returned by a spring. The pumps are built either as separate units for each engine cylinder, or as a multi-plunger, in-line unit for all cylinders.

The pumps with a rotary valve are conveniently built around this valve, and the axis of the individual cylinders, instead of being in one plane, are at equal distances from the axis of the driving shaft and are equally spaced around it and the centrally located rotary valve. Instead of cams, a *swash plate*, also called a *wobble plate*, is used as the driving member, Fig. 9–7, and is keyed to the drive shaft. By the use of shoe-plates, containing spherically shaped sockets, the rotary motion of the swash plate is converted into a straight-line back-and-forth motion which is transmitted to the ball-ended tappets. Each tappet operates one pump plunger. The plungers and tappets are returned by springs, Fig. 9–7. The drive shaft also turns the rotary valve which is located on the same axis as the shaft. The main body of the pump is symmetrical with the drive shaft, and it can be conveniently flange-bolted to the timing-gear housing of the engine.

9–7. Fuel nozzles. The nozzles are either of the *open* or of the *closed* type. The open type usually is a simple spray nozzle with a check valve which prevents the high-pressure gases in the engine cylinder from passing to the pump. It is simple, but as it does not give proper atomization, is not generally used. The closed-type nozzle is used more commonly. Basically, it is a hydraulically operated, spring-loaded needle valve. Most closed nozzles open inward under the pressure acting on the differential area of the needle valve, and are seated by a spring when the pressure is cut off, Fig. 9–8. The larger cylindrical part of the valve has a lapped fit with the nozzle body. There are two main types of such nozzles: the *pintle-type* and the *hole-type* nozzle. The valve of the pintle nozzle, Fig. 9–9a and 9b, is provided with a pin or pintle protruding from the hole in the bottom of the nozzle in which there is a close fit. The fuel delivered by such a nozzle must pass through an annular or ring-shaped orifice. The spray is in the form of a hollow cone, whose outside angle, which may be any angle up to 60°, is determined by selecting certain dimensions. A valuable feature of the pintle

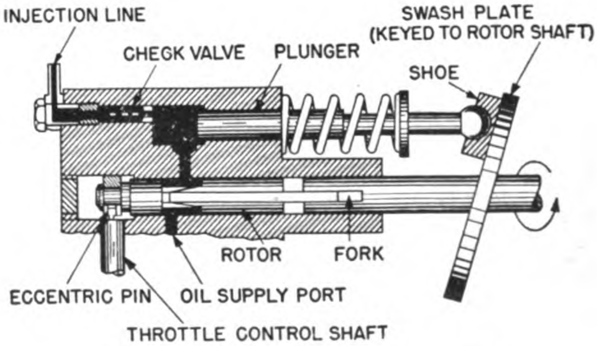


Figure 9–7. Section of Ex-Cell-O pump.

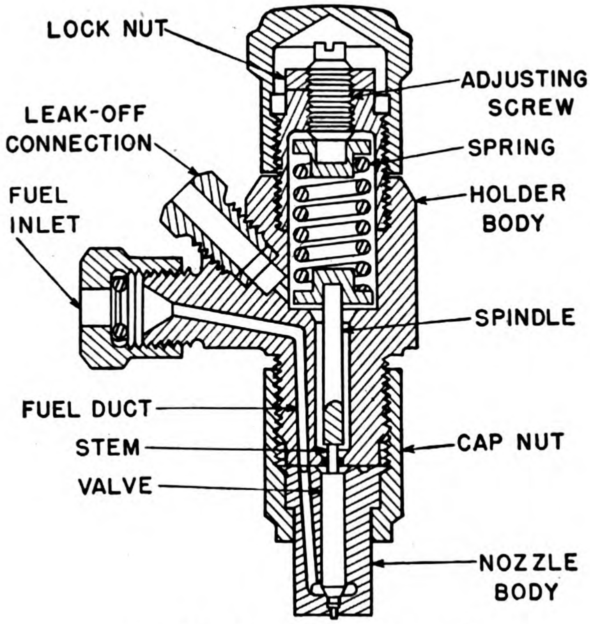


Figure 9–8. Bosch fuel-injection nozzle.

nozzle is its self-cleaning property, which prevents carbon deposits from building up in and around the orifice.

In the hole-type nozzle there may be one, Fig. 9–9c, or several spray orifices, Fig. 9–9d, in the form of straight, round holes drilled through the tip of the nozzle body below the valve seat. The spray from the single-hole nozzle is relatively dense and has a greater penetration. The general spray pattern of a multi-hole nozzle, which may or may not be symmetrical, is determined by the number, size, and arrangement of the holes. Orifices used are from 0.006 in. and up to 0.0225 in. in diameter, and their number may vary from 3 up to as many as 18 nozzles for large-bore engines. Multi-hole nozzles are used generally in engines with an undivided combustion chamber.

Fig. 9–8 shows a pintle nozzle assembled in a nozzle holder with the spring and connections for the fuel-pressure line and leakage drain. The pressure necessary to open the needle valve may vary from 1,500 psi. to about 3,000 psi.

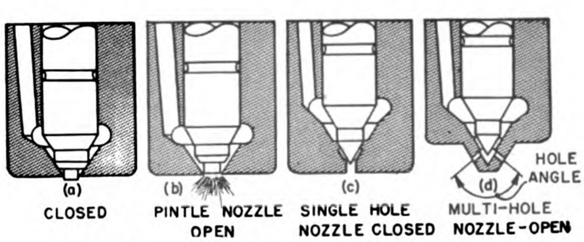


Figure 9–9. Types of closed nozzles.

Fig. 9–10 shows a hydraulically operated fuel nozzle which opens outward. It is of the pintle type, has an opening pressure of about 1,500 psi. to 2,000 psi. and produces a cone-shaped spray. The advantages of this type of nozzle are its compactness and light weight. These factors make it particularly suitable for small-bore high-rpm. engines.

9–8. Unit injector. The unit injector combines a pump and a fuel-spray nozzle in one unit, as shown in Fig. 9–11. The pump is of the jerk type with ports controlled by helical-grooved edges in the plunger. The amount of fuel is controlled by turning the plunger. The nozzle is of the open type with a spherical check valve. The spray tip has several small-diameter orifices. However, there are also unit injectors in existence which have closed-type nozzles with hydraulically operated differential needle valves and multi-hole nozzle tips.

The pump plunger receives its downward motion, the delivery stroke, from a fuel cam through a rocker arm which acts on the *plunger follower*. The plunger

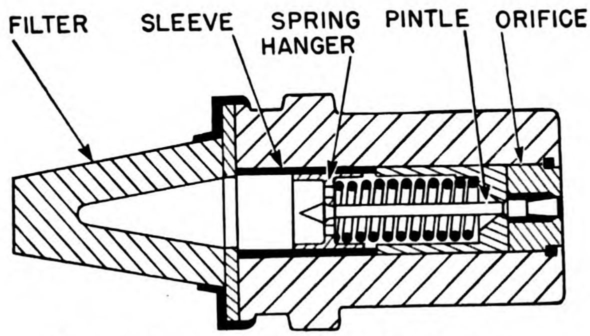


Figure 9-10. Section through Ex-Cell-O nozzle tip.

is returned by the plunger spring. The fuel, under a pressure of about 35 psi., is admitted through a small filter, Fig. 9–11, and fills the annular supply chamber around the pump plunger barrel, called a *bushing*. As the plunger moves downward, fuel is displaced into the supply chamber, at first through the lower port, and later, when the lower edge of the plunger closes this port, through a central and a transverse part or crosswise drilled holes in the plunger and the upper port. When the upper helical edge has covered the upper port, the fuel from the pump plunger barrel is forced down into the nozzle body, opening the spherical check valve, past the flat check valve, and into the spray tip. From there it is forced through the orifices into the cylinder. The fuel-injection pressure is raised approximately to 20,000 psi. as the fuel passes through the nozzle. Injection continues until the lower helix on the plunger uncovers the lower port in the plunger barrel. The fuel then begins to by-pass through the holes in the plunger and through the lower port into the supply chamber. This releases the pressure on the fuel in the plunger barrel, and the check valve spring causes the spherical check valve to seat. On the return stroke, the upward movement of the plunger fills the plunger barrel with fuel oil which flows from the supply chamber through the lower port. The function of the *flat check valve* is only to close the inside of the nozzle against gases from the cylinder.

Turning of the plunger in order to change the effective length of the stroke is accomplished by a *gear* and *rack* connected to the governor or hand throttle. The middle part of the plunger has a hexagonal cross section which slides through a corresponding hole in the gear, thus forcing the plunger to turn with the gear. The effective stroke is determined by the relative positions of the helices and the upper and lower ports.

The advantage of the unit-injector construction is in the absence of long fuel lines which cause pressure waves and sometimes mechanical troubles. However, one disadvantage lies in the high pressures created. Such pressures result in faster wear of the spray orifices and the necessity of dismantling a considerable part of the valve gear in order to take out one unit injector. Another disadvantage is the greater chance of fuel leaks into the engine sump, and dilution of the lubricating oil.

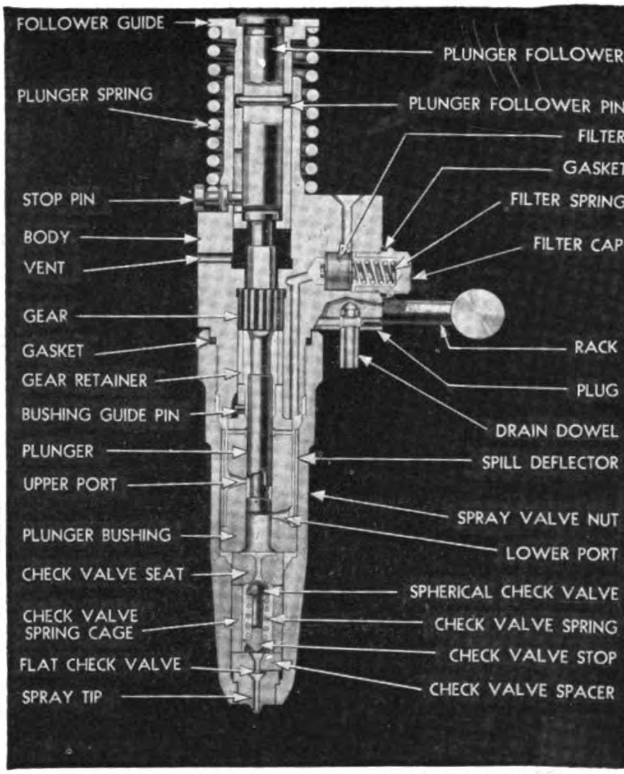


Figure 9-11. Unit injector with spherical check valve.

9–9. Distributor system. Of the several distributor systems used in different makes of engines, the only one in use at present in United States Navy engines is the Cummins system. Although essentially a distributor system, it has the characteristic features of a unit injector and is sometimes classified as such. In this system, fuel under 130 psi. to 150 psi. pressure is supplied by a gear transfer pump, as shown in the schematic drawing, Fig. 9–12, through an indexed rotating distributor to a metering plunger during its downward stroke. This plunger has a variable stroke, controlled by the governor, and receives its upward motion from a multi-lobe cam and the downward motion spring. During the upward stroke, this

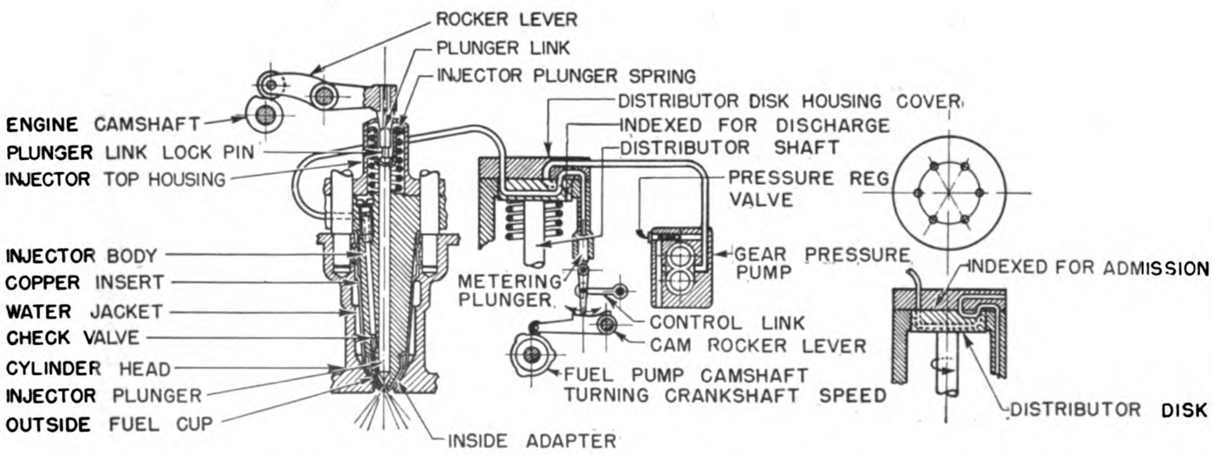


Figure 9–12. Diagram of Cummins distributor-type low-pressure fuel-injection system.

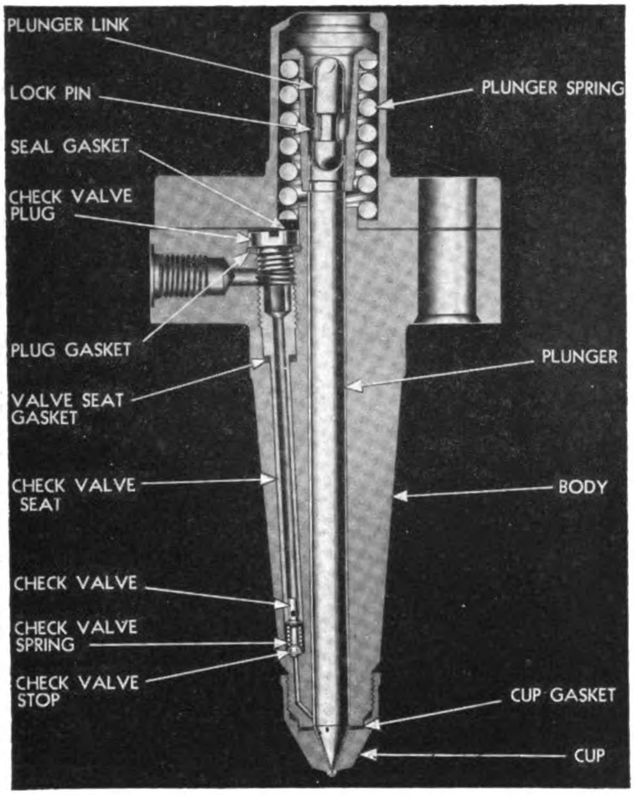


Figure 9–13. Cummins injector, cross section.

plunger sends the fuel through other passages in the same distributor to the individual injectors on each engine cylinder. The fuel enters at the top of the injector, Fig. 9–13, flows through an inlet passage past a spring-loaded *check valve*, and fills the chamber under the injector plunger which, except at the end of injection, is off its seat. The injector plunger is operated by a cam through a *rocker lever* and *link*. As the fuel is delivered from the distributor to the injector during the suction stroke of the engine, the injector plunger is gradually lifted and the fuel fills the space in the *cup* under the plunger. At the time of injection the plunger is pushed downward and the fuel prevented by the *check valve* from returning to the distributor, is injected into the combustion space. It is finely atomized by being forced under a relatively high pressure through six small holes, 0.006 to 0.008 in. in diameter, depending upon the size of the engine.

The advantage of this system is the absence of high-pressure lines and pressure waves. Its disadvantage is the relatively large inertia of moving parts, making it not suitable for very high speeds. Another disadvantage, more or less eliminated in the new-type injector, is dilution of the lubricating oil in the engine crankcase due to leaking of fuel oil past and up the injector plunger.

9–10. Problems. 1. Find the volume of fuel injected at each cycle into the cylinder of a 6x4¼x5x 1,800 two-stroke engine which uses 70 lbs. of fuel per hour. *Ans.* 0.00356 cu. in.

2. Find the diameter of a sphere whose volume is 0.00356 cu. in. *Ans.*, 0.1895 in.

3. Determine the duration of fuel injection in an engine running at 2,200 rpm., if the start of injection occurs at 15.5° b.t.c. and the end of the injection at 5.5° a.t.c. *Ans.* 0.00159 sec.

9–11. Questions. 1. What are the main requirements which a fuel-injection system must fulfill?

2. Enumerate the main methods used with solid injection.

3. Why is air injection not used with high-speed, small-bore diesel engines?

4. What is meant by the term atomization?

5. Why is the common-rail system not suitable for small-bore, high-speed, diesel engines?

6. What are the main parts in a pump-injection system?

7. What fuel-injection pumps are called jerk pumps?

8. What methods are used in jerk pumps to control the metering of the fuel?

9. What types of fuel nozzles are used with jerk pumps?

10. What is meant by a unit injector?

11. What is the main feature of a distributor system?

CHAPTER 10

BEARINGS AND BEARING LUBRICATION

10–1. General. Bearings are important parts in any machine. Proper understanding of the principles underlying their performance is essential to an engine operator who wants to maintain their satisfactory operation, prevent their failure and the resulting stoppage of the engine. The object of bearings is to support rotating shafts and other moving parts, and to transmit loads from one engine part to another.

In order to accomplish their main purpose, and operate without forced stopping and undue wear, bearings must fulfill two additional important functions:

1. Reduce the friction between moving surfaces by separating them with a film of lubricant.
2. Dissipate the heat produced by unavoidable friction.

Classification. All bearings may be divided into two groups: bearings for *rotary* motion, and bearings for *reciprocating* motion.

Bearings for rotary motion can be subdivided, from the standpoint of load, into:

1. *Journal bearings* in which the direction of the main load forms a right angle with the axis of rotation.
2. *Thrust bearings* in which the load acts *along* the axis of rotation.

From the standpoint of *construction* the bearings may be divided into bearings with a *sliding contact* or plain bearings, and bearings with a *rolling contact* in which either steel balls or steel rollers are interposed between the working surfaces.

10–2. Journal bearings. *Bearing loads.* In respect to the type of loading, journal bearings may be divided into two groups:

1. Bearings with a *steady* load, such as electric generators, motors or centrifugal pumps.
2. Bearings with a *fluctuating* load, such as main, crankpin, and wristpin bearings in diesel engines.

The first group of bearings is often called *power* bearings to differentiate them from the second group called *engine* bearings. From the point of view of construction there is practically no difference between the two bearing groups, but from the point of view of operation there is a great difference, since the duty of the engine bearings is much more severe than that of the power bearings.

Diesel engine bearing loads are a combination of two kinds of forces; *gas pressures* acting upon the engine piston, and *forces of inertia* (separate forces of inertia being created by reciprocating parts and by rotating ones). As explained in Sec. 16–3, the forces of inertia of reciprocating parts gradually change each cycle and their direction is reversed every time a piston reaches dead center. The centrifugal forces of inertia created by rotating parts do not change while the engine speed remains constant, but their direction changes with the position of the crank. Depending on the position of the piston, inertia forces at some points are added to, and at other points subtracted from the gas-pressure load. Therefore, with a multi-cylinder engine there may be cases when the bearing loads will reach a maximum at a certain engine speed and decrease both at lower and higher speeds.

Polar diagrams. A good picture of the working conditions of an engine bearing is obtained from a so-called polar diagram. Such a diagram is obtained as follows: first, the total bearing loads, coming from gas pressure, inertia forces of reciprocating parts, and inertia forces of rotating masses, are computed for every 5 or 10 crank-angle degrees over the whole cycle, 360° for a two-stroke and 720° for a four-stroke engine. Next, all of these loads are drawn to a certain scale, so many lbs. to 1 in., starting from one point, called the *pole;* each load is at the angle with the center line of the cylinder as found when calculating the loads. Finally, the ends of these forces are connected by a continuous curve. A sample of such a diagram drawn

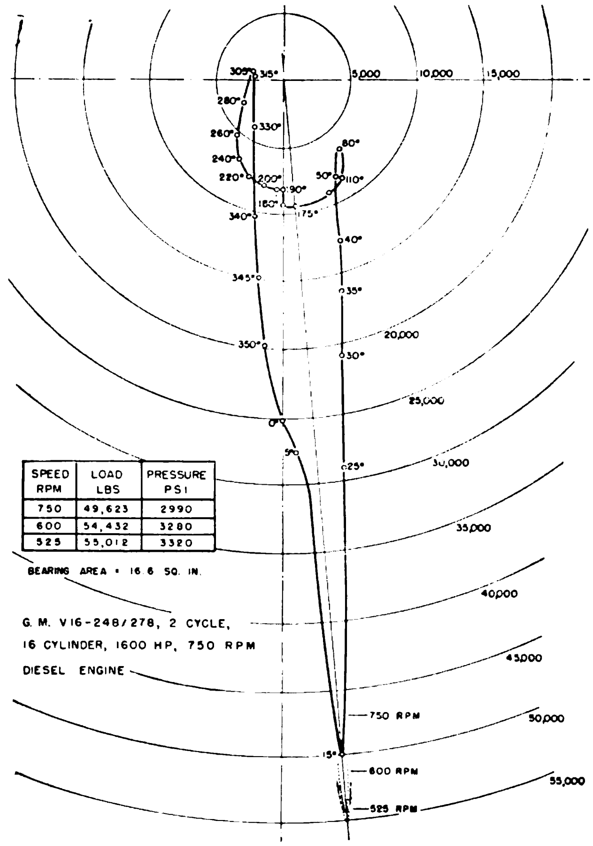


Figure 10-1. Connecting rod bearing loads.

for a connecting-rod bearing is shown in Fig. 10–1. At the same time this diagram and the data on it show that for this engine the bearing loads are highest, not at the normal operating speed of the engine, 750 rpm., but at the lower speed of 525 rpm. When an engine is forced to produce its rated power at reduced rpm., the bearing loads will always increase.

Bearing pressure. A factor by which bearing loads are measured is the ratio of the bearing load, in pounds, to the product of the length, and diameter of the bearing in inches. This factor is called bearing pressure and is expressed in psi.,

$$\text{bearing pressure} = \text{load} \div (\text{length} \times \text{diameter}) \qquad (10\text{–}1)$$

The actual pressure between a journal and its bearing varies around the circumference of the contact surfaces. The pressure determined by expression (10–1) is not the actual pressure but only a certain characteristic, helpful in dealing with bearings.

Operating conditions. Generally speaking, power bearings operate under loads which produce low bearing pressures, not exceeding 150 psi. to 200 psi. The maximum lubricating-oil temperature seldom exceeds 160° F, and the condition of the oil is excellent, not being contaminated or diluted by any foreign matter.

Diesel engine bearings, on the other hand, operate under considerably less favorable conditions: (1) The load fluctuates, the maximum bearing pressures may be as high as 3,000 psi., and mean bearing pressures seldom run under 600 psi. and often exceed 800 psi. (2) The temperature of the lubricating oil is high; in large high-output engines it may reach 180° F and in smaller high-speed engines, often runs up to 200-220° F, in some cases it may even reach 240° F. (3) The oil gradually becomes contaminated with products of oxidation, carbon, sludge formed by water condensed from the products of combustion, and worn-off metal particles, and diluted with fuel oil leaking from the injection system.

Bearing construction. Journal bearings having sliding contact are generally made in two halves, each half bored concentric so that when the two are brought together around the journal a true circle is formed. In the case of small journals, as for wrist pins, the bearings may take the form of a bushing that is slipped over the end of the journal. To replace worn bearing surfaces, bearings usually are equipped with bearing shells. Large power bearings and bearings of low-speed, large engines have cast-iron or cast-steel, or sometimes bronze shells with white metal or babbitt lining and shims of various thickness inserted between the two faces. The shims serve to adjust the clearance between the journal and the bearing which has a tendency to increase gradually due to wear. Fig. 10–2 shows such a bearing construction used as a main bearing supporting a crankshaft or a generator shaft. A similar bearing type with babbitted shells and shims is used for the large end of connecting rods of large diesel engines. High-speed diesel engines use precision bearings (briefly described in Sec. 7–4), both for their main bearings and connecting rods, an example being shown in Fig. 7–13a.

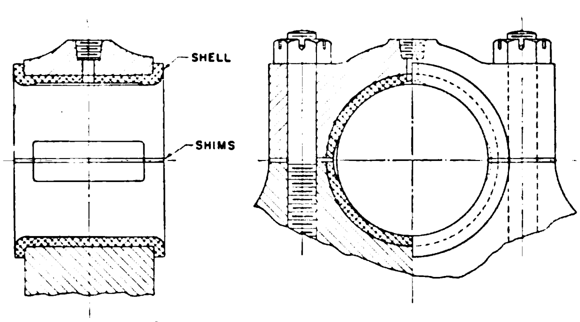


Figure 10–2. Plain journal bearing.

10–3. Bearings with rolling contact. *Advantages.* Properly designed and precision-manufactured bearings with rolling contact have the following advantages over plain bearings with a sliding contact:

1. They will maintain a comparatively accurate alignment over long periods of time.
2. They can carry heavy momentary overloads without failure or seizure.
3. Their power loss due to friction is small.
4. They are particularly adapted to variable-speed operation because their coefficient of friction is practically independent of speed.
5. For the same reason they have a low starting resistance.
6. Their lubrication is simple and requires but little attention.

Due to small power loss, ball and roller bearings are often called *anti-friction* bearings.

Disadvantages. Both ball bearings and roller bearings require precision-made outer and inner races which cannot be split into halves, and therefore can be used only on straight shafts either of constant diameter or with diameter decreasing toward the end of the shaft. Standard ball and roller bearings, therefore, cannot be used on crankshafts with several cranks. Another disadvantage is that their load capacity decreases with an increase of speed; finally, their cost is higher than that of plain bearings.

Ball bearings. Because extreme precision in manufacturing, which is so important for their satisfactory operation, is easier obtained with ball bearings than with roller bearings, ball bearings are more widely used. Ball bearings support the load on a series of

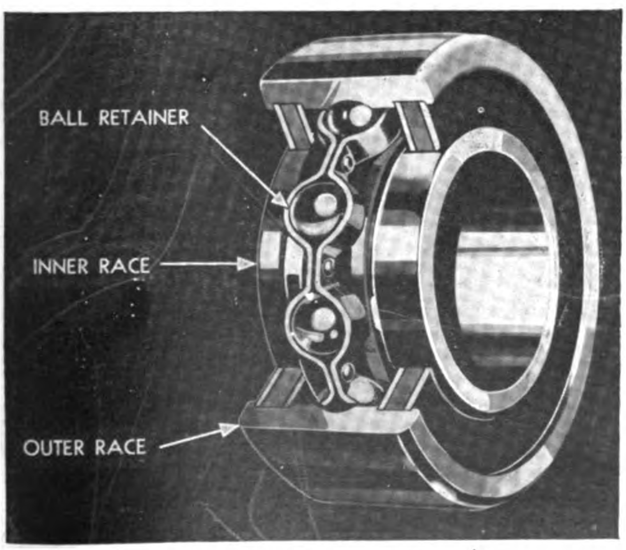


Figure 10–3. Ball bearing.

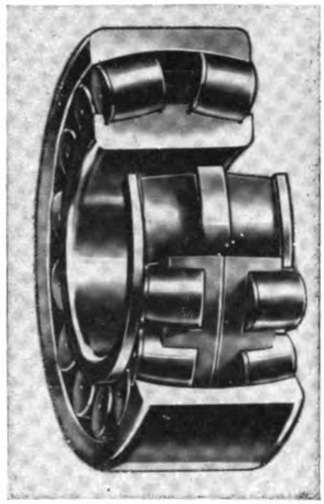


Figure 10-4. Roller bearing.

hardened steel balls, and for simplicity it is sometimes said, that all ball bearings have a *point* contact. Actually, however, because of elastic deformation, they perform with very small contact *areas.* Ball bearings are built to sizes which are standard with every manufacturer and are hence interchangeable. A ball bearing consists of four main elements, Fig. 10–3: (1) an *inner* ring, or *race,* grooved on its outer surface; (2) an *outer race* grooved on its inner surface; (3) steel *balls,* and (4) a *ball retainer* or *cage* for spacing the balls so that they do not touch each other, thus reducing loss of power, wear, and noise.

Roller bearings. Roller bearings are a development of ball bearings and have cylindrical, conical, or barrel-shaped rollers instead of balls. Many of them are interchangeable with standard ball bearings, i.e., have the same outside dimensions. Their load capacity, for the same outside dimensions, is greater than that of ball bearings, due to a larger contact area. However, they are more sensitive to misalignment. Fig. 10–4 shows a roller bearing with cylindrical rollers. Bearings with conical, often called *tapered rollers,* and correspondingly shaped races can be adjusted for axial clearance, and are usually known as Timken bearings, Fig. 10–5. Roller bearings using relatively long rollers of small diameter, 2 to 4 mm. or about 5/64 to 5/32 in., are called *needle bearings* and are used mostly as wrist-pin bearings in some diesel engines. Needle bearings do not use cages for spacing their rollers.

Friction. The total friction in ball and roller bearings is made up, in addition to rolling friction, by sliding friction, which is due to rubbing against ball and roller retainers and in the bearing seals, differences in speeds of other parts, and manufacturing inaccuracies.

The coefficient of friction, discussed in Sec. 10–6, of deep-groove ball bearings varies from 0.0014 to 0.0025; self-aligning ball bearings have a somewhat smaller coefficient of friction. The coefficient of friction of cylindrical roller bearings is also low, and that of tapered roller bearings is slightly higher. The coefficient of friction of needle bearings is about three to four times higher, due to the fact that the needles are used without retainers and have a tendency to get out of parallel.

Lubrication. Ball and roller bearings are lubricated with grease or oil. However, the main object of lubrication of this type of bearing is not reduction of friction, but protection from corrosion. When lubricated with grease, they must not be filled to more than about two-thirds of the space available, otherwise the grease may develop internal friction and overheat the bearing, causing its failure. Both ball and roller bearings are sensitive to particles of foreign matter, such as dust, minute metal chips, carbon, etc. Such particles can easily damage the working surfaces, acting as wedges between the balls or rollers and their races. In order to prevent such conditions, some bearings are made with the inner space effectively sealed from the outside. Due to the absence of leakage, they may not require any refilling with grease during their entire life.

10–4. Thrust bearings. Thrust bearings are used for supporting axial loads, such as are created by the pitch of a ship, by bevel gears, helical gears, hydraulic couplings, etc. Thrust bearings may be divided into three main groups:

1. Plain step or collar bearings with sliding contact.
2. Collar bearings with rolling contact.
3. Collar bearings with tilting shoes to obtain perfect lubrication.

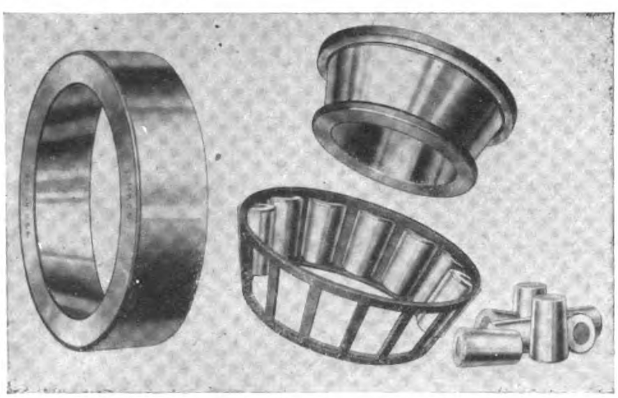


Figure 10-5. Timken taper roller bearing.

Plain bearings. Comparatively small axial loads are taken up by plain collars with their area normal to the shaft axis. These collars are a part of, or are fastened to the shaft and pressed against a plain-bearing surface, usually a part of the journal bearing,

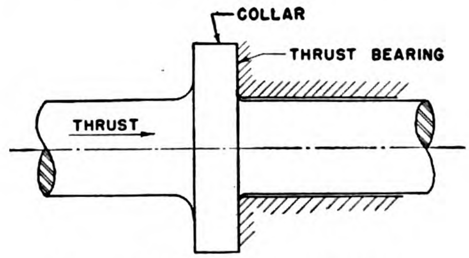


Figure 10-6. Plain thrust bearing.

Fig. 10–6. Lubrication must be furnished to the inside diameter of the collar, as the centrifugal force has a tendency to throw the lubricant out. The thrust bearing surface may be of bronze or babbitt.

Bearings with rolling contact. Heavier axial loads may be supported by a shaft collar by interposing between the collar and the bearing surface steel balls or cylindrical or conical rollers in an appropriate retainer. Ordinary deep-groove radial ball bearings can support a comparatively large axial load if their races are properly supported against axial motion. Special, so-called angular-type ball bearings and tapered roller bearings, can support axial loads equal to and sometimes even slightly higher than their rated radial load.

Tilting-shoe bearings. For heavy axial loads, such as are produced by a ship propeller, the bearing collar is cut into several (six or more) segments, each of which can tilt about a certain fixed point. This action helps to form a lubricating oil film which, when the shaft rotates sufficiently fast, entirely separates the shaft collar surface from the segments. A more complete explanation of this action is given in Sec. 10–6. Such bearings often are called Kingsbury bearings from the name of their inventor and original manufacturer. They can support heavy loads, have a low coefficient of friction, and are suitable both for medium and high rpm. speeds.

10–5. Bearings for reciprocating motion. The two places where a reciprocating motion is encountered in single-acting engines are the wrist pin and the piston. As explained in Sec. 10–6, a reversal of motion is a great handicap to proper lubrication.

Wrist-pin bearings usually are bronze bushings pressed into the piston bosses, sometimes with narrow, slightly helical, oil grooves cut through the full length of the bushing to help the distribution of oil. Similar bushings are used in the small end of the connecting rod of many engines. These bushings may be either of the pressed-in or full-floating type. The maximum bearing pressures run extremely high, up to 6,000 psi., but such high pressures exist only during a small fraction of a second and therefore are not able to squeeze out entirely the protective oil film. The mean bearing pressure is about one-sixth of the maximum. Needle-roller bearings are used in some engines, but the latest trend is back to plain bronze bushings using improved bronze alloys.

Cylinders and pistons. Due to the angularity of the connecting rod, the gas pressure acting on the piston produces a side-thrust pressure between the piston and the cylinder; this pressure, however, is rather small, the maximum value usually being under 100 psi. The lubrication of the piston and cylinder is more than is actually required in all medium-speed and high-speed engines, due to the splash action of the cranks and other rotating parts. As mentioned before, special care must be exerted to remove excess oil from the cylinder walls.

Crosshead guide. In double-acting engines, the side-thrust is taken by the crosshead slipper and its guide. The pressure usually is small, the lubrication does not present any difficulties, and there is no danger of over-lubrication as with trunk pistons of single-acting engines. Fig. 7–15 shows the method of admitting oil under pressure in the oil grooves of the slipper, and also the use of babbitt lining. The crosshead guide is made of cast-iron.

10–6. Principles of lubrication. *Friction.* Regardless of how smooth and true a metal surface may look and feel, it actually is not, but consists of high and low spots. When under a microscope, the cross section appears as shown in Fig. 10–7. When a journal turns in a bearing, there is always a force pressing it against the bearing. The high spots in both surfaces interlock and resist the motion of one surface relative to the other. When this resistance is overcome, the sliding of one surface over the other will be accompanied by a damaging of the softer material. This resistance to sliding is called *friction*. Under the above condition it will be *dry friction*. The ratio of the force which must be applied in the direction of motion (or tangentially to the surfaces), designated F_t in Fig. 10–8, to the normal force, designated F_n, acting at a right angle to the surfaces in contact is called the *coefficient of friction* or briefly,

Coefficient of friction = tangential force ÷ normal force (11–2)

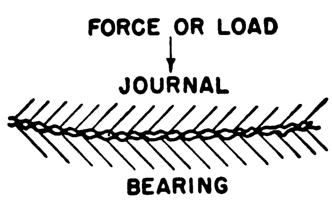


Figure 10–7. Surfaces in contact.

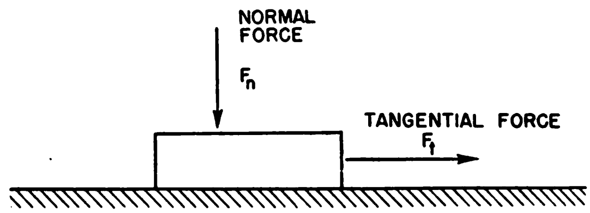


Figure 10–8. Friction caused by a normal force.

Oil film. Due to the tendency of oil to spread on a metal surface and to adhere to it, a thin film can be introduced between the surfaces of a journal and its bearing. The thickness of the oil film may be such that metal-to-metal contact is eliminated entirely. However, some resistance to turning the journal will remain even under these conditions, but it will be of a different nature. It will be the resistance to sliding, of particles of oil adhering to one metal surface in relation to particles of oil adhering to the other metal surface. Such resistance is termed *fluid friction*. The thickness of the oil film which will exist between two metal surfaces in relative motion depends upon several conditions, the main ones being:

1. Bearing pressure.
2. Viscosity of the oil.
3. Relative speed of the moving surfaces.

As stated in Sec. 3–7, viscosity decreases with an increase of the oil temperature.

In the case of a journal and its bearing, the oil film thickness depends also on:

4. *Clearance* or difference between the diameters of the bearing and journal.

5. *Shape* of the *bearing area*, meaning the arc of actual bearing surface, its relative length, ratio of bearing length to its diameter, etc.

An oil film which completely separates two metal surfaces is called a *thick film*. If the oil viscosity decreases or the bearing pressure is increased, some of the oil will be squeezed out and the film thickness will decrease until some of the high spots on both surfaces will come in contact, but the main load still will be supported by the oil film. Such a condition is called imperfect lubrication giving *semi-fluid* or *thin-film friction*.

Lubrication. The object of the lubrication of moving parts is to interpose between the two surfaces an oil film of such a thickness that metal-to-metal contact is eliminated. Such a film is called a *thick film*, and the resulting lubrication is called *thick film* or *perfect lubrication*. If a bearing operates with perfect or thick-film lubrication, there is no wear of the metal surfaces, but the work of overcoming fluid friction is transformed into heat. The temperature of the oil film and of the metal surfaces will increase until the heat dissipated by the bearing and carried away by the oil, if oil circulation is used, will balance the heat created by friction.

With thin-film lubrication the conditions are similar, but a certain gradual wear of the metal surfaces will take place. With dry friction, small particles of metal will be torn from both surfaces, and the heat generated at the points of metal separation may be so high as to weld particles of the bearing metal to the journal surface and to wipe the bearing surface. Ultimately this may result in seizure and destruction of one or both metal surfaces.

Wedge action. Fig. 10–9 shows what occurs in a bearing at various speeds of the journal. The clearances are exaggerated in order to show more clearly the formation of the oil film. The horizontally shaded areas represent the clearance space filled with oil. In (a) the journal is at rest and the oil film has been squeezed out by the load. As the journal begins to rotate, as in (b), due to friction, the journal begins to climb up the wall of the bearing in a direction opposite to that of rotation. However, a layer of oil will adhere to the bearing and another layer to the journal surface. Consequently, oil will begin to be drawn into the wedge-shaped clearance between the rubbing surfaces, at first as a thin film, creating a thin-film lubrication, where the metal surfaces come in contact from time to time. As the journal speed increases, more and more oil will be drawn between the surfaces, the journal acting as a rotary pump which creates a certain pressure in the oil wedge. If the journal speed is sufficiently high, the pressure in the oil between the surfaces will become so high as to be able to support the full load acting on the journal, the latter will begin to float, and the oil pressure will push it toward the other side, as shown in Fig. 10–9c. Fig. 10–10 shows a polar diagram of the pressures in the oil film created by the pump action of the rotating journal. However, in this diagram the pressures are plotted, not from the center, but from the circumference of the bearing. At point *m* the pressure is zero, and gradually is built up to a maximum at point *p*, which is somewhere between point *l* of load action and point *f* of minimum oil-film thickness. From point *p* the oil pressure gradually drops to zero, point *g*, due to leakage. If the bearing has an oil groove in the region of the oil-film pressure, the groove will interfere with the pumping action of the journal, and the oil-film pressures will be lower, as shown in Fig. 10–11. The load-carrying capacity of the bearing will be reduced. Thus, while oil grooves may help to distribute oil over the bearing length, they should not be located in the region supporting the load. In the main bearings of diesel engines, the direction of the load changes in each bearing and between bearings. For this reason, it is safer not to use any oil grooves.

The oil-film pressure must not be confused with the oil pressure in the lubricating system. While the latter usually is from 30 psi. to 60 psi., the oil-film

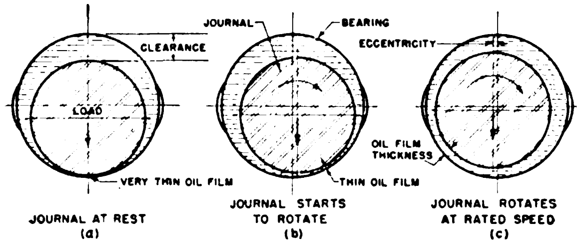


Figure 10–9. Oil film at various journal speeds.

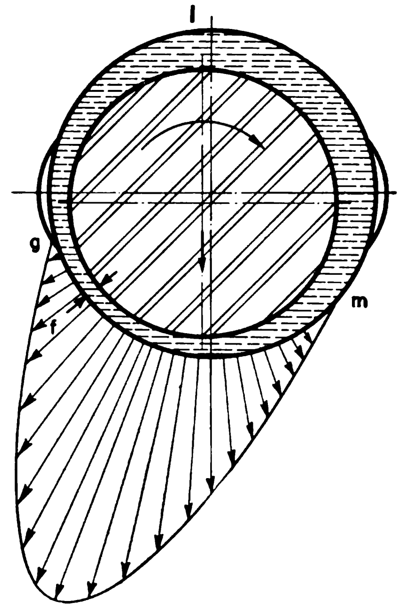


Figure 10–10. Oil film pressure.

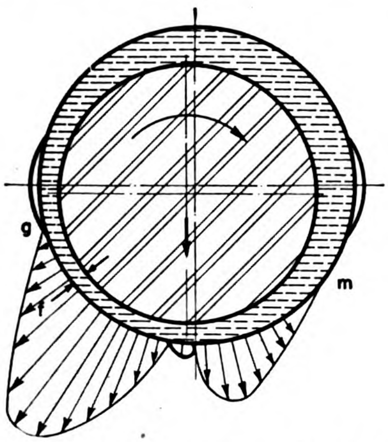


Figure 10–11. Oil film pressure with an oil groove.

pressure, at the point of maximum pressure is considerably higher, reaching several hundred pounds, and under certain conditions may even exceed 1,000 psi.

Clearance. From the above discussion it can be seen that a certain clearance between the journal and bearing which permits the formation of the wedge-type shape is absolutely necessary to obtain the oil-film pressure. However, if the clearance is too great, too much oil will leak out at the bearing ends, and the forces of adhesion of the oil to the metal surfaces will not extend through the whole clearance. In addition, the pumping action will be less effective, the oil-film pressure will be lower, and the load capacity of the bearing will be reduced. The clearances best suited for each bearing, and the limits to which they may increase by wear, are given by the engine manufacturers in their instruction books and should be strictly observed in engine operation.

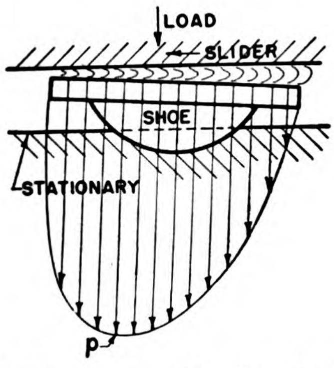


Figure 10–12. Element of Kingsbury thrust bearing.

Kingsbury thrust bearing. The lubrication of this bearing is based on the same principle: the adhesion of oil to metal surfaces and the tendency to build up pressure if oil is drawn into a wedge-shaped space. Fig. 10–12 shows an element of this bearing; the stationary part has shoes which can tilt, being supported on a ball-and-socket joint. The center of this joint is slightly nearer to the discharge edge. Oil drawn in by adhesion to the slider, which moves in the direction indicated by the arrow, due to the difference in the supporting areas, tilts the shoe and produces a wedge-shaped space into which the oil is drawn and raises the pressure as indicated by the pressure curve. The oil film pressure drops, due to leakage after reaching a maximum at point *p*.

Fig. 10–13 shows a cross section of an assembled bearing and its component parts.

10–7. Bearing-lubrication methods. Bearings may be lubricated:

1. Intermittently.
2. Continuously, with an abundant supply of lubricant.
3. Continuously, with a limited supply of lubricant.

Intermittent lubrication may be obtained with oil or grease applied by: (1) dropping oil from an oil can into an oil hole or into a plug or waste; (2) forcing grease from a grease cup or a pressure gun into a hollow space.

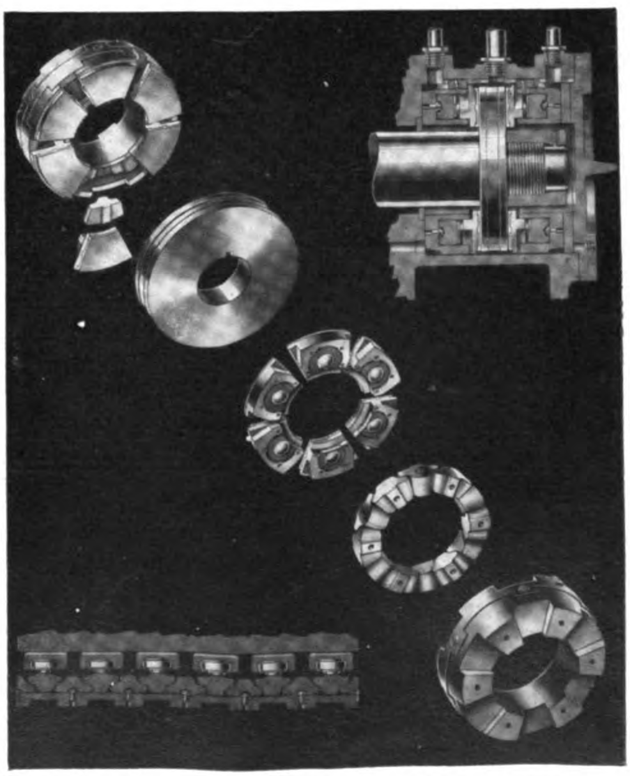


Figure 10–13. Kingsbury thrust bearing.

These methods provide only imperfect lubrication and are used for places where the loads are small and velocities very low.

Limited continuous lubrication may be obtained by: (1) grease cups with spring action; (2) oil reservoirs with a wick which carries the oil by capillarity; and (3) sight-feed drop oil cups.

All three methods are suited to light duty only.

Abundant lubrication alone insures perfect lubrication and may be obtained by:

1. Ring, chain, and collar oiling.
2. Splash and bath lubrication.
3. Force-feed lubrication.
4. Pressure or flooded lubrication.

Ring, chain, and collar lubrication gives satisfactory results only at low and medium speeds. At higher speeds the oil is thrown off by the centrifugal force.

Splash and bath lubrication is satisfactory for light and medium duty, but makes no provision for maintaining a desired oil temperature.

Force-feed lubrication delivers oil to desired points under a very small pressure, but in definite amounts which can be regulated by means of sight-feeds. In Navy diesel engines, this method is applied only for lubrication of the cylinders and piston rods of double-acting engines.

Pressure lubrication, sometimes called *flooded* lubrication is, in diesel engines, the standard method of lubrication of all important parts which do not get oil by splashing from the crankcase. The oil is drawn from the crankcase bottom, called the oil *sump*, by a gear-type pump, pumped through a filter and oil cooler to the header, and then through separate lines to the main bearings. The header is sometimes called the *engine oil gallery*. From the main bearings most of the oil passes through drilled passages in the shaft and crank webs to the crankpin bearings. From the crankpin bearings the oil usually passes through the rifle-drilled connecting rod to the wrist-pin bearings. In some engines, oil from the wrist-pin bearings is used for cooling the piston. Separate lines from the header carry oil to the camshaft bearings, gears, and other parts requiring lubrication.

The oil circulated through the main and crankpin bearings serves not only to lubricate them, to insure formation of the wedge-shaped oil film, but also to cool these bearings, by carrying away the heat into which the work of friction is transformed. The oil pressure in the header is maintained at from 30 psi. to 50 psi., depending upon the engine type. The correct pressure for a particular engine is indicated in the instruction book. The oil-circulating pump has a larger delivery than is actually required, and the excess oil is by-passed from the high-pressure side back into the sump through a spring-loaded adjustable pressure regulator.

10–8. Problems. Find the pressure on a bearing with a 7-in. dia., 5-in. length, when the load is 19,800 lbs. *Ans.* 566 psi.

2. Determine the coefficient of friction between the crosshead and its guide surface in a horizontal engine if it takes an effort of 7 lbs. to slide the crosshead which weighs 152 lbs. *Ans.* 0.046.

10–9. Questions. 1. Into what groups can all bearings be divided?

2. What loads must be carried by diesel engine bearings?

3. Enumerate the advantages of ball and roller bearings.

4. What is the main object of lubricating ball bearings?

5. What types of bearings are used to take axial loads?

6. What are the main places in a diesel engine where bearings for reciprocating motion are used?

7. What is the object of lubricating moving parts?

8. What conditions influence the thickness of the oil film in plain bearings?

9. What is meant by thick-film or perfect lubrication?

10. What is the standard method of diesel engine lubrication?

CHAPTER 11

ENGINE SYSTEMS

11–1. Fuel system. The fuel-oil system in a diesel-powered naval vessel must be installed, operated, and maintained with the same care and supervision as the ship's engines. Inspections, maintenance, and operation of fuel-oil tanks and fuel-handling equipment must be carried out in accordance with United States Navy Regulations and the manuals of the various Bureaus concerned.

The fuel is pumped from the *storage tank* to the supply or *day tank* and from there it is delivered to the fuel injection pumps on the engine. It is good practice to clean the fuel of sediment and water before it enters the supply tank. This is usually done with a *centrifugal purifier*. The fuel is transferred from the supply tank by means of an engine-driven pump, variously called *booster*, transfer, or *primary pump*, through a metal-edge *strainer* and a cartridge-type, replaceable-element *filter* to a header on the suction side of the fuel injection pumps. Excess fuel pumped to the fuel header and from the fuel injection pumps is returned to the supply tank. Any leakage fuel, often called *drippings*, from the injection pumps and nozzles, which may be mixed with used lubricating oil, is normally collected in separate containers or *drip pots* and returned to the main fuel storage tank. Fig. 11–1 shows a typical fuel-supply system.

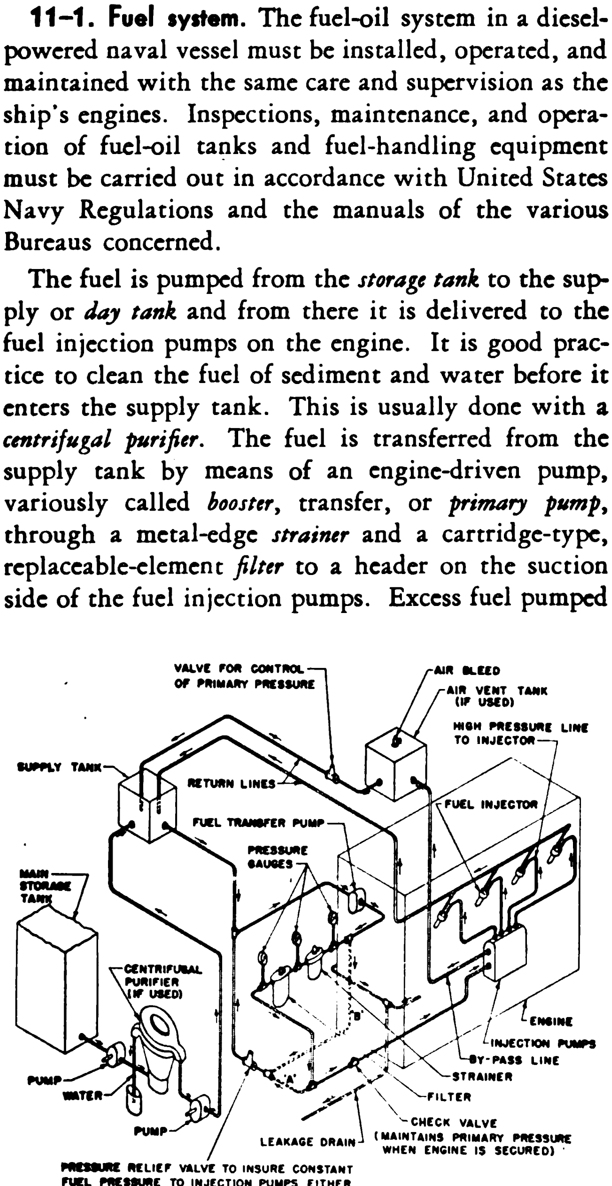


Figure 11–1. Fuel-oil system.

The *centrifugal purifier*, Fig. 11–1, is a machine similar to the separator used for separating cream from milk. The oil enters a revolving bowl which tends to throw any heavy, solid contaminants to the outside of the bowl, followed by an intermediate layer of water which is heavier than the oil but lighter than the solids, and finally a central core of oil. The discharge holes in the top of the bowl are so located that the water can be drawn off separately from the fuel oil. The solid material collects around the periphery of the bowl and must be cleaned out periodically. This cleaning can be done once a day unless idleness of engine or exceptionally clean fuel indicates that more extended periods are permissible.

The *supply* or *day tank* is usually vented to the atmosphere and mounted at a high point in the fuel system to allow all air to escape. The greatest care should be taken to keep all fuel lines under a pressure greater than atmospheric. Also, excessive foaming or splashing of fuel oil in the tanks should be avoided in order to prevent air from becoming entrained with the fuel, as this air interferes with proper operation of the injection pump.

Small-boat installations are often equipped with a *float chamber* instead of a tank, if it is impractical to mount an overhead tank. These chambers are equipped with air-vent valves which require careful cleaning and adjustment to insure trouble-free service.

The engine-driven fuel *transfer pump* is of the positive displacement type, usually with a built-in relief valve to insure constant pressure to the injection system. The fuel strainer and filter are connected into

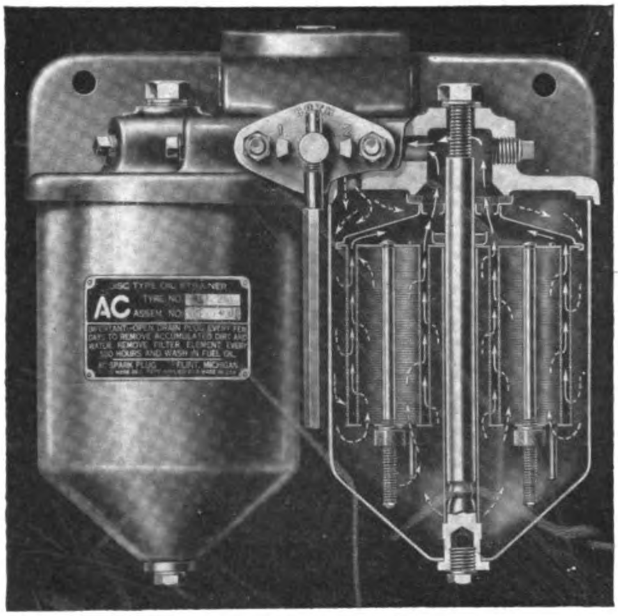


Figure 11–2. Primary fuel strainer.

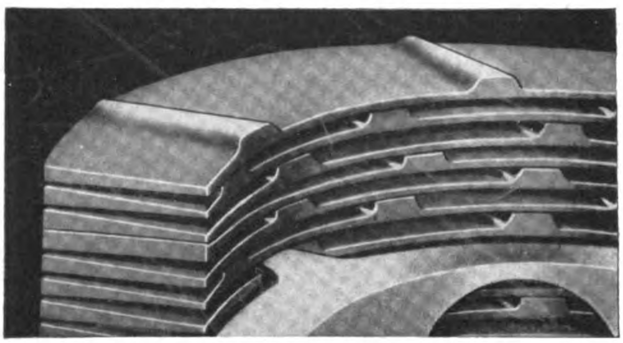


Figure 11–3. Enlarged section of ribbon.

the system on the discharge side of this pump and the pressure drop across these filters increases with time. For this reason, some systems have a relief valve in the line before the cartridge filter, the by-passed oil being returned directly to the supply tank. This valve insures a more constant fuel supply pressure, and also tends to vent the suction side of the system of entrained air.

The *primary metal-edge fuel strainer* used in Navy installations is a duplex type which is equivalent to two complete strainers connected by suitable headers or piping. This arrangement allows either strainer to be completely cut out of the system for cleaning or repair while all of the oil is flowing through the other strainer. Fig. 11–2 shows a typical metal-edge strainer. A magnified view of a portion of the element is shown in Fig. 11–3. The oil flow is from the outside to the inside. In Navy approved strainers, the spaces between the *leaves* or *ribbons* which act as oil passages are between 0.001 and 0.0025 in. The pressure drop across these strainers must not exceed 1.5 psi. with an oil flow equal to the full capacity of the fuel-oil pump. In some engines the duplex strainer is placed between the fuel supply tank and the transfer pump, and in operation may be working under a vacuum.

The *secondary*, cartridge-type, fuel-oil *filter* contains elements which are built to standard Navy dimensions. All new engine construction must incorporate such filter elements. The maximum outside diameter is 3 in., and the over-all length is either 4 in. or 8 in. The elements in these filters should be discarded and replaced with new ones when the pressure drop across the filter reaches 15 psi. The sump of the filters and strainers should be drained as often as practicable, preferably when the fuel is flowing.

11–2. Lubricating systems. One of the main problems in the diesel engine lubricating-oil circulation is the *filtering* of the oil. The following lubricating-oil filter systems are permitted in the Navy: (1) *shunt*, (2) *by-pass*, (3) *sump system*. The selection of the system depends upon the application.

The *shunt system* commonly used for large installations is shown diagrammatically in Fig. 11–4. The oil, after lubricating various engine parts, flows by gravity to the engine sump. From there it is picked up by the oil pump and sent first through a strainer, then a filter, and finally a cooler. From the cooler, it passes back to a header, called the *engine oil gallery*, and then is distributed by different pipes to all places needing lubrication, as explained in Sec. 10–7. The oil pump delivers a constant amount of oil per revolution, but the resistance in the strainer and filter varies as they pick up the contaminants from the oil. For

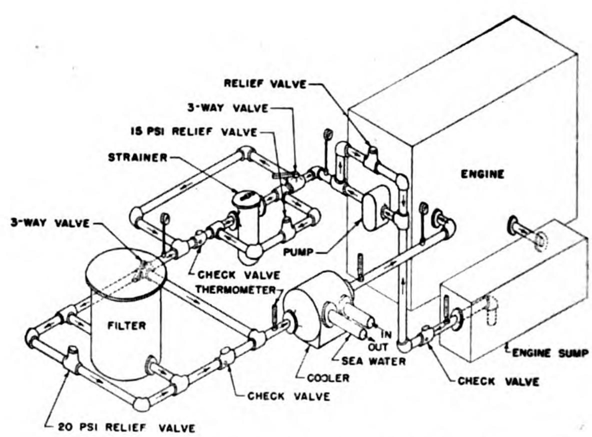


Figure 11–4. Shunt system of lubrication.

this reason, the desired pressure is maintained automatically by by-passing, if necessary, part of the oil around the strainer and filter by means of the pressure relief valves built into the strainer and filter. The three-way valves are used to by-pass these elements when cleaning the strainer or exchanging the filter element.

The *metal-edge strainer* is similar in construction to the strainer used for fuel oil except that it has a larger slot opening, 0.005 in., and must pass all of the lubricating oil with a pressure drop of 5 psi. or less, when the oil viscosity is 170 S. S. U.

The *filter* contains elements built to Navy Standard dimensions of 18 in. over-all length and 7-5/16 in. over-all diameter. Figure 11–5 shows a typical filter for this service. The relief valve is built into the filter housing and the elements may be stacked two-high. In this type of installation, the flow of oil through the filter is limited only by the pressure drop across the filter which is controlled by the setting of the relief valve. The rate of removal of abrasive matter with this installation is higher than with any other. The quantity of oil passing through the filter elements does not subtract from the oil flowing to the engine oil gallery as it does in the by-pass system.

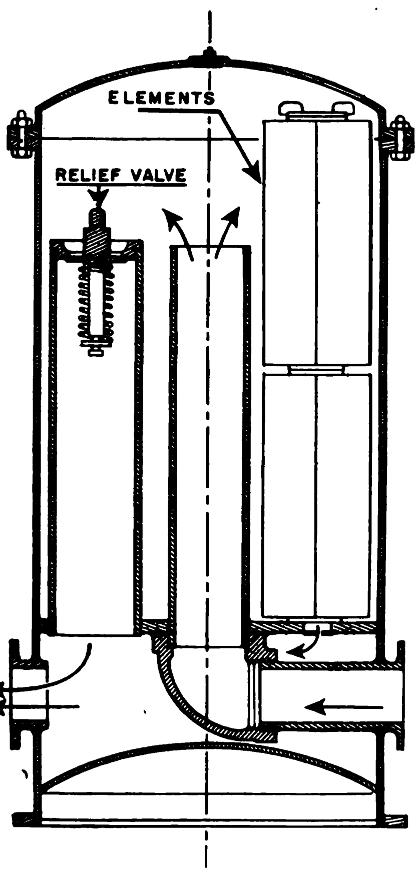


Figure 11–5. Metal-edge strainer.

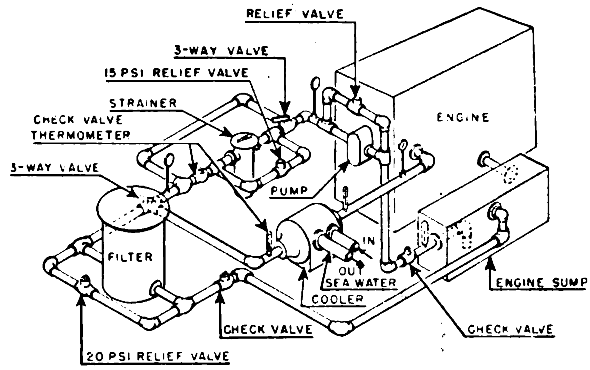


Figure 11–6. By-pass system of filtering lubricating oil.

The by-pass filter system is normally used in small powerboat or land-vehicle installations. It is shown diagrammatically in Fig. 11–6. The arrangement is the same as with the shunt system except that the filter discharges the oil back to the engine sump or crankcase. In order that sufficient oil will be supplied to the engine, the amount by-passed through the filter must be limited. This is done on small filters by putting a 1/16 in. flow-restricting orifice in the standpipe through which the oil is discharged from the filter element. This orifice is shown in Fig. 11–7 of a typical Navy Standard small lubricating-oil filter. The elements have an over-all length of 8 in. and an over-all diameter of 4½ in.

The sump system of filtration, Fig. 11–8, is similar to the shunt system except that the filter is placed in a separate recirculating system with a separate pump. This allows maximum freedom in the operation of the filter. Since the auxiliary filter pump is motor-driven, operation of the filter is possible even when the engine is secured. The oil should always be filtered while hot, 140° F to 180° F, as the solid material is more easily removed from oil having a lower viscosity.

Additive oils. In operating diesel engine lubricating systems containing heavy-duty, additive-type oil, the oil must be replenished at reasonable intervals, because the additives which keep sludge deposits from forming, gradually are being used up. Optimum drain periods can best be governed by frequent oil analysis. The following used-oil property values have been recommended by the Bureau of Ships to indicate when the lubricating oils should be changed:

Neutralization number	0.5
Precipitation number	0.5
Fuel dilution	5.0%

To check the condition of the oil, samples of it

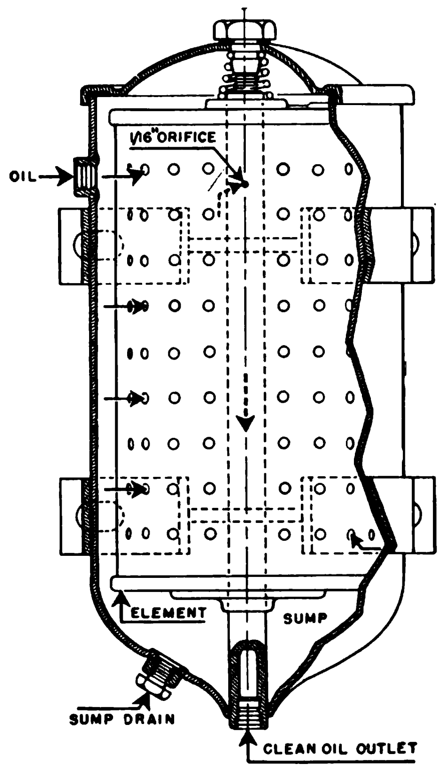


Figure 11–7. Small lubricating oil filter.

must be sent to the nearest Naval chemical laboratory. If equipment to determine the precipitation number and fuel dilution are available on board ship, the determination should be made following the instructions furnished with the equipment.

If facilities are not available for making the necessary analysis, the oil should be changed at 500-hr. intervals, except with small, high-speed engines where 100-hour oil change periods are required. Filter elements should be changed every time the oil is changed unless the oil-drain period is unusually short due to dilution or some other abnormal condition.

Centrifuges, if used for purification of lubricating oil, are normally connected into the system in the same manner as the filter in Fig. 11–8. They should be operated continuously when the engine is operating and as long thereafter as practicable, or until the oil viscosity has increased to about 200 S. S. U. due to natural cooling.

Additional features. The lubricating system also has a line from a *priming pump* whereby oil can be circulated through the engine before starting.

There should be a hand-controlled by-pass around the cooler to regulate the oil temperature at part loads, in order to maintain it as close to full-load conditions as possible.

11–3. Cooling systems. *Heat transfer in diesel engines*. Heat is developed in diesel engines by the compression of air in the cylinders, by combustion of fuel in the combustion chambers, and as a result of friction between the rubbing surfaces. So long as the temperature of the engine is greater than that of the surroundings, there will be a tendency for the heat to flow from the engine. Thus, the heat developed in the engine, which is not converted into useful work, will be dissipated by heating its surroundings.

As stated in Sec. 2–7, heat may be transferred by three methods: conduction, convection, and radiation. Heat transfer *within* the engine occurs by all three methods. Heat transfer *from* the engine, however, occurs chiefly by convection of heat in the exhaust gases and in the cooling water flowing from the engine, and by radiation from the hot external engine surfaces to the cooler surfaces of the engine room.

In a diesel engine, approximately one-third of the heat energy in the fuel is regularly converted into useful work, approximately 30 per cent is lost as heat in the exhaust gases, and another 30 per cent is lost as heat to the cooling water. The remainder is lost in overcoming friction which, being transferred into heat, goes partly to the cooling water and partly to the surroundings, being dissipated by radiation. These figures vary with different engine models, and with the load. However, they will be used in the problems to follow.

The heat lost to the exhaust and by radiation depends chiefly upon the engine load and cannot be controlled directly. The operating temperature of the engine must be regulated by controlling the cooling system.

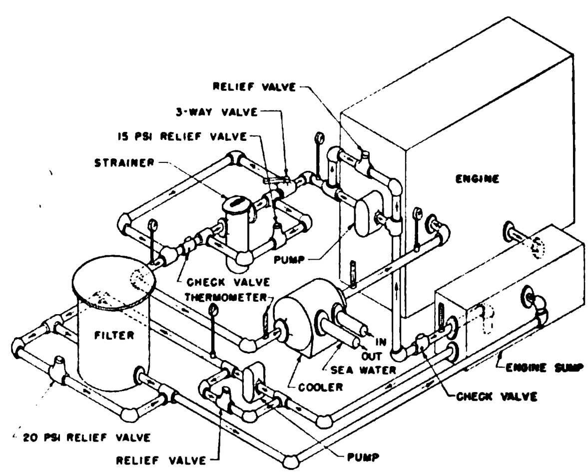


Figure 11–8. Sump system of oil filtering.

Purpose of cooling. Cooling must be provided for internal combustion engines, mainly for the following reasons:

1. To prevent breakdown due to overheating of the lubricating oil film separating the engine rubbing surfaces.
2. To prevent loss of strength due to overheating of the metal in the engines.
3. To prevent excessive stresses in or between the engine parts due to unequal temperatures within the engine.

The cooling medium or coolant used in internal combustion engines is either water or air. However, marine diesel engines for obvious reasons use water. The water used as coolant should not contain any impurities which might form deposits on the inside of the engine water passages and thus impair the heat transfer. For this reason, *fresh water*, or water which has been treated to remove scale-forming impurities, is used almost universally as the coolant for Navy diesel engines.

Cooling requirements. In order to keep the temperature of a diesel engine within practical limits, approximately 30 per cent, as previously mentioned, of the heat from the fuel burned in the cylinders must be transferred to the cooling system.

> *Example 11–1.* It is required to find how much heat must be dissipated by a 250-hp. engine that uses 114.6 lbs. of fuel per hour, the heat value of the fuel being 18,500 Btu./lb.
>
> The heat developed is
> $114.6 \times 18{,}500 = 2{,}120{,}100$ Btu./hr.
>
> The heat to be dissipated is 30% of it, or
> $2{,}120{,}100 \times 0.30 = 636{,}030$ Btu./hr.
>
> The heat dissipated per bhp. must be
> $636{,}030 \div 250 = 2{,}544$ Btu./hp.-hr.
>
> or $2{,}544 \div 60 = 42.4$ Btu./hp.-min.

This heat must be transferred to the cooling water with a small temperature difference, usually from 10° F to 20° F, between inlet and outlet water in order to maintain the temperature of the engine parts within the required limits. To provide the heat transfer necessary, the cooling water must be circulated at a rapid rate. The amount of water which must be circulated to absorb the heat with a given temperature rise may be readily determined.

> *Example 11–2.* Find the gallons of water it is necessary to pump per minute to cool a diesel engine putting out 500 bhp., burning fuel having a heat value of 18,500 with a specific fuel consumption of 0.458 lb./bhp.-hr., if the temperature difference between the inlet and outlet of the cooling water is 15° F.
>
> Heat developed per hour $= 18{,}500 \times 0.458 \times 500 = 4{,}236{,}500$ Btu./hr.
>
> Heat lost to cooling water, assuming as stated before 30 per cent.
>
> $4{,}236{,}500 \times 30 \div 100 = 1{,}270{,}950$ Btu./hr.
>
> or per minute $= 1{,}270{,}950 \div 60 = 21{,}183$ Btu./min.
>
> But: 1 lb. of water absorbs 1 Btu. during each 1° F rise; therefore: water required to absorb 21,183 Btu. with a 15° F rise $= 21{,}183 \div 15 = 1{,}412$ lb./min. Since one gallon of water weighs 8.33 lbs.: gallons water required per minute $= 1{,}412 \div 8.33 = 169.5$ gpm.

In order to operate the engine with the desired small temperature differences, all Navy diesel engines use forced circulation of the coolant by means o pumps of various types, described in Sec. 12–3. The power required to operate these pumps amounts to 1 to 2 per cent of the engine output, depending upon the resistance to flow of the water passages and the rate of circulation required.

Cooling. Where a large source of cool water is available, as in lakes or rivers, it may be used directly as a coolant. Engines in this case, use the so-called *open cooling system* in which the cooling water is pumped directly through the engine and discharged overboard. This system has also been used with sea water, but is not satisfactory for the best engine performance. When sea water is used directly in the cooling system, salt deposits will be formed in the engine water passages if the water outlet temperature exceeds approximately 120° F, which is too cool for efficient operation. To permit more efficient operation, marine engines use the so-called *closed cooling system.*

In the closed system, a supply of fresh water is circulated continuously by means of a pump through the engine jackets and then through a jacket-water cooler back to the engine in a closed circuit, Fig. 11–9. The fresh water is cooled as it passes through a cooler by means of the sea water. The sea water is pumped by a second pump from the *sea chest*, flows over the oil tubes of the lubricating-oil cooler, then

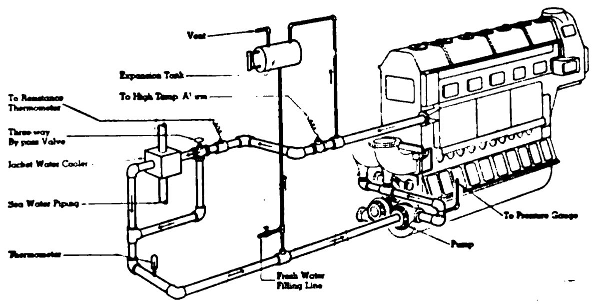


Figure 11–9. Closed cooling system.

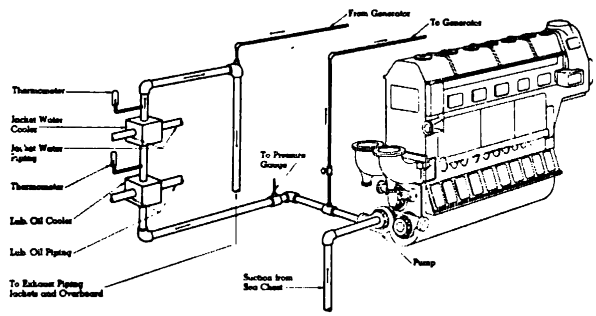


Figure 11–10. Sea-water system.

over the outside of the fresh-water pipes or tubes in the cooler, and then is discharged overboard, Fig. 11–10.

With this system, the temperature of the cooling water leaving the engine may be maintained from 160 to 180° F which is the most favorable temperatures for efficient performance of a diesel engine. There is no danger of scale forming in the water passages in the engine, because only fresh or softened water is used in the engine itself. Salt deposits will not be formed in the sea-water passages because the temperature rise through the heat exchanger may be limited to from 10 to 20° so that the maximum sea-water temperature can be kept below 100° F. The disadvantages of this system are that additional equipment is required; separate pumps for the fresh and sea water, a heat exchanger, a fresh-water supply, and additional piping. Also, greater work is necessary to drive two pumps. The gain in performance, however, more than offsets these disadvantages and practically all Navy diesel engines use the closed cooling system.

In the latest Navy engines, the fresh water coming from the cooler first passes through the lubricating oil coolers, and after that into the pump and through the engine jackets, as shown in Fig. 11–11. This arrangement prevents corrosion of the lubricating oil cooling elements and eliminates the danger of salt water getting into the lubricating system. Fig. 11–11 also shows a cross-connecting pipe between the fresh-water pipe from the engine and the suction line to the pump. This cross pipe has a fixed orifice inserted in it which admits a certain, predetermined amount of hot jacket-water into the suction line and thus maintains a desired temperature of the water admitted to the engine. Fig. 11–11 shows also a temperature-regulating valve before the water cooler which permits part of the water to by-pass the cooler, again to prevent excessive cooling of the jacket water.

In Fig. 11–12 is seen the sea-water pipe for the arrangement illustrated in Fig. 11–11. Both Figs. 11–10 and 11–12 show smaller pipes leading from the sea-water pressure pipe to the generator and back into the discharge line. The water is circulated through the air cooler and carries away the heat produced by the operation of the electric generator.

Sea-water pumps may be either of the centrifugal or positive displacement type, since the flow through the water cooler is maintained constant for a given engine speed and no regulating valves are necessary in the sea-water circuit. Most fresh-water pumps, however, are of the centrifugal type which permits regulation of the flow without building up excessive pressures.

As shown in Figs. 11–9 and 11–11, the closed fresh-water pipe system has an *expansion tank* connected to the inlet and outlet pipes to provide for the change of volume of the water in the closed system. When the temperature of the circulating water rises, its volume increases, and the excess volume is discharged into the expansion tank. When the temperature decreases, the shrinkage of volume of the water is made up from the expansion tank.

11–4. Air-intake system. The combination of devices through which air is supplied to diesel engines is called the *air-intake system.*

The purpose of the intake system is to provide the air required for the combustion of fuel. However, an

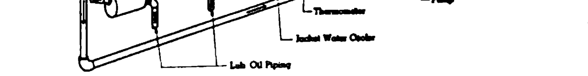


Figure 11–11. Closed cooling system.

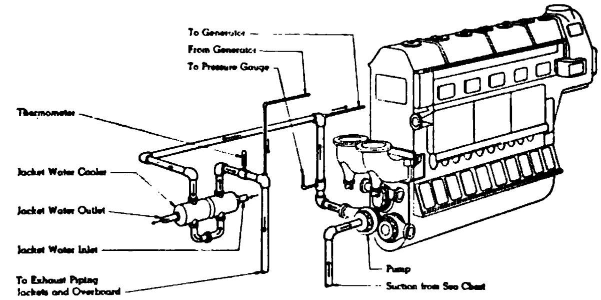


Figure 11–12. Sea-water system.

intake system may have to perform one or several of the following additional functions:

1. Scavenging the cylinders of the burned gases, in two-stroke engines.
2. Supercharging, to increase the power output of an engine.
3. Cleaning the intake air.
4. Silencing of the intake noise.

The intake system is made up of the following parts:

1. Air cleaner.
2. Air intake silencer.
3. Air intake boxes or header.
4. Intake valves or ports.
5. Blower for supplying scavenge air in a two-stroke engine or for supercharging a four-stroke engine.
6. Air coolers for a supercharged four-stroke engine.

However, it should be noted that some of these parts may be absent in a particular engine.

Air cleaners. An air cleaner or filter is a device used for removing dust particles from air to prevent injury of internal engine surfaces by abrasion. Screens used to prevent rags or other objects from entering the intake system are not considered air cleaners. The resistance to air flow through an air cleaner must be as small as possible in order to obtain a high volumetric efficiency for a four-stroke engine, as discussed in Sec. 6–2. This means that the air velocity through the cleaner must be low, and hence, large flow areas are required. Large flow areas in turn necessitate bulky filters for which the space is generally not available in crowded ship installations. Therefore, air cleaners are seldom used on large engines. Small-boat engines are their principal field of application. Of the many air filter types, the one most generally employed in naval service consists of a pad of metallic gauze which is periodically wetted with oil, Fig.

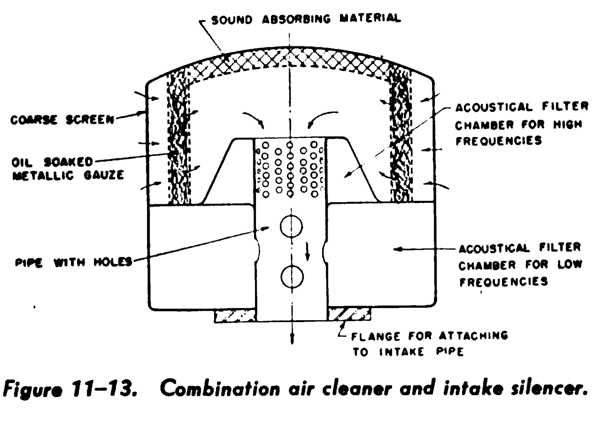


Figure 11–13. Combination air cleaner and intake silencer.

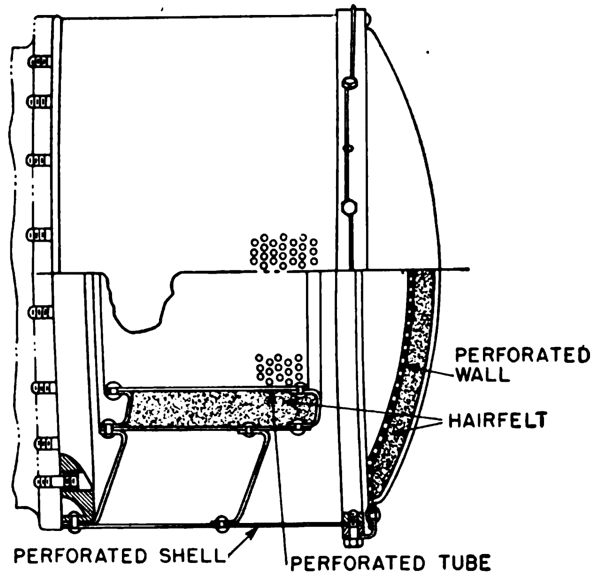


Figure 11–14. Air intake silencer.

11–13. The air drawn into the engine passes through this pad and particles of dust adhere to the oily gauze. The filter element must be removed from time to time and cleaned to prevent clogging and resulting obstruction to air flow.

Intake silencers. An intake silencer is a device used to reduce the noise of engine or blower intake. Intake silencers are of two general types. One type uses a sound-absorbing material, such as fireproofed hair felt placed around air passages having perforated walls, Fig. 11–14. Sound passes through the perforation and is partly absorbed by the felt. Another type, shown in Fig. 11–13, has one or more chambers connected to the air passage by openings. The chambers block the passage of certain sound waves through the silencer without obstructing air flow. Intake silencers of this kind are sometimes referred to as *acoustic filters*. Air cleaners and intake silencers are often built as one unit, as shown in Fig. 11–13.

Intake headers and piping. In order not to reduce the volumetric efficiency of four-stroke engines, the intake headers and piping should offer the least possible resistance to air flow. Sharp turns and small flow areas must be avoided. In two-stroke engines, restrictions in the air piping decrease the amount of scavenge air and maximum power output. Fig. 11–15 shows the intake receiver surrounding the cylinder of a two-stroke engine with a Roots blower.

Intake valves and ports. The importance of proper timing is discussed in Sec. 6–5.

Blowers and scavenging. Most modern two-stroke diesel engines use Roots-type rotary positive-displacement blowers for supplying scavenge air. The blowers are described in Sec. 12–1. One method of placing the blower is shown in Fig. 11–15. Very few engines use centrifugal blowers driven by gears from the crankshaft. The pressure developed in a centrifugal blower is proportional to the square of the rpm. of the blower. Due to this characteristic, the pressure drops off rapidly when the speed is lowered. When a centrifugal blower is driven by the engine itself through gears, there is a minimum engine speed below which the blower fails to develop sufficient pressure to deliver air into the cylinder. This, obviously, causes the engine to stop and explains why centrifugal blowers are not used generally for scavenging two-stroke engines.

Positive-displacement blowers have an important advantage over centrifugal blowers, in that the amount of air delivered per revolution changes comparatively little with speed and discharge pressure. This enables the engine to maintain a high mean effective pressure and torque at low speeds.

Supercharging. In two-stroke engines the same blowers are used for supercharging as for scavenging. However, for supercharging they must deliver more air which means high scavenge pressures.

Four-stroke engines are supercharged either by positive-displacement rotary blowers or by centrifugal blowers. The positive displacement blowers are of the same type as used for scavenging two-stroke engines and are driven in a similar way from the crankshaft, either by gears, chains, or belts. The centrifugal blowers are driven by turbines operated by exhaust gases coming from the engine which the blower serves.

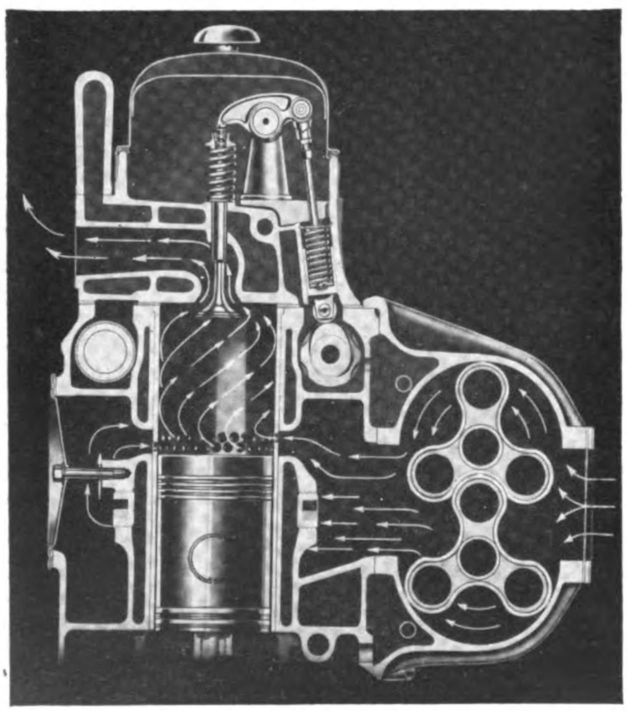


Figure 11–15. Air intake system through blower.

11–5. Exhaust system. The combination of devices through which exhaust gases leave a diesel engine is called the exhaust system.

The primary purpose of the exhaust system is to conduct exhaust gases from the engine cylinders to the atmosphere with a minimum of flow resistance. Exhaust systems usually also perform one or several of the following functions:

1. Muffle exhaust noise.
2. Quench sparks.
3. Remove sparks and other solid material from exhaust gases.
4. Furnish energy to turbine-driven superchargers.
5. Furnish heat to evaporators for making fresh water, or for heating purposes.

Ordinary exhaust systems usually consist of the following parts arranged in the order mentioned:

1. Exhaust valves or ports;
2. Exhaust manifold;
3. Exhaust pipe;
4. Muffler;
5. Tailpipe.

Special exhaust systems in naval installations may have one or several of the following additional parts:

1. Spark arrester;
2. Turbo-supercharger;
3. Exhaust heat evaporator or heater.

Exhaust ports and valves. The various arrangements of exhaust ports and valves used on two- and four-stroke cycle engines are discussed in Secs. 5–5 and 8–5.

Exhaust manifold. Exhaust manifolds are used on multi-cylinder engines to collect the exhaust gases from the several cylinders. A manifold is a gas receiver to which each cylinder is connected (see Fig. 11–17). Exhaust manifolds are generally cooled by water flowing through a water-jacket surrounding the manifold.

Exhaust pipe. The exhaust pipe connects the exhaust manifold to the muffler. In order to reduce transmission of the engine vibrations, avoid stresses set up by expansion of the hot pipe, and simplify installation of the muffler, the exhaust pipe, or at least part of it is made of flexible piping, Fig. 11–17.

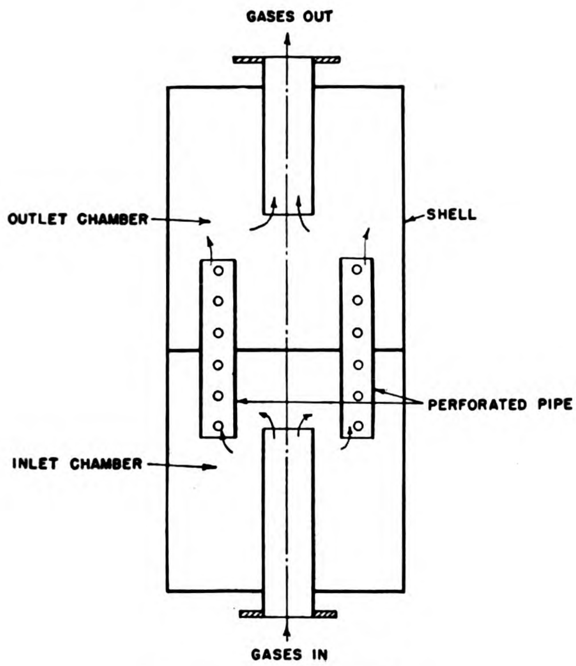


Figure 11–16. Dry muffler.

Tail-pipe. The tail-pipe is connected to the muffler outlet and discharges the exhaust gases to the atmosphere.

Mufflers. Mufflers are devices used for quieting exhaust noise. There are two types of mufflers used with Navy engines, the dry and the wet types. In the wet-type mufflers, water is mixed with the exhaust gases. Wetting the exhaust gases serves to quench sparks, contributes to silencing, and cools the exhaust, thereby eliminating the need for insulation. The disadvantages of wet-type mufflers are that they usually offer a higher back pressure and shorter service life, due to corrosion, than dry-type mufflers. By proper design and materials these disadvantages can be reduced.

There are many different mufflers in use, both wet and dry, and no design is accepted yet as standard in the Navy. Fig. 11–16 shows a dry-type muffler. Fig. 11–17 shows a wet-type muffler in which additional water is admitted to the tail-pipe and all the water is drained through it. In some wet mufflers, separate water drains are provided. However, dry mufflers must also have plugs in suitable places to drain water, in case it is condensed out of the exhaust gases which may contain up to 10 per cent of steam formed from combustion of hydrogen in the fuel oil.

Silencing. Sound is a vibration produced by the release of high-pressure gases from the engine cylinders at each exhaust event. The higher the frequency of vibration, the higher is the sound, and the more noticeable to the ear. When this high and frequent vibration is suppressed, noise is reduced. The high velocity of the escaping gases has little to do with the noise. The silencing effect in modern mufflers is obtained by trapping the high-frequency vibrations without interfering with the free flow of exhaust gases. This can be produced by properly proportioning muffler chambers and internal passages.

Some wet-type exhaust systems used on small boats consist only of an exhaust pipe into which water is admitted.

Spark arresters. Spark arresters are devices used in dry-type exhaust systems for separating and trapping sparks and other solid material in the exhaust gases. The separation is obtained through an abrupt change of the direction of flow of the gases. Most spark arresters include also some device for inducing a whirling motion of the exhaust gases, such as the spiral vanes shown in Fig. 11–18. The centrifugal force developed by the whirling action throws solid particles toward the inside wall of the cylindrical casing. From there they slide down and fall into a

Parts shown on Figure 11-17.

1. Exhaust pipe.
2. Inlet.
3. Bolting flanges.
4. Exhaust inlet pipe.
5. Sea water inlet.
6. Pipe through baffle.
7. Shell.
8. Outlet compartment.
9. Outlet pipe.
10. Outlet—exhaust gases and sea water.
11. Clean out opening.
12. Inlet chamber.
13. Baffle.
14. Drain plug.
15. Bolting flanges.
16. Sea water inlet.
17. Hull plate.
18. Tail pipe.

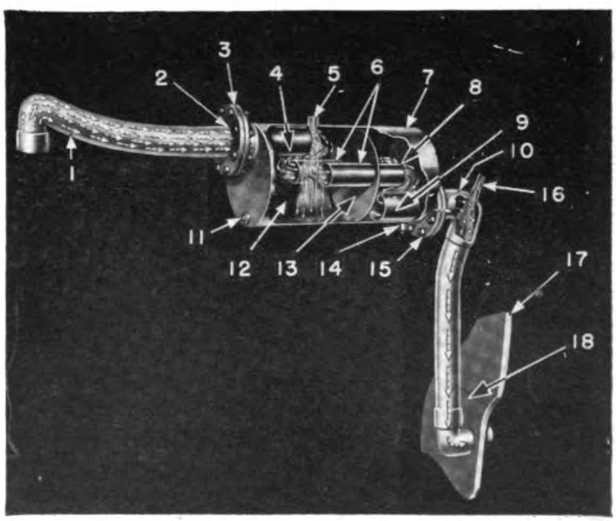


Figure 11–17. Exhaust manifold and wet muffler.

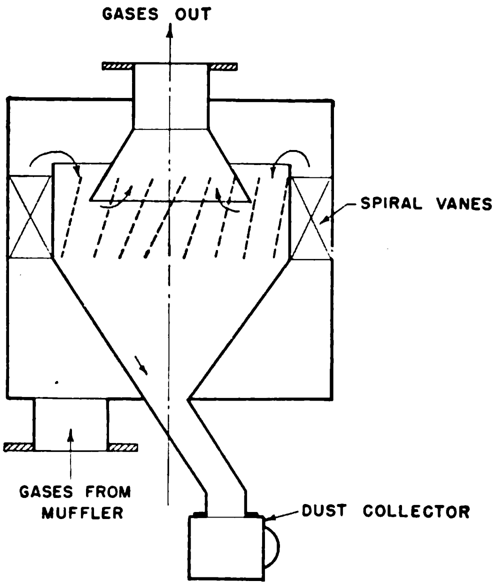


Figure 11–18. Spark arrester.

conical collecting chamber from which they are removed periodically. The exhaust gases are generally admitted off center, but are carried out through a central passage. Spark arresters are often included as a part of the dry muffler.

Exhaust-heated evaporators. Exhaust-heated evaporators are devices which use the heat of the exhaust gases for evaporating sea water. The resulting steam or vapor is then condensed, and since salt is not evaporated, the condensate is fresh water. Thus, an evaporator is the same as a still. A typical evaporator is shown diagrammatically in Fig. 11–19. Such evaporators are used principally on some older submarines and surface craft. Due to operating difficulties, exhaust-heated evaporators have been largely supplanted by electrically heated units.

11–6. Problems. 1. Assuming that 30 per cent of the heat developed by combustion of the fuel in a 425-bhp. engine is transmitted to the cooling water,

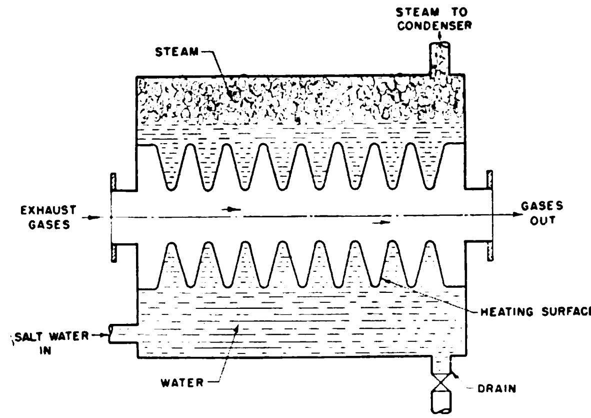


Figure 11–19. Exhaust-gas heated evaporator.

find the heat transmitted per min. if the engine uses 0.395 lb. hp.-hr. of fuel having a heat value of 18,000 Btu./lb. *Ans.* 15,600 Btu./min.

2. Determine the amount of fresh water which must be circulated per min. to carry away the heat from the engine of problem 1 if the water temperatures are 158° F at the inlet and 170° F at the outlet. *Ans.* 156 gal./min.

3. Determine the temperature of the sea water leaving the heat exchanger, through which the fresh water is circulated in problem 2, if the temperature in the sea is 81° F and the capacity of the sea-water circulating pump is 100 gal./min. *Ans.* 99.7° F.

11–7. Questions. 1. What type of strainers and filters are used in the fuel-oil system?

2. What lubricating-oil filter systems are used in the Navy?

3. What is the purpose of cooling diesel engines?

4. What are the advantages of the closed cooling system?

5. What parts may be present in a diesel air-intake system?

6. What are the possible functions of the exhaust system?

7. In what type of mufflers are spark arresters used?

CHAPTER 12

AUXILIARIES

12-1. Definition. Auxiliaries include equipment which, generally speaking, is not part of a diesel engine itself. In most cases, it is not made by the engine builder, but by a manufacturer specializing in this type of equipment. Frequently the items of auxiliary equipment are also used in other than diesel-engine installations. The auxiliary equipment discussed in the present chapter includes: (1) blowers, (2) pumps, (3) heat exchangers, and (4) piping.

12-2. Blowers. As stated in Sec. 11-4, blowers used with diesel engines are either of the engine-driven positive-displacement rotary type, or of the centrifugal type, usually driven by an exhaust-gas turbine.

Positive-displacement blowers. The *Roots blower* consists of an oblong housing with flat-end plates, inside which rotate in opposite directions, two accurately machined *rotors*, also called *impellers*. These impellers, mounted on two parallel shafts, have extending parts called *lobes*, the impellers of Fig. 12-1 having three lobes. One of the impellers is driven from the engine crankshaft through a train of gears or a silent chain; the other one is driven by a pair of spur gears, called *timing gears*, from the first impeller shaft. These gears *synchronize* the rotation of the impellers, that is give them their *correct relative positions* at all times, and thus prevent them from touching one another. All clearances—namely: (1) between the end cover or plates of the housing and the ends of the impellers, (2) between the cylindrical walls of the housing and the tips of the lobes, and (3) between the surfaces of the impellers—are made very small in order to prevent an appreciable air leakage from the discharge side to the intake side. The impellers rotate as shown in Fig. 12-1. The entering air is trapped in the space enclosed by the curved side of an impeller, and the side and end covers of the housing. As the impeller rotates further, the space containing the trapped air is opened to the air receiver or manifold. The resistance to the air flow between the receiver and engine cylinder raises the pressure in the receiver, and the air from it, flowing back into the blower, compresses the incoming air.

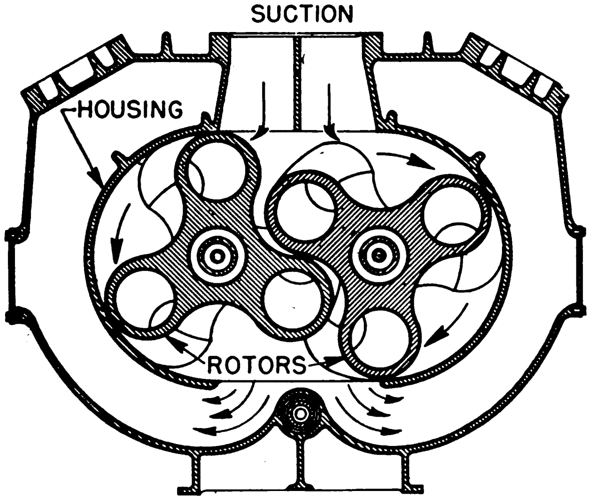


Figure 12-1. Roots-type blower.

Blowers of the Roots type differ in their construction. Some have impellers with two, some with three lobes; also some have the lobe surfaces *cylindrical*, others *helical*, also called *spiral*. The object of using three lobes and helical surfaces is to obtain a more even, less pulsating air delivery, and thus to decrease the noise of the blower, which becomes particularly objectionable at higher speeds. Helical surfaces also give a whirling motion to the air as it leaves the blower which aids in creating turbulence in the combustion chamber. Fig. 12-1 shows a three-lobe blower with helical surfaces.

Whitfield blower. Another positive-displacement rotary blower, of the Whitfield type, is shown in Fig. 12-2. It consists of two rotors connected by timing gears. One of the rotors has four spiral lobes, the other resembles a two-lobe, helical Roots-blower

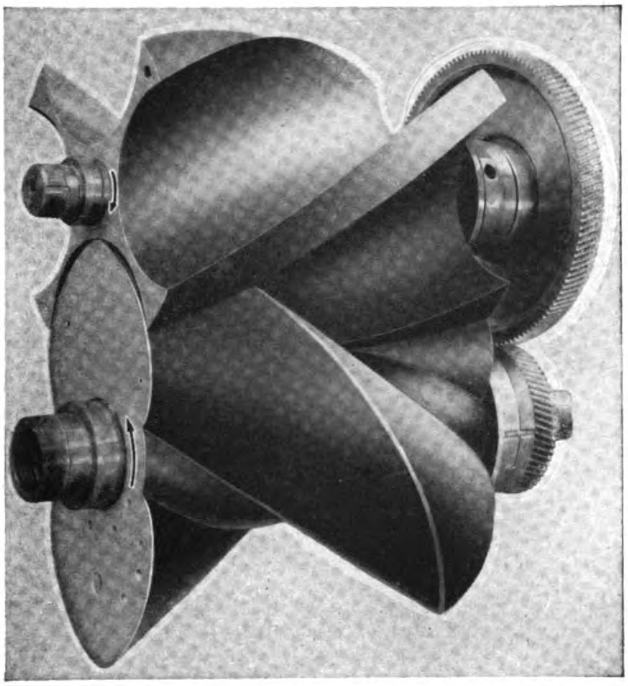


Figure 12-2. Impellers of a Whitfield air blower.

rotor, and turns at twice the speed of the first rotor. They rotate in a housing similar to a Roots housing, but the intake and discharge openings are in the end plates. With the rotation as indicated on Fig. 12-2, the intake opening is in the left plate, the discharge in the right plate. The four-lobe rotor traps the air and moves it from the intake toward the discharge. The two-lobe rotor turning in the opposite direction, gradually reduces the space in which the air is confined, and thus compresses it before it is discharged. The efficiency of this relatively new-type blower is claimed to be higher than that of the Roots type, and with the four intakes per revolution, it gives a more even flow of air, and should be less noisy. At present this blower is used only on one Navy double-acting engine type, the H.O.R., but its advantages may induce engine builders to employ it in other engines, too.

Exhaust-driven centrifugal blowers. A centrifugal blower, Fig. 12-3, consists of a rotor or impeller, which has *blades* or vanes around its circumference and is similar to the impeller of a centrifugal pump, and a case or *scroll* which contains fixed *vanes* to guide the flow of air. The air is sucked in axially from the left, as shown in Fig. 12-3. The pumping action results from the centrifugal force which throws the air outward into the scroll and increases the air pressure. The impeller is driven by a gas turbine. The impeller and the turbine disk are secured to a common shaft. The exhaust gases are admitted axially from the right, pass between guide vanes, and then through channels formed by the vanes of the turbine disk. In doing so, they give up part of their kinetic energy. A turbo-blower is mechanically independent of the engine which it serves. Its only connection to the engine is the exhaust pipe to the turbine, and the air pipe from the blower to the intake system of the engine. Since the temperature of the exhaust gases entering the turbine at full load usually is 700 to 800° F and in some engines is even up to 1,100 ° F, turbine parts must be heat-resisting, and a proper allowance must be made for their expansion. Parts of the housing have water jackets for cooling and the water should be circulated, if possible, before the engine is started and after it is secured. The speed of the turbo-blowers varies from about 8,500 rpm. to 10,000 rpm. for larger engines, and up to 16,000 rpm. to 20,000 rpm. for smaller engines. Turbo-blowers are used only for supercharging four-stroke engines.

12-3. Pumps. A diesel engine installation uses a large number of pumps of various sizes and types to deliver (1) fuel oil to the injection system and to circulate (2) lubricating oil, (3) fresh water for engine cooling, and (4) sea water through various heat exchangers.

Fuel pumps. The fuel-transfer or booster pump is attached to and driven by the engine. In the larger

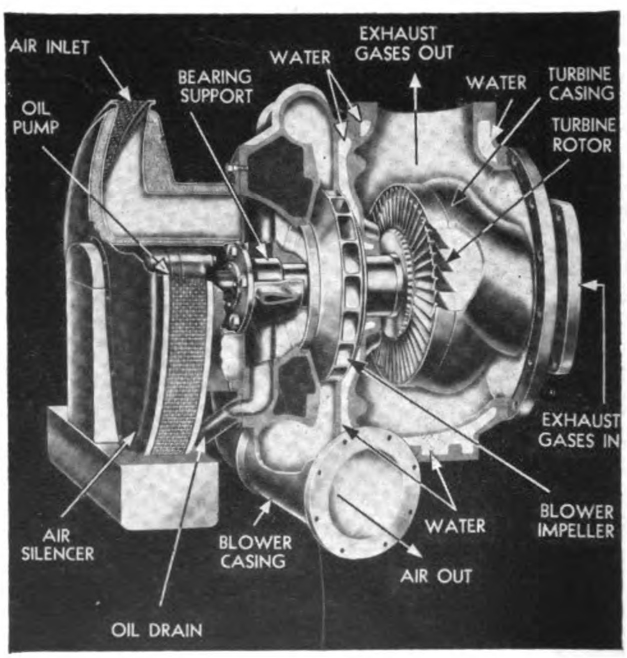


Figure 12-3. Exhaust turbine driving centrifugal blower.

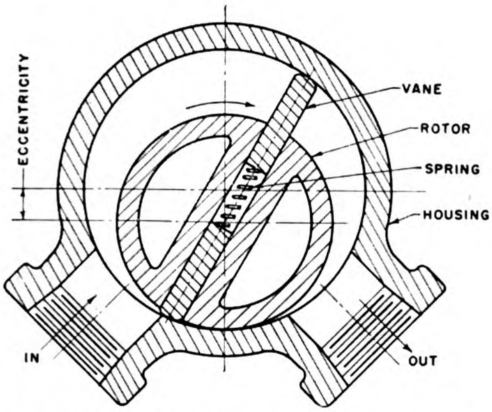


Figure 12–4. Rotary pump with sliding vanes.

engines this pump is of the positive-displacement rotary type whose rotors are straight spur gears keyed to the drive and the driven shafts. Its action is identical with that of a Roots blower.

The gears, shafts, and end plates are made of nitrided alloy steel with hardened surfaces. The pump housing is made of nickel-chromium cast iron. The shafts run in bronze bushings pressed into the end plates. Some engines use positive displacement rotary pumps with an eccentric rotor and two vanes sliding in it, as shown in Fig. 12–4.

With direct reversible engines, the direction of rotation of direct-connected rotary pumps is also reversed. The flow of oil is maintained in the required direction by special check valves which automatically connect the suction and discharge sides of the pump with the corresponding oil lines.

Some fuel-injection pumps, mostly those used on smaller engines, have a booster pump with a reciprocating plunger built as part of the injection pump. In other injection pumps, like the Bosch jerk pump, the fuel booster pump is a separate unit with a reciprocating piston, but is bolted to the side of the injection-pump housing and is operated by a cam on the camshaft of the injection pump, Fig. 12–5. The motion is transferred to the pump piston 4 through a roller 6 and tappet 1. The plunger and tappet are returned by two coil springs; 9 is the suction and 10 the discharge valves.

Lubricating-oil pumps. These pumps are always of the positive-displacement rotary type, Fig. 12–6. Their rotors are sometimes a pair of spur gears, but more often a pair of helical or herringbone gears. The gears are made of high-grade alloy steel and the pump body is made of alloy cast iron. The pumps are generally driven at 1,100 rpm. to 1,300 rpm. and have a capacity ranging from 6 to 100 gal. per min., depending upon the size of the engine. When using a gear pump for pumping lubricating oil, care must be taken to prevent the pump from becoming air bound. Due to a comparatively high viscosity of the oil it is difficult to clear the pump of air once it is air bound. Large engines and double-acting engines receive special treatment in the lubrication of the pistons and cylinder walls. Plunger-type pumps are used to deliver oil to these surfaces. These pumps are independent of the lubrication system proper.

Water pumps. Most modern diesel engines use centrifugal pumps for circulating both the fresh and sea water. The usual pump consists of an impeller with vanes curved in the direction opposite to that of rotation, and a spiral housing or scroll with the cross section increasing toward the outlet, Fig. 12–6. The water inlet is at the center, axial, the outlet is tangential. The pressure is due to the centrifugal force, which during the rotation of the impeller, throws the water toward the tip ends of the vanes. The flow of the water through the housing and into the pump discharge is also assisted by the tangential velocity which is imparted to the water. The pumps operate at speeds from about 1,400 rpm. to 3,500 rpm., depending on the size and design.

Water pumps used on direct-reversible engines usually have straight radial vanes and a concentric housing, Fig. 12–7. The efficiency of a pump so constructed is lowered by the radial shape of the vanes, and in order to obtain the necessary pressure and capacity, the pump speed must be somewhat higher than in an ordinary centrifugal pump. Some direct-

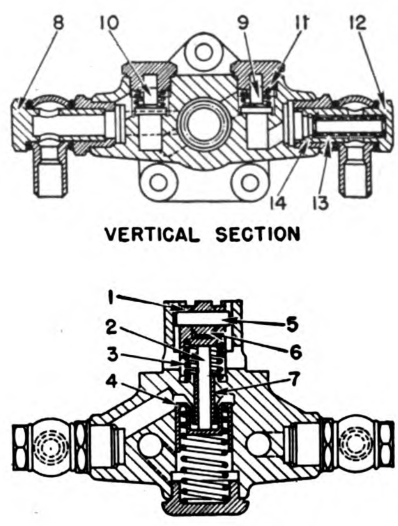


Figure 12–5. Reciprocating fuel transfer pump.

reversible engines have a unidirectional pump drive, i.e., the pump is connected to the engine by means of a special reverse gear, whose driving shaft, connected to the pump shaft, is made to turn in the same direction regardless of the direction of rotation of the drive shaft which is connected to the engine.

Water pumps are driven from the engine crankshaft by gears or chains.

For sea water circulation, small high-speed and some medium-size engines use positive-displacement gear-type pumps. The advantage of this type of pump is a more or less constant delivery per revolution, irrespective of speed.

As mentioned in Sec. 11–4, the pressure of a centrifugal blower or pump increases as the square of the rpm., and the delivery increases as the cube of the rpm. This means, that for variable-speed engines, the discharge of water-circulating pumps drops rapidly with a lowering of the engine speed. Fortunately, in Navy variable-speed engines the load comes from turning the propeller, and the power absorbed by the propeller itself drops as the cube of the rpm., so that the cooling requirements thus remain more or less in agreement with the pump performance.

In order to prevent corrosion, the impellers and bodies of the water pumps are made of bronze or gun metal, and the shaft of monel metal.

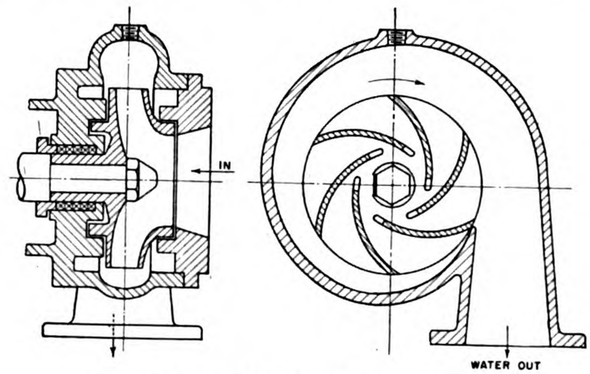


Figure 12–7. Centrifugal water pump.

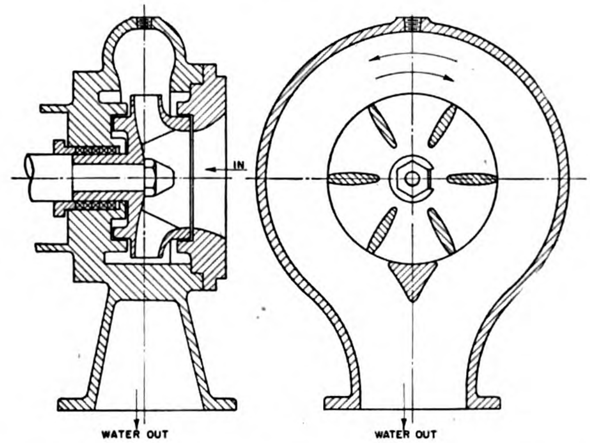


Figure 12–8. Centrifugal pump for rotation in either direction.

12–4. Heat exchangers. A heat exchanger is a device through which two fluids, and sometimes air are passed, separated by a metal wall and having different temperatures. The hotter fluid gives up heat to the colder one so that the temperatures of the fluids, when they are leaving the apparatus, approach one another. Depending upon its purpose, a heat exchanger may be called a cooler or a heater. In Navy diesel engines, heat exchangers are used principally as coolers, to cool lubricating oil, fresh water, and sometimes air. However, in some cases they are used as heaters, to heat water by hot exhaust gases.

Construction. Three basic cooler construction types are in use with Navy diesel engines. The majority are of the so-called *radiator* type with flat tube-like elements; some smaller engines use the so-called *plate-type* construction for cooling lubricating oil only; some larger engines use regular *shell-and-tube* coolers.

Radiator-type coolers. The Harrison radiator is made up of a tube bundle, or *core*, and an enclosing *case* which is divided into three pieces, a center and two end covers, as shown in Fig. 12-9. The tubes have oblong sections, about 1-1/16 in. by 3/16 in., while the ends are brazed into tube sheets which are parts of the admission and discharge headers. The cooled

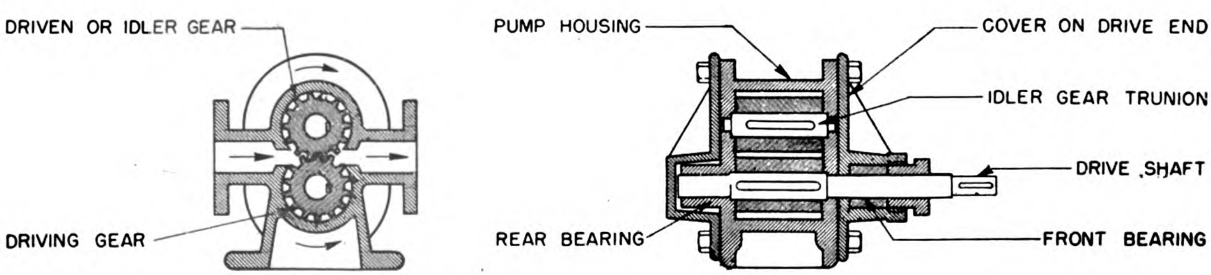


Figure 12–6. Lubricating oil pump, gear type.

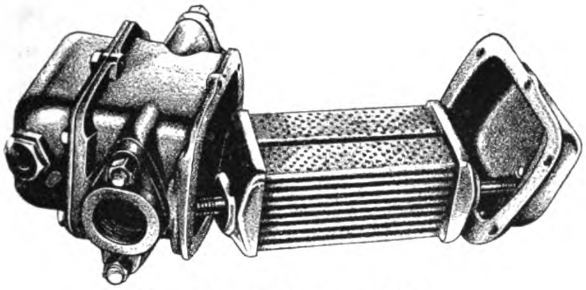


Figure 12–9. Harrison radiator.

medium, water or oil, enters one header, passes through the tubes to the other header and from it out to the engine. The cooling medium, usually sea water, enters the housing on one side and leaves at the opposite side. The flat tubes are reinforced by rows of round cross tubes brazed into the flat sides. These cross tubes or struts also increase materially both the inside and outside contact areas with the fluids. Fig. 12–9 shows a lubricating oil cooler with two rows of nine tubes each, and Fig. 12–10 a fresh water cooler with four rows of 11 tubes each. In Fig. 12–10 the tube element assembly and one section are shown.

When used in a lubricating-oil cooler, the tubes are sometimes made without the cross tubes but with inside baffles which create turbulences in the oil flow, and thus increase the heat transfer from the oil to the water on the outside.

Plate-type cooler. A plate-type cooler consists of hollow plates, 3 in. by 4 in. or 3 in. by 6 in., with the flat side about 3/16-in. apart, and with inside baffles which reinforce the flat sides and give better heat transfer. The oil is admitted through a large hole at one end of the hollow plate and expelled through a hole at the other end; the cooling water passes along the flat sides. Several plates placed side by side provide a parallel passage for the oil. The size and number of plates used depend upon the amount of oil to be circulated. In Fig. 12–11 the element and one plate section are illustrated.

Shell-and-tube-cooler. A typical cooler is shown in Fig. 12–12. The oil or fresh water to be cooled enters one header, passes through the tubes and leaves from the other header; the cooling sea water is admitted to the shell at one end, passes in a zigzag course directed by the baffle plates, and leaves the shell at the opposite end. When cooling fresh water, it sometimes is passed through the shell, and the sea water is passed through the tubes.

Zinc electrodes. Marine cooling systems must be provided with zinc electrodes mounted in the sea-water inlet to control electrolysis, which occurs in the cooling system from stray electric currents in the sea-water line. The purpose of the zinc is to provide a terminal which will ground the stray current, and thus concentrate any electrolytic action by corroding the zinc rather than the other parts of the cooling system. The zinc electrodes normally will be corroded rather fast; they must be inspected at regular intervals and exchanged before they become too small, normally every three to six months. In the radiator-type heat exchangers, the zinc is inserted in the form of bolts, called zinc *pencils*, projecting inside. In shell-and-tube heat exchangers, the zinc pieces are plates fastened inside the shell, if sea water is circulated through the shell; if sea water is circulated through the tubes, zinc pieces are fastened inside the header.

Pressures. During the operation of the coolers the pressure of the cooling water should always be less than that of the oil, to prevent leakage of the water into the oil system should the cooler develop any leaks. For the same reason, in water coolers the sea-water pressure should be kept lower than the fresh-water pressure. With some engines, however, this requirement is not fulfilled.

12–5. Piping. *Resistance.* Every pipe presents a certain resistance to the flow of the fluid which it

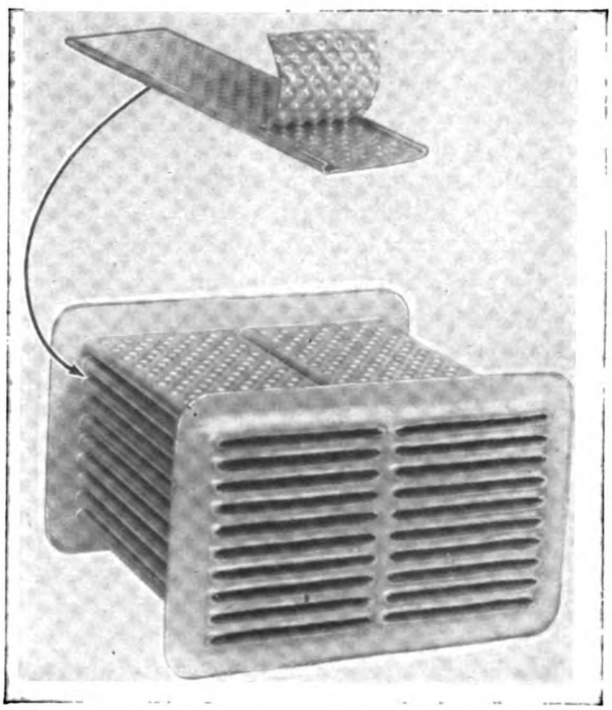


Figure 12–10. Strut tube heat exchanger.

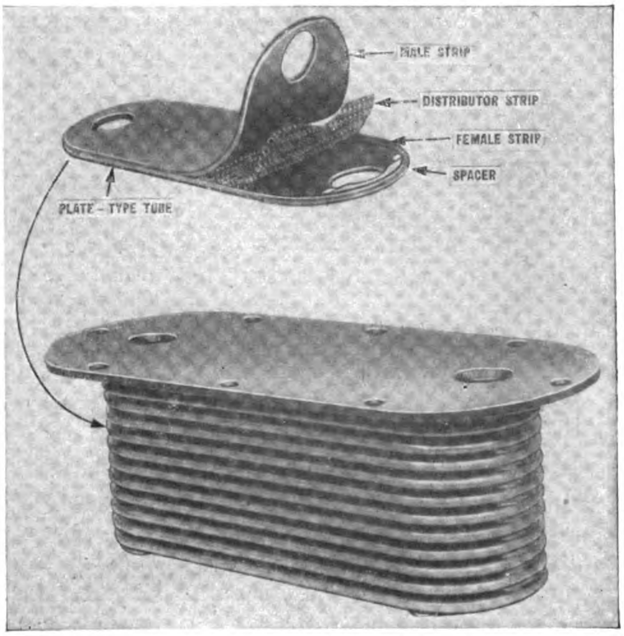


Figure 12–11. Plate assembly of lubricating oil cooler.

handles. The resistance of flow increases approximately as the square of the velocity of the fluid. Since the velocity is inversely proportional to the cross-section area, any reduction in area or size of the pipe will increase the flow resistance and hence decrease the flow rate. The resistance also increases with the length of the pipe and with every elbow and valve through which the fluid must pass. Different valve types present different resistances, the resistance of an ordinary globe valve, for example, being higher than of a gate valve. The resistance in a pipe can be observed by a drop of pressure in the pipe in the direction of the flow. In renewing a pipe line, care must be taken not to increase the resistance by making some changes in the original layout.

Flexible piping. Pieces of flexible piping or of rubber hose are sometimes inserted in lubricating-oil and soft-water pipe lines to reduce the transmission of engine vibrations, or to prevent breaking of pipe lines when the engine is installed on flexible vibration-isolating cushions, while tanks or heat exchangers are fastened to a solid foundation.

Relief valves. In many places a certain pressure in the pipe line must be maintained either for the sake of keeping up a definite rate of flow, or for the protection either of some piece of equipment or of the pipe itself. In order to make sure that the pressure will not exceed a certain desired amount, pressure-relief valves are installed as mentioned in other chapters.

A typical pressure-relief valve is shown in Fig. 12–13. The *spring* is adjusted by means of an *adjusting screw* to push the valve or disk with a certain pressure down on its seat. When the pressure under the disk exceeds a predetermined value, the disk is lifted, compressing the spring slightly and a certain amount of

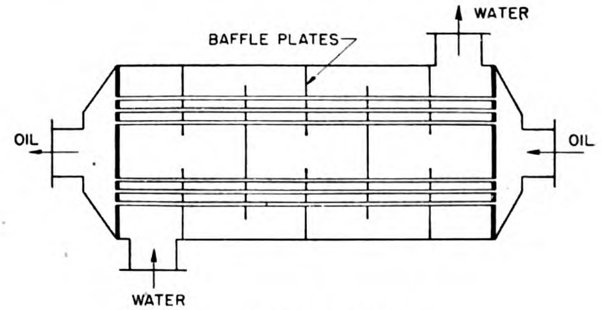


Figure 12–12. Oil cooler.

the liquid is discharged into the *bonnet* and from there into a low-pressure line. This discharge lowers the pressure under the disk, and therefore the spring pushes it down, until the pressure rises again. The *regulating ring* reduces the discharge area and makes the discharge slower, preventing a sudden large discharge, and thus helping to maintain a more uniform

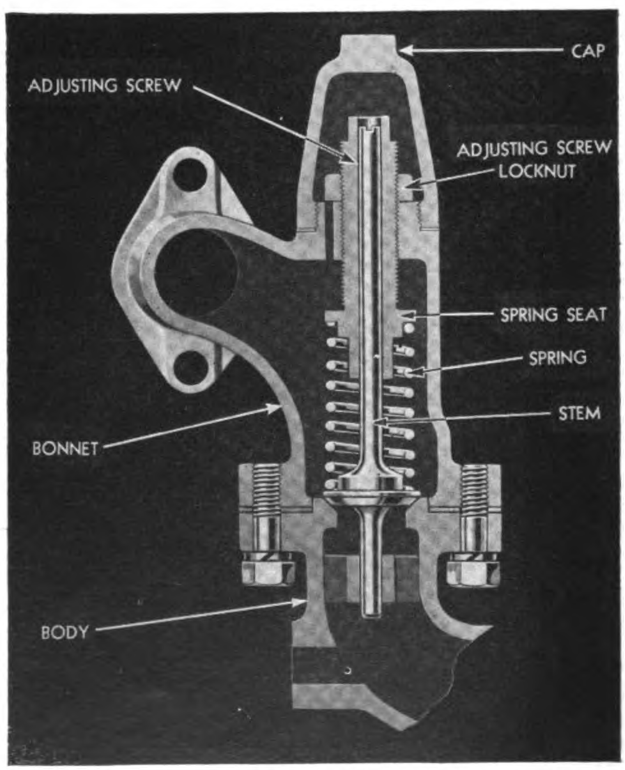


Figure 12–13. Fuel oil relief valve.

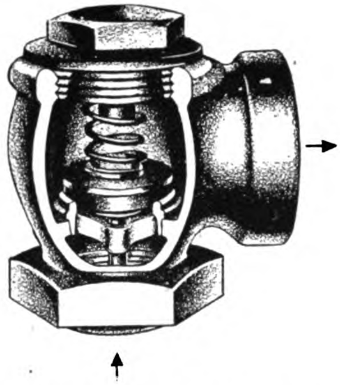


Figure 12–14. Check valve.

pressure in the pipe line. The regulating ring is only a refinement and many relief valves do not have it.

A *check valve* insures that the flow in a pipe is maintained in a certain direction. As already stated, whenever a liquid moves in a pipe, the pressure *behind* a certain point is always *smaller than before* it, the difference of pressures being the force which pushes the liquid. This pressure difference lifts a check valve, Fig. 12–13, against the force of gravity and the spring, and keeps it open. If the flow stops, the pressure before the check valve drops, and the weight of the disk, assisted by the spring, will close the valve and prevent the liquid from flowing in the opposite direction. Check valves are made to be installed in a horizontal or vertical pipe, or are to be used instead of an elbow fitting as shown on Fig. 12–14. This last one is called an *angle* check valve.

Temperature regulators. In order to maintain approximately constant temperature in the engine under various load conditions, Navy diesel engines have automatic temperature regulators. These regulators are automatic valves which are operated by *thermostatic elements* set to open or close at the temperature desired. A thermostatic element consists of a corrugated brass or monel metal pipe with deep corrugations called bellows. The inside of the bellows is filled with a volatile liquid such as ether or alcohol, which expands rapidly when the temperature of the water, in which the element is inserted, goes up. The large number of deep corrugations increases the motion of the free end of the bellows in respect to the fixed end, and thus makes the element more sensitive to a change of volume and pressure of the inside liquid with respect to a change of its temperature. In small engines, the expansion or contraction of the liquid in the element closes or opens a valve in the pipe fastened to the element, and thus decreases or increases the water flow through the heat exchanger. In large engines, the temperature-regulation valve operated by the bellows is placed where convenient in the fresh-water piping, and the action of the bellows is obtained by remote control. The remote control consists of a steel bulb filled with a volatile liquid, inserted in the water outlet from the engine, and connected to the bellows by a fine tube which transmits the pressure change in the bulb, as indicated on Fig. 11–11.

Devices other than bellows are used to operate bypass valves. One of them consists of a strip made up of two metals of different coefficients of heat expansion and forming a spiral coil. The outer end is fastened solidly, and the inner end is fastened to a shaft. When the temperature of this bimetallic coil changes, its free inner end moves, rotates the shaft, and opens or closes a flat-hinge-type valve. Another device consists of a cylinder and piston with some readily evaporating substance between them. When the substance melts and evaporates, it moves the piston and thus operates a disk valve. The disk valve is returned toward the seat by a spring when the pressure under the piston begins to drop.

12–6. Questions. 1. What types of blowers are used for scavenging two-stroke engines?

2. For what purpose are centrifugal blowers used when driven by an exhaust-gas turbine?

3. What is the difference between centrifugal water pumps used with engines rotating in one direction and those used with direct-reversible engines?

4. For what purposes are heat exchangers used with diesel engines?

5. What is the object of inserting zinc electrodes in sea-water cooling systems?

6. What are the purposes of using flexible piping?

7. What is meant by a thermostat or thermostatic regulator?

CHAPTER 13

ENGINE CONTROLS

13–1. Diesel engine loads. To control an engine means to keep it running at a desired speed, either in accordance with, or regardless of the changes in the load carried by the engine. The control of an engine depends upon two factors: its own performance characteristics, and the type of load which it drives. The total load of an engine consists of two parts: (1) the *internal* load, or friction horsepower due to the friction, windage, and pumping losses within the engine itself and required by the attached engine-driven auxiliaries; and (2) the *external* load, connected to the main drive shaft which takes the useful or brake horsepower output of the engine.

As explained in Sec. 6–2, the indicated horsepower is equal to the total load, or the sum of the friction and the brake horsepower. The indicated horsepower necessary to balance this total load at any speed depends upon the pressures developed during combustion of the fuel and the resulting mean indicated pressure. The combustion process, and the mean pressure developed by it, depend primarily upon the quantity of fuel injected on the firing stroke, if all other conditions remain substantially the same.

Most diesel fuel-injection systems are designed so that the amount of fuel delivered at a fixed setting or position of the fuel control is approximately the same per injection, regardless of the engine speed. The total load at any speed, therefore, can be balanced by regulating the fuel control to the setting required to give the necessary mean indicated pressure. In other words, the setting of the fuel control regulates the speed of a diesel engine under any load.

The frictional forces or the internal load of an engine increase rapidly with an increase in the engine speed. This also is usually true of the external load, so that, in general, greater mean indicated pressures are required to balance the total load at higher speeds. An example of this type of external load is a direct-connected propeller. The power required to drive a propeller is shown in Fig. 13–1, together with the friction power, and their sum gives the indicated power required from the engine at various speeds. With this type of total load, the speed of a diesel engine will increase only if more fuel is injected and will decrease if less fuel is injected. This type of load, which increases with speed more rapidly than the engine output increases with speed, is essentially self-governing under normal operation. This is true because any fuel setting will provide the combustion pressures which balance the total load on the engine only at one speed. The speed under these conditions, then, will vary in proportion to the fuel setting, and if the load conditions were perfectly constant, manual regulation of the fuel control would provide adequate speed control. However, the propeller load on a ship is not constant even at a constant engine speed, because of the pitching of the ship and the uncovering of the propeller in a seaway. To prevent overspeeding of the engine when the propeller load suddenly decreases, governors are provided with practically all Navy diesel engines.

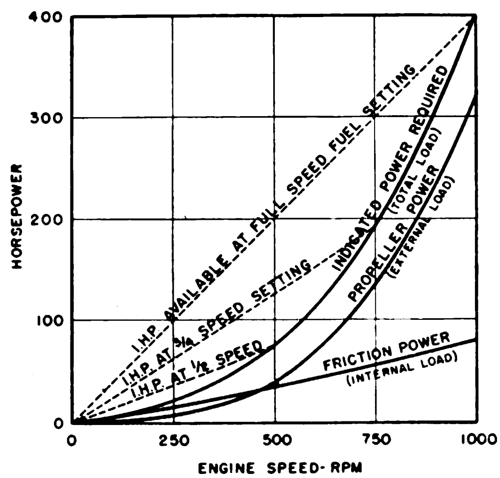


Figure 13–1. Loads on engine connected to a boat propeller.

When an engine is operating at constant speed, the internal load remains approximately the same. Under these conditions, the combustion pressures necessary to balance the total load will vary directly with changes in the external load. An example of this type of load is a direct-connected electric generator, which normally must operate at constant speed. An increase or decrease in electric load connected to the generator will require correspondingly greater or less combustion pressures to keep the engine speed constant. It is necessary, therefore, in order to maintain constant speed operation, to change the amount of fuel injected in proportion to the external load changes.

When two or more engines are connected electrically or mechanically to the same load, they cannot independently vary in speed. Examples of this type of load are the multiple-unit diesel-electric power systems, and the multiple-unit propulsion systems in which two or more engines drive a single propeller shaft. Under these conditions, the fuel injected in each engine must be regulated so that the total load is distributed proportionally between the engines while their relative speeds are maintained.

When an engine is running without a load at slow idling speed, the friction power is relatively small. The quantity of fuel, therefore, which must be injected per firing stroke to balance the internal load, is extremely small, especially in small engine cylinders. These small fuel quantities are difficult to inject regularly and as a result the *idle-speed stability* is poor, i.e., the engine speed will vary. Furthermore, a slight adjustment of the fuel-control setting at idle position will cause relatively large changes in the amount of fuel injected. This, in turn, results in comparatively large changes in engine speed, making it very difficult to regulate an engine under idling conditions. The problem of regulating the fuel control to meet these various operating loads is often further complicated by certain characteristics of the fuel-injection system itself. Under these conditions it is not possible to regulate manually the fuel control of a diesel engine satisfactorily and some type of governor is necessary.

13–2. Governors. *Functions.* A governor is essentially a speed-sensitive device, designed to control the speed of an engine under varying load conditions. The type of load and the degree of control desired determine the kind of governor to be used. Governors for diesel engines may be classified according to their primary functions as follows:

1. *Constant-speed governors:* to maintain the same engine speed from no load to full load.

2. *Variable-speed governors:* to maintain any desired engine speed from idle to top speed.

3. *Speed-limiting governors:* to control the minimum engine speed and to limit its maximum speed, or to limit the maximum speed only.

4. *Load-limiting governors:* to limit the load which the engine will take at various speeds.

Some governors are designed to perform two or more of the above functions by incorporating their operating mechanisms in the same unit.

Governing characteristics. Governors employed for controlling the speed of diesel engines must have certain characteristics to fit the type of load which the engine is to drive. The principal characteristics which determine the degree of governor control of the engine may be defined as follows:

1. *Speed droop* is the decrease in speed of the engine from no load to full load expressed in rpm., or as a per cent of normal or average speed, usually the latter.

2. *Isochronous governing* is maintaining the speed of the engine truly constant, regardless of the load—in other words, with perfect speed regulation or zero speed droop.

3. *Hunting* is the continuous fluctuation of the engine speed, slowing down and speeding up, from the desired speed, due to *over-control* by the governor.

4. *Stability* is the ability of the governor to maintain the desired engine speed without fluctuations or hunting.

5. *Sensitivity* is the change in speed required before the governor will make a corrective movement of the fuel control, and is generally expressed as a per cent of the normal or average speed.

6. *Promptness* is the speed of action of the governor. It may be expressed in terms of the time in seconds required for the governor to move the fuel control from no-load to full-load position. Promptness depends upon the power of the governor—the greater the power, the shorter the time required to overcome the resistances.

Speed droop. If n_1 designates the no-load speed, n_2 the full-load speed, often called normal speed, then the speed droop in per cent can be calculated from the expression:

$$\text{Speed droop} = 100(n_1 - n_2) \div n_2 \qquad (13\text{–}1)$$

Often speed droop is referred to the average speed n which is evidently

$$n = (n_1 + n_2) \div 2 \qquad (13\text{–}2)$$

and using n, expression (13–2), instead of n_2 in the denominator of expression (13–2) gives:

$$\text{Speed droop} = 200(n_1 - n_2) \div (n_1 + n_2) \qquad (13\text{–}3)$$

Example 13-1. Find the speed droop of the governor if the engine has a normal speed of 1,400 rpm. and a no-load speed of 1,450 rpm. Compare both methods of computation.
By expression (13-1):
Speed droop = 100(1,450 − 1,400) ÷ 1,400 = 3.57 per cent.
By expression (13-3):
Speed droop = 200(1,450 − 1,400) (1,450 + 1,400) = 3.51 per cent.
The difference is negligible.

Hunting. It is said that a governor is hunting if its flyweights and control sleeve do not assume at once a definite position corresponding to the speed, but move up and down. Hunting occurs due to a lag in action of the control mechanism, indicating poor sensitivity and resulting in a large speed change before any governor action takes place. The engine will slow down or speed up too much, therefore, before any corrective regulation of the fuel control is made. When the controls begin to move, they will continue to move even after the correct speed has been obtained, and this will result in an *over-correction* of the engine speed in the opposite direction. After that the governor will start to act in the opposite direction, etc. The engine speed will fluctuate or hunt due to this over-control by the governor.

Hunting may also occur due to the slowness of action of the corrective control mechanism as indicated by poor promptness, if the power of the governor is too small. This lag in action will permit too great a change in engine speed to occur during a change in engine load before the proper correction is made, even if the governor is sufficiently sensitive. Due to this lag, the speed of the engine will be changed too much in the opposite direction before the governor action stops. Poor promptness thus results in hunting, due to over-control of the governor in the same manner as with poor sensitivity.

Any method of reducing the friction in the operating mechanism of a governor will tend to increase its stability.

Sensitivity. If an engine is running and its governor is in equilibrium, it requires a considerable change in engine speed, due to a change in the load, before the governor will begin to act and adjust the fuel delivery to correspond to the changed load. This lag in governor action is caused by friction and lost motion in the governing mechanism, and is called *sensitivity*. Sensitivity is determined by testing an engine first with an increasing and then with a decreasing load. Due to lag in the governor action, the speed corresponding to a certain load which was obtained when the load was increasing is always lower than the speed corresponding to the same load when decreasing. Sensitivity is expressed numerically as the difference of the two speeds divided by their average, in per cent. Thus, the greater this figure, the less satisfactory is the sensitivity of the governor. Usually the difference between the two speeds is greatest near half-load conditions.

The sensitivity is calculated from an expression, which according to the definition, is identical with expression (13-3), if n_1 and n_2 are assumed to be the highest and lowest speeds with the same load.

$$\text{Sensitivity} = 200\,(n_{max} - n_{min}) \div (n_{max} + n_{min}) \quad (13\text{-}4)$$

Sensitivity is often defined also as fluctuation *above* and *below* the average speed, meaning that one-half of the difference of $(n_{max} - n_{min})$ is assumed as lying above and the other half below. With this definition, expression (13-4) changes to:

$$\text{Sensitivity} = \pm 100\,(n_{max} - n_{min}) \div (n_{max} + n_{min}) \quad (13\text{-}5)$$

Example 13-2. Find the sensitivity of the governor which at one-half load begins to act with an increasing load at 1,417 rpm., and with a decreasing load at 1,429 rpm.
According to expression (13-4):
Sensitivity = 200(1,429 − 1,417) ÷ (1,429 + 1,417) = 0.84 per cent.
By expression (13-5) sensitivity will be evidently ±0.42 per cent.

Types of governors. Practically all governors used on diesel engines for the Navy are of the type in which

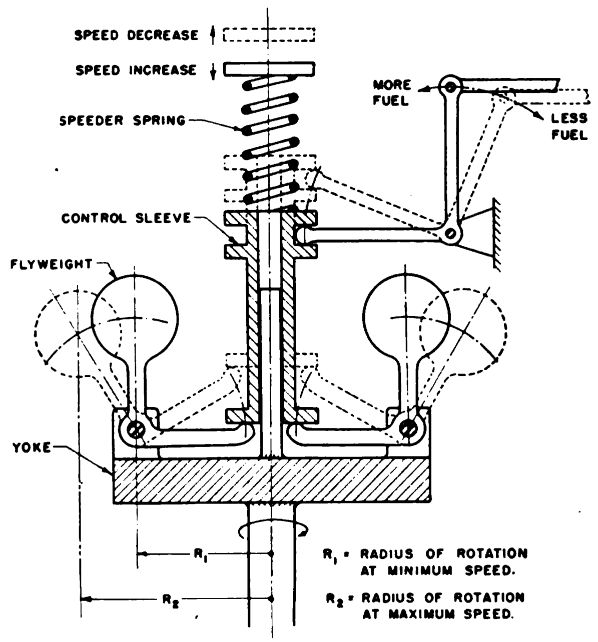


Figure 13-2. Elementary spring-loaded centrifugal governor.

the centrifugal force of the rotating weight is balanced by a helical coil spring. They are commonly known as *spring-loaded centrifugal governors*. Centrifugal governors may be classified into two main types depending upon the regulating force employed to operate the fuel control. These types are:

1. *Mechanical governors*, in which the centrifugal force of the rotating weights directly regulates the fuel supply by means of a mechanical linkage which operates the fuel-control mechanism.

2. *Hydraulic governors*, in which the centrifugal force of the rotating weights regulates the fuel supply indirectly by moving a hydraulic pilot valve controlling oil under pressure, which operates the fuel-control mechanism.

13–3. Spring-loaded centrifugal governor. A spring-loaded centrifugal governor is illustrated diagrammatically in Fig. 13–2, showing the low- and high-speed positions of the weights. The governor has two rotating weights, called *flyweights* or *flyballs*. These are fastened to the upper ends of ball-crank levers mounted on pivots at their corners to the *yoke*. The yoke is usually connected by gears to revolve with the engine. The inner ends, or *toes*, of the flyweight levers bear against the thrust bearing of the *control sleeve*, which operates the fuel-regulating mechanism. The *speeder spring*, often referred to simply as the governor spring, bears against the upper end of the control sleeve and tends to move it, together with the fuel-regulating mechanism, downward in the direction to supply *more fuel*. The centrifugal force acting outward on the flyweights has a tendency to move the control sleeve, together with the fuel-regulating mechanism, *upward* against the action of the spring, in the direction to supply *less fuel*. When the centrifugal force of the rotating flyweights is exactly balanced by the force of the spring, the control sleeve assumes a fixed position, the fuel-regulating mechanism remains at a certain setting, and the engine speed remains constant so long as the load does not change.

If, however, the load on the engine decreases, the engine will begin to speed up because the fuel-regulating setting for the original load will supply more fuel than is necessary for the reduced load. As the speed of the engine increases, it drives the governor faster also, and increases the centrifugal force of the flyweights. This increase in centrifugal force moves the control sleeve, in the direction to supply *less fuel*, compressing the spring further until the centrifugal force again is balanced by the increase in spring force. The reduction in fuel supplied prevents the engine from increasing in speed more than is necessary for the governor to operate the fuel-control mechanism the amount required to balance the reduced load.

If, on the other hand, the load on the engine increases, the engine will begin to slow down because the fuel supplied for the original load is less than is necessary for the increased load. As the speed of the engine decreases, the centrifugal force of the governor flyweights also decreases. This unbalances the spring force which then moves the control sleeve, in the direction to supply more fuel, until the decrease in centrifugal force is balanced by the decrease in spring force due to the increase in its length.

Constant-speed governors. With mechanical governors, in which the control sleeve is directly connected to the fuel-regulating mechanism, the spring length must be changed a certain amount for every change in setting of the fuel control. This requires a centrifugal force different from that of the flyweights, and consequently, a different engine speed in order for the governor to balance every change in load. Therefore, all mechanical governors have a speed droop and so cannot provide isochronous governing. By proper design, however, the speed droop of mechanical

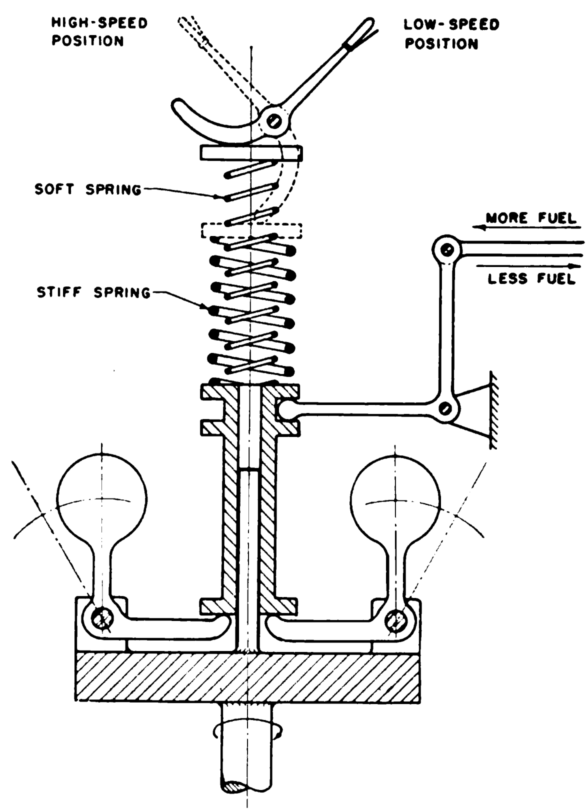


Figure 13–3. Two-speed centrifugal governor.

governors can be held to four to five per cent, so that for practical purposes such governors may be classed as the constant-speed type.

Variable-speed governors. If the control speed of an engine must be changed while in operation, a variable-speed governor is required which may be adjusted to maintain any desired control speed within the operating range of the engine.

The simplest method of obtaining variable-speed governing with spring-loaded centrifugal governors is to provide a means for varying the initial compression of the speeder spring as shown diagrammatically in Fig. 13–2. Thus, if the initial spring force is increased by compressing the spring to a shorter length, the engine speed must increase before the centrifugal force developed by the flyweights can balance the greater spring force. Or, if the initial compression of the spring is decreased, the engine speed necessary for the centrifugal force of the flyweights to balance the reduced force will be decreased.

Two-speed governors. In many installations, it is desirable that the governor control the engine so that it idles at low speeds rather than at high speeds under no-load conditions. In order to accomplish this with a mechanical governor it is necessary to provide two different springs. One is a soft spring, which takes a small force to compress it per inch, to provide better sensitivity and lower speed droop at low speeds; the other is a stiff spring, which takes a larger force to compress it per inch, to provide sufficient stability at high speeds.

The springs may be arranged to act either singly or in combination. An example of an elementary form of this type of governor is shown in Figure 13–3. For low-speed operation, the governor control is put in the position in which only the soft inner spring is acting, thereby providing better sensitivity and speed regulation under idle conditions. For high-speed operation, the governor control is put in the position in which both springs must act together, which provides better control under full-speed conditions.

Influence of friction. Friction has a pronounced effect on the characteristics of governors. The friction in a governor includes all the forces which oppose the motion of the control sleeve and the fuel-control mechanism. In a spring-loaded centrifugal governor, the force to move the control sleeve must be supplied by the difference between the centrifugal force of the flyweight, and the force of the speeder spring during a period of unbalance. In other words, the change in speed necessary to actuate the governor or make it operate depends directly upon the total frictional resistance which opposes the governor action. Any method of decreasing the forces which must be overcome by a governor to operate the fuel-control mechanism will, therefore, increase its sensitivity.

Every governor depends upon a change in speed for its *corrective action*, so-called because it *corrects* the engine speed. The extent of the speed change necessary to produce this action determines the amount of the resulting movement of the control mechanism. Thus, the more sensitive the governor, the less will be the corrective action required after a change of the engine load. The rapidity of the control movement, which depends on the promptness of the governor action, also influences the amount of corrective action required when the speed changes. Both these characteristics affect the stability or steadiness of the governor by their effect on hunting.

Shortcomings. Mechanical governors have the following shortcomings:

1. They have poor sensitivity, since the speed sensitive element also must furnish the force to move the engine speed control.
2. Their power is relatively small unless they are excessively large.
3. They have an unavoidable speed droop and therefore cannot provide real constant speeds, *necessary when driving A.C. generators which must be held to exact speeds.*

Practically all of these shortcomings of mechanical governors may be overcome through the use of hydraulic governors.

13–4. Elementary hydraulic governor. The only method of obtaining truly constant speed or isochronous governing from a spring-loaded centrifugal governor is to restore the speeder spring to its original tension after every speed change, regardless of the movement of the fuel-control mechanism. When the speeder spring is kept under the same tension, the centrifugal force of the flyweights necessary to balance it will be developed only at one corresponding speed. The governor, therefore, must regulate the fuel supply to keep the engine at the same speed regardless of load. This can be accomplished with a spring-loaded centrifugal governor by using an indirect connection between the control sleeve and the fuel-control mechanism. Such an indirect connection may be provided by an independent energy source to operate the fuel-control mechanism, which is most conveniently furnished by oil pressure in hydraulic governors, produced by a special pump.

An elementary form of hydraulic governor is illustrated in Fig. 13–4. The speed-sensitive element in the governor is a pair of spring-loaded flyweights and a helical coil spring. The speed-sensitive element operates a pilot valve which controls the flow of oil to and from a hydraulic power piston shown at right. When the governor is operating at control speed, the lower end of the plunger of the pilot valve registers with and just closes the ports in the pilot-valve bushing and there is no flow of oil, Fig. 13–4a. When the governor speed rises above the speed at which the pilot ports are closed, the flyweights move out, raising the pilot-valve plunger, Fig. 13–4b. This opens the port from the power piston to the drain into the sump. The spring on the power piston forces the power piston down, toward no-load position. The oil displaced drains through the center of the pilot-valve bushing. The fuel-control mechanism is connected to the power-piston rod end. When the governor speed decreases below the control speed, the flyweights move in, lowering the pilot-valve plunger, Fig. 13–4c. This opens the port to the power piston and connects it to a supply of oil under pressure. This pressure oil acts on the power piston, forcing it up toward full-load position.

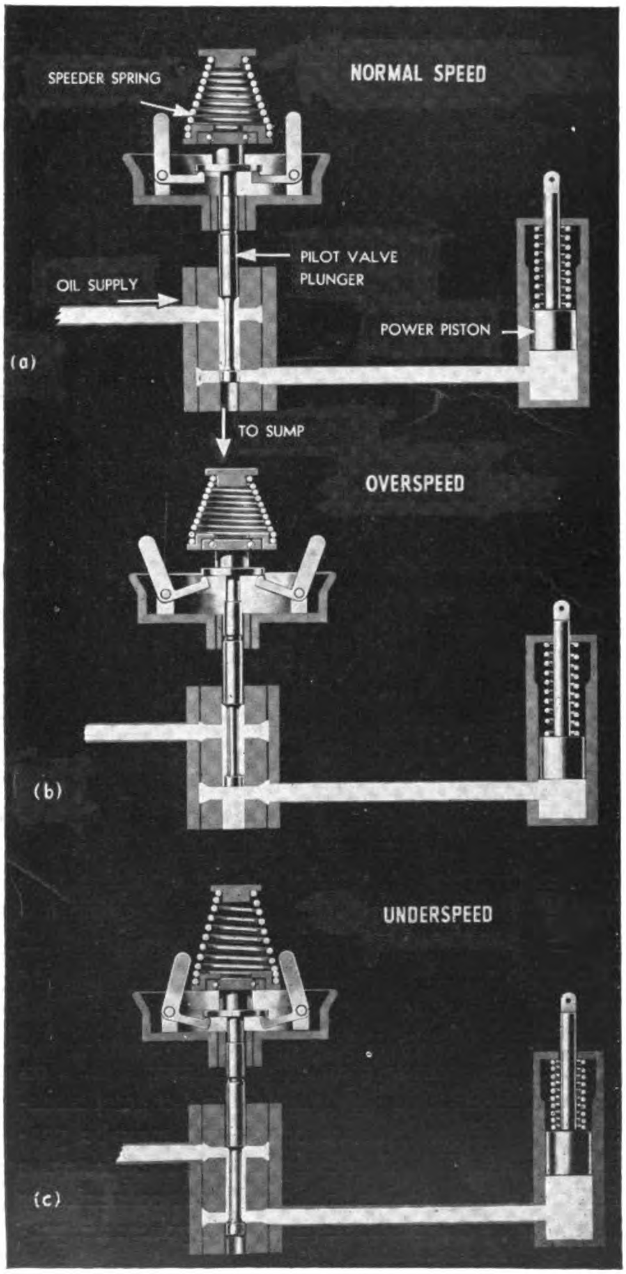


Figure 13–4. Elementary hydraulic governor.

The length of the lower end of the pilot-valve plunger is exactly the same as the width of the ports in the pilot-valve bushing. Since the plunger is moved directly by the speed-sensitive element, there is one speed and only one speed at which the ports will stay closed. Fig. 13–4 shows an elementary isochronous or constant-speed hydraulic governor. From the above discussion it is clear that with this governor there is no direct relation between the governor speed and the setting of the engine fuel controls. If the governor speed drops below the fixed, or *control* speed, the power piston will be moved upward and will increase the fuel supply. If the governor speed rises above control speed, the power piston will be moved down, toward decreased fuel. The power piston will be set at any required position between no-load and full-load. The engine will operate, under equilibrium conditions, exactly at control speed.

The pilot valves are constructed so that the fluid pressures are balanced and produce no thrust on the plungers. The oil pressure applied to the recess of the pilot-valve plunger in Fig. 13–4 acts equally on the ring-shaped areas at both ends and no axial force is produced. The plungers in governor pilot valves are usually rotated in the bushings to maintain an oil film between these parts and thus reduce friction and prevent sticking. The holes in the pilot-valve sleeve are arranged in pairs so that there is no side thrust exerted on the plunger. The lower fitted length of the plunger, called the *land*, exactly covers the holes in the bushing, so that a slight movement of the plunger starts a change in the fuel setting. All of these factors contribute to the high sensitivity of hydraulic governors, some of which will respond to a speed change of 0.01 per cent. This high sensitivity makes it possible for a hydraulic governor to maintain a control speed with great precision. Sensitivity is also important because it permits the governor to act immediately when a speed change is just beginning and thus prevents developing of larger speed fluctuations.

13–5. Actual hydraulic governors. *Compensation.* As explained in Sec. 13–3, hunting due to overshooting will result if an engine returns to control speed after a speed change, and the fuel controls are set as required during the speed change. The simple hydraulic governor has the same fault. As long as the speed is above or below the control speed, the simple hydraulic governor will continue to adjust the fuel system, to decrease or increase the delivery of fuel to the engine. There is always a lag between the moment that a change in fuel setting is made and the time the engine reaches a new equilibrium speed. Therefore, the engine will always return to control speed with the fuel delivery over-corrected, and hunting due to overshooting will result.

To avoid overshooting, a governor mechanism must anticipate the return to normal speed, and must discontinue changing the fuel control setting slightly before the new setting required for sustaining the control speed has actually been reached. A mechanism which enables a governor to anticipate the return to control is termed a *compensating device.* Every hydraulic governor must have some type of compensating device.

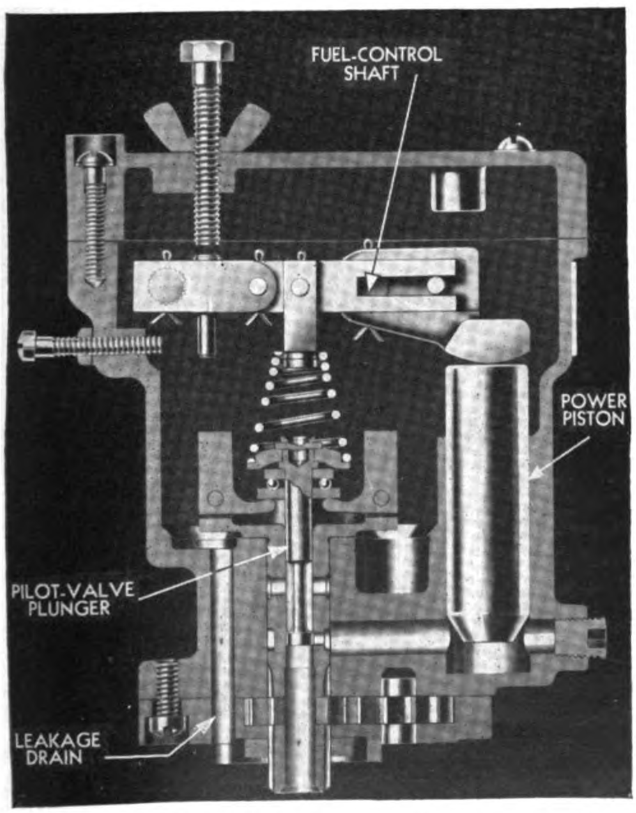


Figure 13–5. Hydraulic governor with compensation.

The simplest method of compensating for a hydraulic governor is to provide a speed droop with increase in load. While this method does prevent truly isochronous governing with this type of hydraulic governor, the speed droop can be held to a minimum and the governor still possesses the advantages of fine sensitivity and large regulating forces.

One mechanism employed to provide this compensation is illustrated in Fig. 13–5. The lever controlling the tension of the speeder spring is moved by linkage with the fuel-control mechanism. In operation, the movement of the hydraulic power piston, which regulates the fuel-control setting, also acts to change the tension of the speeder spring. Thus, an increase in fuel corresponding to an increase in load is produced by an upward motion of the power piston. This motion raises the right end of a lever fastened to the end of a shaft and thus turns the shaft, which operates the fuel-control mechanism. At the same time, a pin in the fuel-control lever located in a slot of the forked lever slightly lifts the upper seat of the spring and thus decreases the spring force. This lowered spring force will require less centrifugal force, and consequently, a lower control speed is necessary for balance. By adjusting the leverage of the linkage between the fuel-control and the speeder spring, depending upon the characteristics of the engine, the amount of speed droop necessary to obtain the compensation required will be provided and stable operation will result.

Isochronous governor compensation. In order to obtain compensation of hydraulic governors and yet maintain truly isochronous governing, another method must be used. One mechanism which provides compensation without speed droop is illustrated in Fig. 13–6. The pilot-valve plunger operates in a movable pilot-valve bushing in which are located the ports controlling the oil flow. The movement of this valve bushing during a speed change is controlled by the *receiving compensating* piston to which it is attached. It should be noted that under constant-speed operation, the compensating spring will hold the pilot-valve bushing in its central position. Thus, until there is an actual change in the fuel-control mechanism, the action of the pilot valve is the same as in the simple hydraulic governor shown in Fig. 13–4. The compensating action of the valve bushing is controlled hydraulically by transfer and by leakage of oil pressure between the compensating *receiving* piston and the

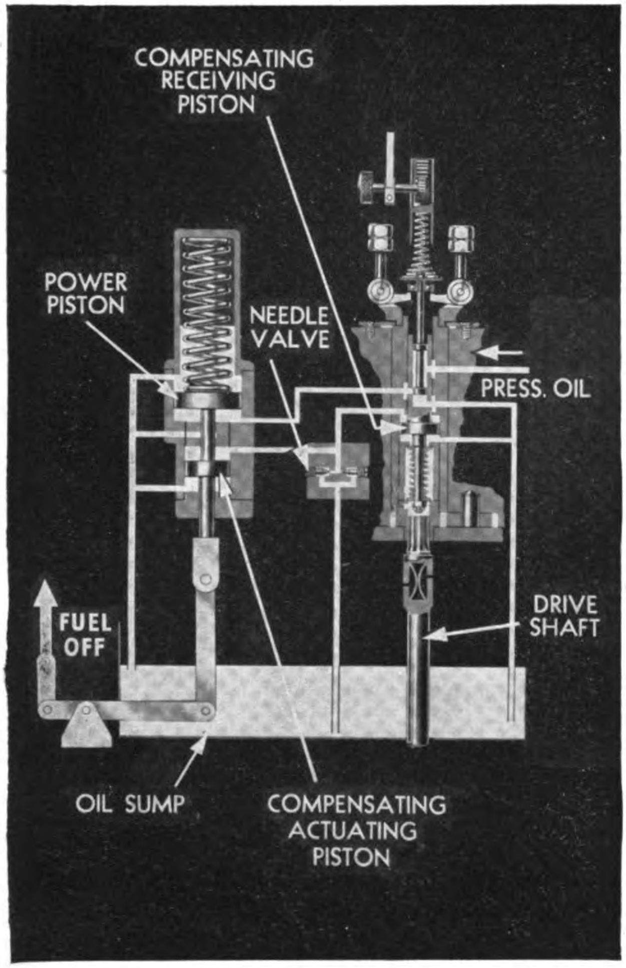


Figure 13–6. Hydraulic governor with speed droop.

compensating *actuating* piston. The rate of compensation is adjusted to fit the engine characteristics by regulating the oil leakage through the compensation needle valves.

The operation of this hydraulic compensating device during the change in load may be explained as follows: when the load on the engine increases, the speed will start to decrease, causing the speeder spring to overbalance the reduced centrifugal force of the flyballs. This action moves the pilot-valve plunger down, as shown in Fig. 13–7, so that the control port in the bushing, which is normally covered by the lower land on the pilot-valve plunger, is uncovered and put in connection with the supply of oil under pressure. The oil under pressure then flows through the open control port to the bottom of the power-piston cylinder, begins to lift the power piston, and to move the fuel control toward more fuel in the same manner as in an elementary hydraulic governor. In this governor, however, the actuating piston is connected to the power piston and thus moves up with it. As the actuating piston moves up, it forces the oil above it through passages, partly to the compensation needle valves and partly to the top of the receiving piston. During a rapid movement of the power piston, however, very little oil can leak past the needle valves. When the power piston first moves up, the oil from the actuating piston will force the receiving piston down against the action of the compensating spring. The pilot-valve bushing will thus move down as the fuel-control mechanism is moved toward less fuel. When the pilot-valve bushing moves down, the control port will come opposite, and again be covered by the lower land on the pilot-valve plunger, which previously had moved down under the action of the speeder spring. This will cut off the oil pressure to the power piston and prevent further increase in the fuel supply.

When the fuel supply is increased, however, the speed of the engine will start to pick up, and as the engine speed gradually returns to normal, the centrifugal force of the flyweights will move the pilot-valve plunger back to its normal central position. By proper adjustment of the compensation needle valves, the oil above the receiving piston can be made to leak out at a certain rate. This will allow the compensating spring to act to gradually move the receiving piston up as the oil is forced back to the needle valves. Thus, the pilot-valve bushing may be returned to its normal central position at the same rate as the pilot-valve plunger, so that the central port is kept covered and no further regulation of the fuel control takes place.

When there is a decrease in load and the engine starts to speed up, the same sequence of operations takes place in the opposite direction. In this case, the increased centrifugal force due to the increase in engine speed moves the pilot-valve plunger up. This uncovers the control port and allows the oil trapped in the bottom of the power-piston cylinder to drain back to the oil sump. The power piston thus moves down under the action of the power spring and moves the fuel control in the direction toward less fuel. As the power piston moves down to decrease the fuel, the actuating piston is also moved down with it. This movement draws oil partly through the compensation needle valves and partly from above the receiving piston. If this action occurs rapidly, very little oil is drawn through the needle valves, and thus, the receiving piston is moved up against the action of the compensating spring. Since the pilot-valve bushing also moves up with the receiving piston, the control port will be closed when it comes opposite the land on the pilot-valve plunger. This will cut off the oil line from the bottom of the power-piston cylinder, thus trapping the oil and preventing further decrease in the fuel supply. When the fuel supply is decreased, however, the speed of the engine will start to slow down, and as the speed gradually returns to normal, the flyweights move in and the pilot-valve plunger will return to its normal control position. During the speed correction, the compensating spring begins to move the pilot-valve bushing back to its central position also. As the speed decreases, oil will be drawn up from the oil sump to permit the receiving piston to return to its normal central position at the same rate as the pilot-valve plunger. By proper needle-valve adjustment, the pilot-valve plunger may be returned to its normal central position, so that after the initial fuel change, the control port is closed and no further fuel regulation takes place.

If the power piston is moved the first time to give exactly the fuel setting required to balance the change in load before the control port is closed by the compensating system, and the port remains closed while the engine returns to control speed, there is no hunting at all. This condition is known as *dead-beat governoring* and is difficult to obtain in practice. However, if the governor compensation is adjusted correctly, only a slight amount of hunting of small magnitude will occur following a load change which will quickly be damped out, resulting in stable operation throughout the operating range.

Load distribution. An engine equipped with an isochronous governor can carry any load between no load and the maximum load or overload that the governor will permit. If two or more engines are coupled to a single load they cannot all be equipped with truly isochronous governors, unless the engines are connected to electric generators having special characteristics which allow for operation in parallel with isochronous governors. Since isochronous governors permit any fuel setting within the capacity of the system, providing the speed remains constant, they are incapable of distributing the load between two engines in a predictable manner. Engines which are to be operated in parallel must use governors with a speed droop. Not more than one engine in such a system may be governed truly isochronously. A simple method of introducing speed droop is to use a suitable linkage between the fuel-control mechanism and the speeder-spring compression regulator, such as shown in Fig. 13–5. A similar mechanism may be attached to a compensating isochronous hydraulic governor to obtain speed droop. Adjustment of the lever linkage may be made to obtain the desired speed droop required to distribute the load adequately.

13–6. Load-limit governors. At any particular engine speed, there is a rather definite maximum sustained load which an engine can carry without damage. An ordinary governor, whether isochronous or built with speed droop, is sensitive only to speed. If the engine slows down, the governor will increase the fuel supply even though this may overload the engine. To prevent the governor from increasing the fuel supply beyond that required for a safe load, the governor may be equipped with a maximum-fuel stop. If a fixed stop were used, it would permit the engine to be overloaded at low speeds if it were set to allow the maximum safe load at full speed. To give full protection throughout the full engine-speed range, and at the same time to allow the engine to develop its maximum permissible power, the governor may be equipped with a variable maximum-fuel stop which will permit the delivery of the maximum safe fuel supply at any engine speed. Such a device is termed a *bmep. limiter* or *torque limiter.*

The maximum fuel delivery permitted by a torque limiter may be set slightly below the smoke limit for the engine, so that the engine will automatically give practically smokeless operation.

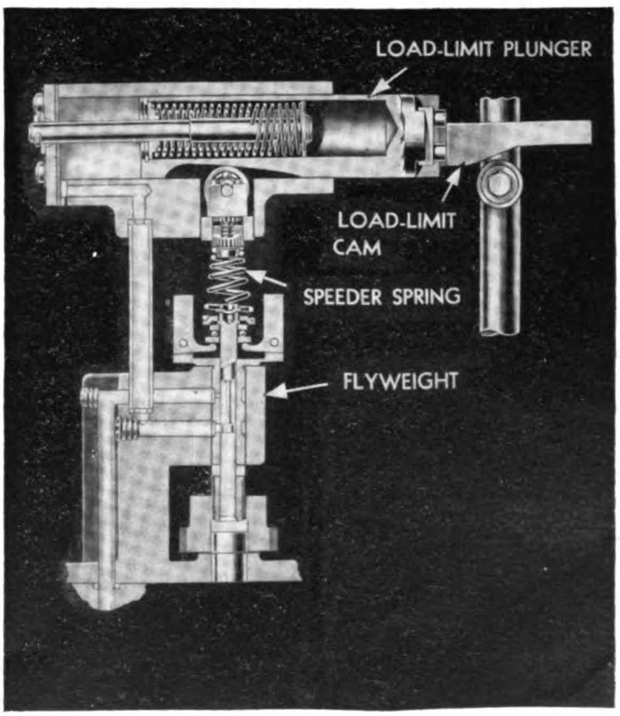


Figure 13–7. Load-limit hydraulic governor.

If a very heavy load is put on an engine so that a maximum-fuel stop comes into operation, then the engine speed will drop below the governor control speed, and the engine will operate with a fixed rack setting as if it had no governor. Most types of loads demand less driving torque as speed decreases, and in most cases the engine will be able to carry the load without stalling at some reduced speed. The engine will develop a somewhat reduced torque and a greatly reduced power as its speed decreases, but it will probably still be able to drive the ship or run an overloaded generator at some below-normal speed.

A typical load-limit governor is illustrated in Fig. 13–7. It consists of a spring-loaded centrifugal governor which operates a hydraulic pilot valve for controlling the load-limiting mechanism. This mechanism consists of a spring-loaded plunger or piston to whose right end is attached a stop or cam which limits the movement of the main fuel controls. It will be noted that there is also a sloping cam surface cut into the lower side of the load-limit piston on which rides the roller attached to the top of the speeder spring. The position of the load-limit piston thus controls the compression on the speeder spring. In operation, when the speed increases, the centrifugal force of the flyweights moves the pilot-valve plunger up and uncovers the control port, permitting oil trapped in the load-limit cylinder to drain out and allowing the spring-loaded piston to move to the left. This moves the load-limit stop to the position permitting maximum opening of the fuel controls. When the engine slows down, the pilot-valve plunger will move down and open the control port so that it connects with the supply of oil under pressure. The oil pressure acting on the load-limit piston will move it to the right until the control port is again covered. As the load-limit piston moves to the right, the roller riding on the piston cam surface will move up to reduce the speeder-spring compression, so that the centrifugal force of the flyweights will be balanced at a lower speed. Thus, for every engine speed there is a corresponding position of the power piston which will provide the necessary spring compression to balance the centrifugal force of the flyweights and maintain the pilot-valve plunger in its control position covering the control port.

At very low speeds, the load-limit piston must be moved all the way to the right before the speeder-spring compression can be reduced sufficiently to balance the low centrifugal force of the flyweights. As a result, the load-limit cam will stop the movement of the fuel control at its lowest limit, by means of the steep slope on the end of the cam. This will prevent overloading of the engine at low speeds where the engine is more likely to smoke. By properly designing the slope of the load-limit cam, the load for each position corresponding to a certain engine speed thus will be limited to prevent overload.

Another type of load-limit governor is sometimes used to prevent too much fuel injection during engine starting. This consists of a fuel-control stop which is operated by compressed air, and functions only when the air-starting system is operating. In order to permit the hydraulic governor to function immediately upon starting, a booster servomotor, i.e., an auxiliary power supply, is employed to furnish oil pressure to operate the controls until the regular oil pump supplying the governor can build up sufficient pressure. The booster servomotor is operated by compressed air from the air-starting system during the engine-starting period.

13–7. Overspeed governors and trips. Overspeed governors are employed as safety devices to protect engines from damage due to overspeeding from any cause. When an engine is equipped with a regular speed governor of any type, the overspeed governor will function only in the event of failure of operation of the regular governor. If the engine speed is manually controlled, the overspeed governor will function in case the speed increases beyond a safe limit before the operator can control it.

Since overspeed governors are essentially emergency controls, they must operate either to stop combustion or to limit the combustion pressures in the engine cylinder in order to slow it down. This control of combustion may be obtained by either regulation of the fuel or of the air supply. Most overspeed governors function to cut off or limit the fuel supply to the engine cylinders. In some two-stroke engines, however, it is possible for an engine to run away by burning lubricating oil which may happen to be taken in with the fresh air. Where this may occur, the governor is arranged to cut off the air supply to the cylinder and thus to stop the engine. Overspeed governors which bring an engine to a full stop by cutting off all the fuel or air supply are commonly termed *overspeed trips*. If the overspeed control merely slows the engine down, but allows it to continue to run at safe operating speeds, it is better termed an *overspeed governor*. In Navy service, particularly for main propulsion diesel engines, overspeed governors are preferred to overspeed trips since the latter type may leave a ship without power at a critical moment.

Overspeed governors and trips of all types depend upon a spring-loaded centrifugal-governor element for their action. The spring in this case is preloaded to a force which will overbalance the centrifugal force of the flyweights until the engine speed rises above the desired maximum. When this speed is reached, the centrifugal force overcomes the spring force and puts to action the controls which cut off or limit the air or fuel supply.

The actual operation of the fuel or air controls may be accomplished directly by the centrifugal force of the overspeed governor, as in a mechanical governor, or it may be supplied by oil pressure, as in a hydraulic governor. In an overspeed trip, the shut-off control may be operated by the force of a power spring which is put under tension when the trip is manually reset and is held there by means of a latch. When the maximum speed limit is exceeded, a spring-loaded centrifugal flyweight will move out and trip the latch, allowing the power spring to operate the shut-off control.

13–8. Limitations of performance. The power which can be developed by a given-sized cylinder and piston stroke is limited by: (1) engine speed, and (2) mean effective pressure developed in the cylinder.

Engine speed is limited by the inertia forces produced by the moving reciprocating parts which increase very rapidly with an increase of rpm., and also by lubrication difficulties caused by heat which is developed by friction.

Mean effective pressure is limited by:

1. Heat losses and efficiency of combustion.
2. Volumetric efficiency in four-stroke, and scavenge efficiency in two-stroke engines, as discussed in Sec. 6–2. The amount of air charged into the cylinder and available for combustion depends upon these efficiencies. Supercharging increases the amount of air and thus helps to increase mean effective pressure.
3. Completeness of the mixing of the fuel and air. This depends upon the fineness of the fuel atomization, sufficient penetration, and good distribution of the fuel.

For each engine, the limiting or maximum permissible brake mean effective pressure is prescribed by the manufacturer and must never be exceeded. In a direct-drive ship, the mean effective pressure developed is determined by the rpm., and increases very rapidly even with a small increase of rpm. over the maximum rated value. In an electric-drive ship, as long as the rating of the generator, to which the engine is connected, is not exceeded, the mean effective pressure will be within the safe limit.

Overloading a diesel engine means exceeding the limiting mean effective pressure and ***should never be done.***

13–9. Problems. 1. Determine the speed droop, referred to full-load speed, of an engine which has a normal speed of 2,000 rpm. and an idling speed of 2,670 rpm. *Ans.* 2.69 per cent.

2. Determine the speed droop of the engine in problem 1, referred to the average speed. *Ans.* 2.66 per cent.

3. Determine the sensitivity of the governor which permits the engine speed, at rated load, to stay between 2,109 and 2,093 rpm. *Ans.* 0.76 per cent.

4. Determine the sensitivity of a governor which at rated load maintains the engine speed between 1,807 and 1,793 rpm. and express it as fluctuation above and below the rated speed. *Ans.* ±0.39 per cent.

13–10. Questions. 1. What method is used to control engine speed when the load changes?

2. Enumerate the types of governors in use with diesel engines.

3. Enumerate the main characteristics of a diesel-engine governor.

4. What is the *speed droop* of a governor?

5. What types of governors are used in Navy diesel engines?

6. What means are used to obtain variable-speed governing?

7. What is *hunting* of a governor?

8. What is the main feature of any hydraulic governor?

9. What is the purpose of a load-limiting governor?

CHAPTER 14

STARTING AND REVERSING

14–1. Requirements. *Methods.* In order to start a diesel engine, the engine crankshaft must be turned over by some outside means so that the air in the cylinder at the top center is compressed to such a temperature and pressure that fuel injected by the injection system will ignite and produce a power stroke.

The first requirement for starting a diesel engine is to turn it over with *sufficient speed.* If the engine is turned over very slowly, the unavoidable small leaks past the piston rings and also through the intake and exhaust valves will permit part of the air to escape during the compression stroke. This may lower the compression pressure and temperature at the end of the stroke below the values necessary to ignite the injected fuel. A heat loss from the compressed air to the metal walls of the compression space is also greater at low speeds when the duration of the compression stroke is longer, and this further lowers the temperature of the air. Therefore, there is a minimum speed which the engine must attain before ignition will occur and the engine will start firing. The starting speed depends upon the type and size of the engine, its condition, and the temperature of the surrounding air. In some engines the starting speed is around 70–75 rpm., while in some small engines it may be as high as 250 or even 300 rpm. There is no definite relation between the starting and operating speed of an engine. However, all other conditions being equal, an engine starts at a lower speed when it is in its best operating condition, has well-seated piston rings and valves, correct timing, and when there is no excessive friction in the engine or in the engine-driven auxiliaries.

The second important requirement is a *correct compression ratio.* If the compression ratio is not sufficiently high, the final temperature and pressure of the compressed air charge again will not be that required for ignition. A new engine, naturally, has the correct compression ratio. However, wear of bearings may lower the piston position and with it the compression ratio. A late closing of the intake valves, caused by incorrect take-up of wear in the valve mechanism or some other error in the valve timing, also may decrease the effective compression ratio.

Starting methods. Modern Navy diesel engines use only two methods for starting: (1) electric starting for small engines and (2) compressed-air starting for medium-sized and large engines.

However, in some installations, large engines are also started electrically.

14–2. Electric starting. The arrangement normally used is of the same general design as an automobile-engine starter but is more powerful, both because Navy diesel engines are larger than automobile engines, and because a compression-ignition engine requires a larger amount of power to turn it over than a spark-ignition engine. The power required to first turn the crankshaft of a cold diesel engine and then bring it up to starting speed, generally is a little less than 10 per cent of its rated output, but in some cases may amount to as much as 20 per cent, especially with small engines. With large engines it is less, down to about three or four per cent of the rated output.

Electric-starting systems use direct current because the electrical energy in this form can be stored in batteries and drawn upon when needed for starting, after which it can be replenished by recharging with an engine-driven generator.

The electric-starting system consists of:

1. Storage battery.
2. Direct-current or D.C. electric motor.
3. Mechanical engagement between the motor and engine crankshaft.
4. An auxiliary electric generator to charge the battery, if the engine does not drive a D.C. generator as its main load.

5. Necessary cables, wires and switches to complete the electrical system.

Storage battery. Storage batteries usually have an electromotive force of 24, sometimes of 32 volts. Some small engines use 12-and 16-volt batteries. The capacity varies from about 175 to 225 amp.-hours.

Electric motors. The motors are usually of the series-wound, heavy-duty type, which, for a short time, a few seconds, can carry a 100 per cent over-load.

To prevent overheating of the starter motor due to the heavy current drawn, some electric-starting systems employ a timing device in conjunction with the magnetic switch, designed to disconnect it after about 15 seconds if the engine does not start. It should be noted that electric starters are designed to produce maximum power for short periods only, and should not be operated for more than 30 seconds at a time.

Mechanical engagement. This engagement consists of a spirally splined portion of the motor-armature shaft, a shift sleeve, a pinion with a splined bore which enables it to move along the shaft, and a spring. When the engine must be cranked, the pinion is moved to the outer end of the splined shaft either manually or by magnetic action, and its teeth mesh with teeth of a gear which is part of the flywheel. When the motor begins to turn, it turns the flywheel, and the spline is cut so that when the torque is transmitted from the motor to the flywheel, the axial component due to the spiral shape of the spline presses the pinion against a collar on the end of the shaft. When the engine begins to fire, the circumferential velocity of the pinion and the torque is reversed. This pushes the pinion along the spline away from the shaft collar and out of mesh with the gear.

Cables and switches. Although the voltage used for most starting motors is comparatively low, as stated above, the current drawn during starting may rise to values of 500 amperes or more, so that heavy cables and switches must be used. In order to keep the electrical losses as low as possible, the main cables are made as short as possible. The starting switches are usually of the magnetic type which can be remotely controlled by a push-button switch and require less current to operate.

14–3. Compressed-air starting. Starting a large diesel engine requires the expenditure of considerable energy in a relatively short time. One of the simplest methods for storing large quantities of potential energy is by compressing air into tanks. The compressed air may then be used as the source of starting energy by expanding it in the engine cylinders. This requires a starting-valve gear which adds but little extra weight or size to the engine, and is the method generally employed on large engines.

The compressed air used for starting can be readily replenished over a long period of time, after the engine has been started, by means of a small compressor which requires only a small amount of power. The compressor may be driven directly from the engine or from a separate power source, such as a small hand-started engine. A separately driven air compressor is always used with Navy diesel engines, even in addition to an engine-driven one, to insure a positive supply of starting energy.

With most air-starting systems, the compressed air is admitted to the top of the main engine cylinder under a pressure of from 100 to 400 psi., through special poppet starting valves mounted in the cylinder head. Some large engines use air pressures up to 600 psi. The starting valves are timed to open when the pistons are in the position corresponding to the start of the normal expansion stroke. The air pressure thus acts on the pistons which turn the engine as rapidly as necessary for starting. When the engine is turning fast enough, the fuel begins to ignite as it is injected and the air supply is cut off.

In engines with 10 or less cylinders, usually all cylinders are equipped with air-starter valves. In 12- and 16-cylinder engines, only one-half of the cylinders have air-starter valves. In double-acting engines, an

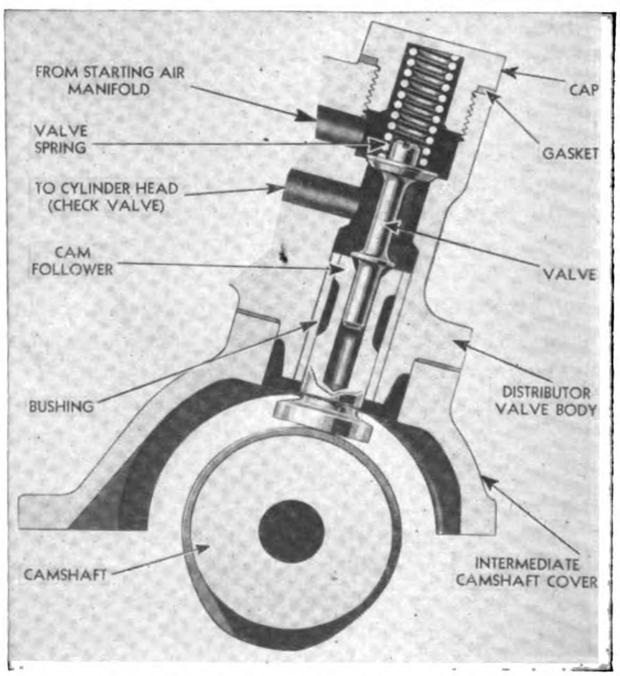


Figure 14–1. Air starter distributor valve.

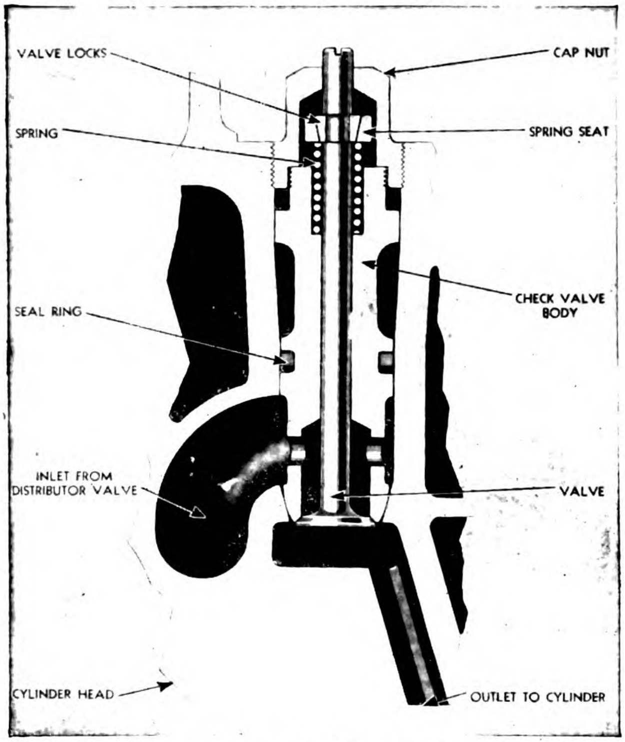


Figure 14–2. Air starter check valve.

air-starting valve is located in each cylinder, but only in one of the cylinder heads, usually in the upper head. All multicylinder naval engines can be started by the air pressure at any position of the crankshaft without barring the engine over. Two- and three-cylinder engines require barring over.

There are two methods used for controlling the timing of the air starting valves: (1) direct mechanical operation by means of individual cams and valve gear, and (2) indirect operation by means of an air distributor. Most naval engines use the indirect method, which itself has many variations.

Distributor starting-valve gear. Air starting distributors usually consist of cam-operated valves which are timed to admit the compressed air to the cylinders through automatic air-operated poppet valves. Where the distributor valves are located near the cylinders, the main compressed-air supply is connected to the distributor which delivers it through the air passages to each starting valve. The starting valves in this case consist merely of spring-closed check valves which are opened directly by the air pressure delivered to them in proper sequence by the distributor. Fig. 14–1 shows a cam-operated distributor valve and Fig. 14–2 the air-starter check valve from the same engine.

Where the distributor is located some distance from the cylinder head, the starting valves are generally *pilot-operated* by the air pressure delivered from the distributor. An example of this type of valve is shown in Fig. 14–3. In this case, the distributor is connected by small air-lines to an air chamber above each starting valve, containing an operating piston connected to the top of the valve stem. The air pressure delivered from the distributor at the proper time acts on the top of this piston, which has sufficient area to open the valve against the force of its closing spring. The main compressed-air supply is delivered to the valves through the air-header connection to a passage below the balance piston. The pressure of the main compressed-air supply, acting on the inner side of the valve head; will not open the valve because it is a balanced valve. This means that the air pressure also acts on the bottom of the valve-stem plunger, which has the same area and thus balances the forces. The main air supply, therefore, will be admitted to the cylinders only when the distributor delivers a pilot charge of air to actuate the piston on top of the valve stem. In this way, only a small amount of air is used to control the valves, and very little pressure is lost from the compressed-air supply.

The tappet of the distributor valve in Fig. 14–1 rides constantly on the cam. In most engines there are provisions to bring the tappet in contact with the cam only during the starting period. This is usually done by compressed air. When the engine is running, a spring holds the air-distributor valve and tappet out of contact with the cam, Fig. 14–4a. But when air is admitted through a hand-operated air-start control valve to the inside of the plug, air pressure overcomes the spring force and pushes the tappet down into contact with the cam, Fig. 14–4b. The cam in this case is a round disk with a flat recess; when the tappet goes into the recess, as shown in Fig. 14–4b, the piston valve above it will admit air to the check valve in the cylinder. When the tappet is pushed up by the

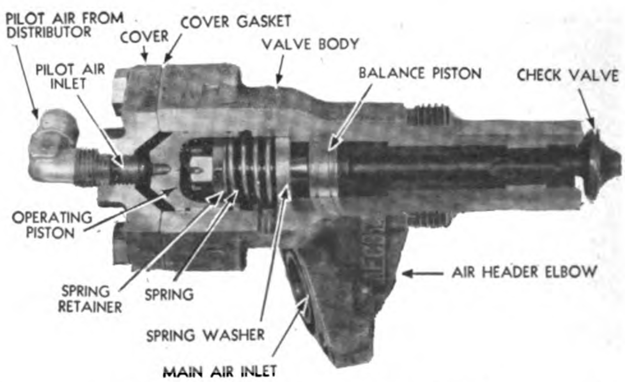


Figure 14–3. Air operated starter valve.

round part of the disk cam, the air to the check valve is cut off. When the engine starts firing, the air-start control valve is closed, the air pressure is relieved, and the spring pushes the tappet away from the cam as shown in Fig. 14–4a.

14–4. Cold-weather starting. Diesel engines are built, generally speaking, to start and to operate at an average atmospheric temperature around 70° F. When a diesel engine must be started after it has been secured for several hours at a temperature considerably lower than 70° F, one or both of the following may be encountered:

1. The starting device may not turn the engine fast enough to produce the necessary compression temperature.
2. Even if the engine is being turned over fast enough, the compression temperature may be below the required minimum for igniting the fuel.

Slow turning results from an increase of the viscosity of the lubricating oil. Since the frictional resistance to turning over of an engine is directly proportional to the lubricating oil viscosity, it can be decreased only by lowering the viscosity of the oil. The greatest friction is in the oil film between the piston and cylinder walls and the most effective remedy for overcoming it, is to heat the cylinder walls by increasing the temperature of the water in the jacket. Some small engines intended for cold-climate operation, have an electric heater built into the cylinder jacket. Another procedure is to admit low-viscosity oil to the lubricating-oil pressure pump just before securing the engine, in order to obtain a low-viscosity oil-film between the pistons and cylinder walls. Sometimes even the lightest diesel lubricating oil is too heavy and it may be necessary to preheat it, or if no facilities for preheating are available, to dilute it with kerosene, up to about 10 per cent.

Low air temperature. Assuming that other factors remain constant, the final temperature of compression is directly proportional to the initial temperature of the air charge.

Example 14–1. If the initial air temperature is 70° F, the final compression temperature in a cold, small diesel engine, with a compression ratio 14:1, when being turned over at 200 rpm., will be about 700° F. Find the final compression temperature with the same cranking speed if the initial air temperature drops to 40° F.

As stated in Sec. 2–4, such calculations must be made with all temperatures expressed in absolute degrees or degrees Rankine.

The normal room temperature will be $70+460=530°$ R, and the compression temperature will be $700+400=1,160°$ R, or an increase of $1,160 \div 530 = 2.19$ times.

Therefore, with the lower intake temperature of $40+460=500°$ R, the compression temperature will be $500 \times 2.19 = 1,095°$ R or $1,095-460=635°$ F. Thus a drop of the intake temperature of 30° F lowers the final compression temperature by $700-635=65°$ F, and this may easily prevent the ignition of the injected fuel.

Example 14–1 shows that the influence of a drop of intake temperature is comparatively large.

Methods and devices used in diesel engines to overcome low intake-temperature influence in diesel engines may be enumerated as follows:

1. Electric intake-air heaters.
2. Electric flame primers.
3. Glow plugs.
4. Punks.
5. Fuel-pump overload devices.
6. Injection of ether or gasoline into air intake.
7. Increase of the compression ratio.

All these devices are used only in smaller engines cranked by an electric motor and batteries. Large engines are kept in engine rooms, where a temperature around 70° F is maintained, and no special devices are needed for cold-weather starting.

Electric heaters for intake air consist of coils of resistance wire installed in the intake manifold and heated by current drawn from the starting storage battery. This is a good method and is used rather extensively.

Flame primers consist of a hand-operated fuel-oil pump, a fuel spray nozzle, and an electric spark plug with a continuous vibrating spark coil. The nozzle and spark plug are installed in the engine air-intake manifold, thus giving a continuous series of sparks. The finely atomized fuel spray is directed against the

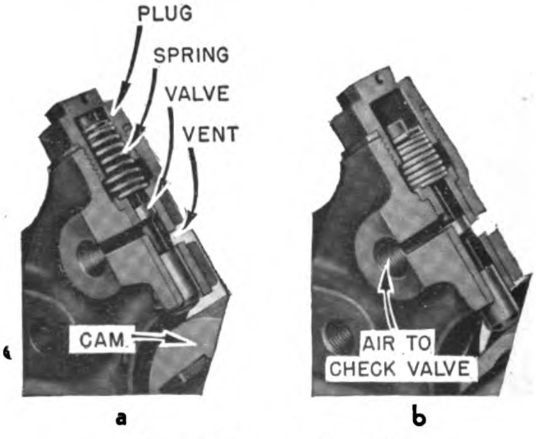


Figure 14–4. Pilot air valve.

spark plug electrodes, is ignited, and thus heats the air drawn into the engine during the starting. This method is quite satisfactory and is rather extensively used in several Navy engines.

Glow plugs are located in the cylinder heads of the engines so that the incoming air passes by them. They are heated by the current from the storage battery and their action is quite satisfactory. Unfortunately, being in the combustion space during the time that the engine is in operation, their thin wire soon deteriorates and therefore they are seldom used in naval engines. In some engines they are used simultaneously with an electrical resistance-wire air heater.

Punks. Some engines have a steel rod which reaches into the combustion space opposite the fuel injection spray. This rod has an axial hole drilled into its inner end and a screw thread on the outer end to hold it in place. Before starting the engine, the rod is screwed out, a piece of punk is inserted into its cavity, the punk is ignited, and then both are placed back into the cylinder head. When the engine is turned over, the fuel spray strikes the glowing punk and ignition of the fuel occurs. This simple arrangement is not generally used because only in a few engines can the punk be so placed that the fuel spray can reach it.

Fuel-pump overload devices. Regardless of the method by which a diesel oil is produced, it is always a mixture of different hydrocarbons, some being more volatile and some less. By increasing the amount of fuel injected per stroke, the air-fuel ratio is decreased, resulting in an increase in the relative amount of the more volatile, more easily ignited fuel particles. This method requires changes in the engine construction and is used only with a few engines.

Ether or gasoline for priming is used if the engine does not have an adequate cold-weather starting device, or in conjunction with an air heater in case of extremely cold weather. A very small amount of this highly volatile fuel, poured by hand into the air-intake opening, will be evaporated by the time it reaches the cylinder and will assist the ignition of the regular fuel injected into the cylinders. This method is effective but dangerous, as even a small amount of ether or gasoline is apt to produce excessive combustion pressures, overstrain the cylinder-head studs, and in extreme cases, may wreck the engine.

Increased compression ratio. As mentioned before, the higher the compression ratio, the higher will be the final compression temperature. In some engines the combustion chamber is divided into two spaces with a connection between which can be closed by a valve or cock. When the engine is running, this valve is open and the compression ratio is between 11.5:1 and 13:1. Before starting, the valve is closed and the outer space is cut off, increasing the compression ratio about four points, or to between 15.5:1 and 17:1. Such an increase of the compression ratio will give a final compression temperature increase of about 150° F, which in most cases is sufficient to insure ignition without failure.

14–5. Reversing. Small diesel engines for boat propulsion are usually connected to the propeller shaft by means of reverse gears discussed in Sec. 15–6. Large diesel engines for ship propulsion, when driving the propeller mechanically, are usually direct-connected to it, and therefore, must be provided with some system of direct-reversing for astern operation. For safe maneuvering, the reversing mechanism must be capable of quickly slowing down the engine to a complete stop and then starting it up in the opposite direction. Furthermore, this must be accomplished against the action of the propeller, which tends to turn it in the original direction by the motion of the ship through the water.

The only satisfactory method for reversing direct-connected diesel propulsion engines is by compressed air in conjunction with the air-starting system. This is accomplished by changing the timing of the air-starting valves so that the compressed air will be admitted to the cylinders to oppose the original direction of rotation. At the same time, the timing of the fuel injection and the intake and exhaust valves is changed to correspond with the new direction of rotation. Thus, as soon as the crankshaft rotation is reversed, the engine will start and operate in the opposite direction.

When the direction of rotation of the crankshaft is reversed, all engine-driven pumps must also be operated in the opposite direction. In the case of recipro-

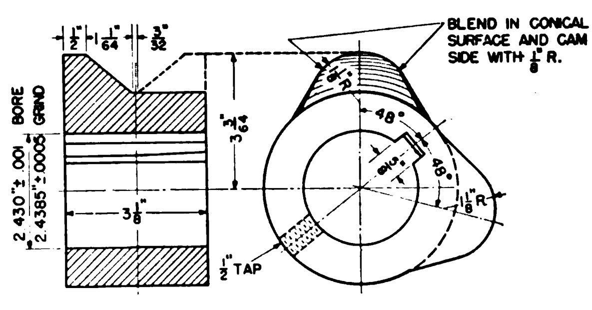


Figure 14–5.
Intake or exhaust cam of a direct-reversible engine.

cating pumps, this does not make any difference. Geared pumps are provided with automatic reversing check valves, which, by the pressure of the fluid delivered by the pump, keep the discharge piping connected with that side of the pump, from which the fluid is being delivered. Positive displacement blowers have either a similar arrangement or change-over valves operated mechanically. Centrifugal pumps for these engines usually are built with straight vanes, as in Fig. 12–8, designed to deliver the same amount regardless of the direction of rotation.

Four-stroke engines. When a four-stroke cycle engine is reversed, the order of the operating strokes is completely changed. The reversing system, therefore, must change both the valve timing with respect to the crankshaft position, and the sequence of valve operation as well. This is achieved by using a separate set of cams to operate the valves in each direction of engine rotation. Ahead and astern cams for each valve are provided side by side on the camshaft, and one set of cams is brought into operation by the reversing gear, depending upon the direction of rotation desired. There are two general methods used to bring the second set of cams into position to operate the valve gear for reversing: (1) by sliding the camshaft endwise, or (2) by shifting the cam followers.

When a sliding camshaft is used, the cams are provided with beveled edges or ramps to permit the cam followers to slide over them when the camshaft moves endwise. An illustration of a cam used with this type of camshaft is shown in Fig. 14–5. The force to slide the camshaft must be sufficient to overcome the pressure of the valve springs which are compressed when the followers are lifted as the ramps of the cams slide under them. The force required for sliding the camshaft is usually provided by compressed air acting on a piston connected to the shifting lever.

In order to reduce the force necessary for sliding the camshaft, the cam followers on some engines are lifted clear of the cams while the camshaft slides endwise into place opposite the second set of cams. The cam followers are then returned to their operating position in contact with the second set of cams.

Instead of sliding the camshaft endwise, some engines are provided with cam followers which are shifted from contact with one set of cams to the other while the camshaft remains in place. One such arrangement employs double roller followers mounted opposite each set of cams on movable links in the valve gear as shown in Fig. 14–6. The valve gear is designed so that one set of rollers will be in contact with the corresponding set of cams for valve operation in one direction. To provide valve operation in the opposite direction, the reverse gear is arranged to swing the movable links in position so that the other set of rollers will contact the second set of cams.

When the cams which control the timing of the air-starting valves and the fuel injector are all mounted on the main valve camshaft, then the entire reversing process can be accomplished by providing a second complete set of cams for them to use with one of the previous systems outlined. If, however, the air-starting and fuel-injection systems are operated separately, then other methods must be employed to reverse the timing.

One method used to change the timing of separately driven air-starting and fuel-injection systems is to provide symmetrical cams on the individual camshafts which operate these systems. By rotating these camshafts a few degrees with respect to the crankshaft which drives them, when the engine is reversed, the opposite side of the cam will operate the air starting and fuel injection in the correct timing for reverse engine rotation.

When these separate camshafts are driven by means of helical gears, a simple method of rotating them slightly with respect to the crankshaft is to slide axially one of the gears which drives them. The angle of the gear teeth and the distance the gear moves control the number of degrees that the separate camshafts are rotated. This method may be used where a sliding main-camshaft helical-drive gear, drives the separate camshaft gears. When the main camshaft and gear slides endwise for reversing, the separate camshaft gears are rotated to change the timing of the air-starting and fuel-injection system.

Two-stroke engines. While any of the previous methods for changing the valve timing of a four-stroke engine could be applied to the valve gear of a two-stroke engine, a simpler type of reverse gear can be employed. Since the port timing is fixed, the timing of the air-starting valves, the fuel-injection and

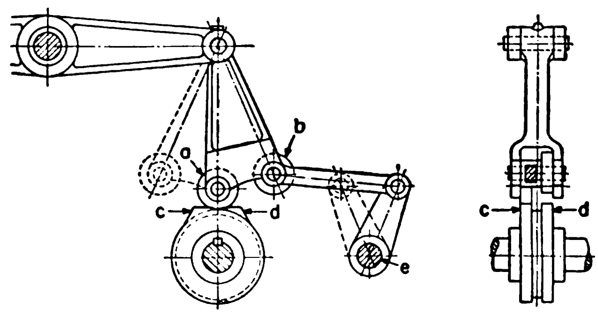


Figure 14–6. Roller-type reversing gear.

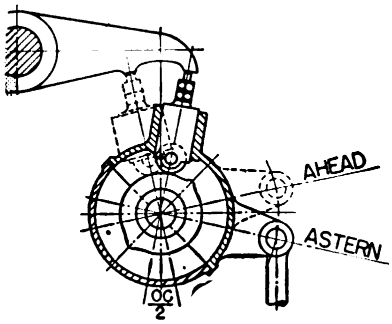


Figure 14–7. Reverse gear with movable roller.

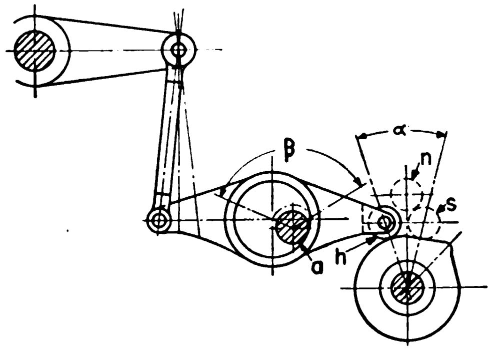


Figure 14–8. Reverse gear with eccentric motion.

scavenging valves, if used, may be changed merely by rotating the cam followers a few degrees with respect to the crankshaft, as shown in Fig. 14–7. The valve lift in the astern position is less than in the ahead position on account of the changed leverage, but that is unimportant. It is convenient to reduce the reversing angle by one-half, by using a half-speed camshaft and a double-nosed cam. Another gear of the same type is shown in Fig. 14–8. By turning the eccentric fulcrum *a* through an angle *B*, the roller *h* is displaced from the ahead position to the astern position *S*. In the neutral position *n*, the roller does not touch the cam.

Where the camshafts are gear-driven from the crankshaft, they may be rotated with respect to the drive gear by means of sliding helical gears or splines. When they are chain-driven, the timing of the camshafts with respect to the crankshaft is changed by movable idler sprockets which shorten the effective chain on one side while lengthening it on the other side of the drive sprocket.

In an opposed-piston engine, the crankshaft connected to the pistons which control the exhaust ports is set ahead of the other crankshaft connected to the pistons controlling the scavenge parts by several degrees of rotation when operating in the normal ahead direction. The crankshafts are geared together in this fixed relative position, and when the engine is reversed, the crankshaft controlling the exhaust ports will lag behind the crankshaft controlling the scavenging air ports. This condition is not desirable for best performance. However, it can be tolerated for less than full-power operation astern. The reverse gear in this case changes only the timing of the air-starting valves and of the fuel injection to provide astern operation.

14–6. Problems. 1. With the outside air temperature 65° F, the final compression temperature in a small engine that is cold, when being turned over, will be about 705° F. Find the final compression temperature if the outside temperature is 32° F. *Ans.* 632° F.

14–7. Questions. 1. What methods are used for starting Navy diesel engines?

2. What is the reason for using d.c. motors in Navy diesel electric-starting systems?

3. What Navy diesel engines are equipped with compressed-air starting?

4. What are the difficulties in starting a diesel engine exposed to a low temperature?

5. What methods are used to help start diesel engines under cold-weather conditions?

6. Why does a decrease of the combustion space help in cold-weather starting of a diesel engine?

7. What method is used to reverse the direction of rotation of a direct-reversible diesel engine?

CHAPTER 15

CLUTCHES AND GEARS

15–1. Definitions. A device to connect two shafts so that they will act as one is called a *coupling*. A device to connect or disconnect at will a part which transmits power, such as a gear, pulley, etc., to or from a rotating shaft is called a *clutch*. If two shafts must be connected or disconnected while in operation, the device to be used has the features both of a coupling and a clutch and is called a *clutch coupling*. However, in practice such a device usually is called simply a *clutch*, as for example, the connection between the engine shaft and the drive shaft in an automobile.

Gears. The word gear has many meanings, even in connection with machinery. One meaning, very often used, refers to a wheel with teeth meshing with a second similar wheel and used to transmit power. Another meaning refers to an auxiliary mechanism that performs a definite function in a complete machine, such as valve gear, discussed in Chapter 8. A third meaning refers to a combination of toothed wheels or gears used as a unit and transmitting a torque, usually with a change of speed or direction of rotation, such as a reduction gear or reverse gear. The present chapter deals with the latter type of gears.

Clutch couplings or simply *clutches* are used on direct-propulsion Navy diesel engines for the purpose of disconnecting the engine from the propeller shaft when needed. In small engines, clutches are usually combined with reverse gears used for maneuvering of the vessel. In large engines, special types of clutches are used to obtain special coupling or control characteristics and for the prevention of torsional vibration. The types of clutches used with Navy diesel engines are:

1. Friction clutches.
2. Pneumatic clutches.
3. Fluid or hydraulic clutches.
4. Electro-magnetic clutches.

15–2. Friction clutches. *Usage*. Friction clutches are commonly used with smaller, high-speed engines, up to 500 hp. However, certain friction clutches, in combination with a jaw-type clutch, are used with engines up to 1,400 hp., and pneumatic clutches, with a cylindrical friction surface, with engines up to 2,000 hp.

Action. In friction clutches in use with Navy diesel engines, the torque is transmitted from the driving shaft to the driven shaft through friction created by an axial force which produces contact between two or more surfaces. The friction surfaces may be either flat, Fig. 15–1a, or conical, Fig. 15–1b. The advantage of a conical surface is that a given axial effort produces a greater clutch-engaging pressure due to the wedging action of the cone angle *a*. This permits the use of a clutch with a relatively smaller diameter. The wedging action increases with a decrease of the angle *a*. However, a very small angle requires a large axial motion to engage and disengage the clutch, and considerable effort is necessary to disengage it when the clutch is running. Greater capacity in a given diameter of a plate clutch is obtained by the use of two or more parts of friction surfaces, as shown in Fig. 15–2. In this case, all plates are made so that they can slide axially, but one plate in each pair turns with one shaft and the other with the housing which is

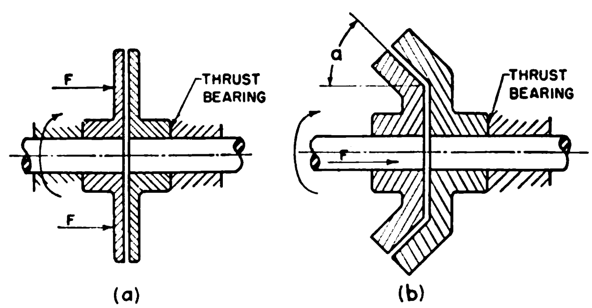


Figure 15–1. Types of friction clutches.

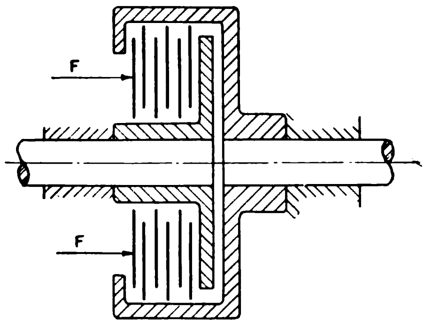


Figure 15–2. Multi-plate friction clutch.

secured to the other shaft. With the same axial engaging effort, the torque transmitted by a multi-plate clutch is directly proportional to the number of pairs of friction surfaces. Similarly, cone clutches are often made with two cones, as in Fig. 15–3, thus doubling their capacity. Finally, there are clutches in use with both cone- and plate-friction surfaces in the same housing.

Types. Friction clutches can be classified into *dry* and *wet* types, depending upon whether the friction surfaces operate dry or with a lubricant. The designs of both types are similar, except that the wet clutches require a larger friction area because of the reduced friction coefficient between lubricated surfaces. The advantages of wet clutches are smoother operation and smaller wear of the friction surfaces. Wear occurs due to slippage between the surfaces which takes place not only during engagement and disengagement, but to a certain degree during operation as well. Some wet-type clutches are filled with oil which is charged only periodically; in other clutches the oil is circulated continuously, being a part of the engine-lubricating system. Such a friction clutch must incorporate provisions which will prevent worn-off particles from being carried by the circulating lubricating oil to the bearings, gears, etc.

Friction surfaces. The friction-surfaces are usually made of different materials, one being of cast iron or steel; while the other is lined with some asbestos-base composition or sintered iron or bronze for dry clutches, and made of bronze, cast-iron, or steel for wet clutches. Cast-iron surfaces are to be preferred because of their better bearing qualities and greater resistance to scoring or scuffing. Sintered blocks are made of finely powdered iron or bronze particles molded in forms to the desired shape under high temperature and pressure.

Engaging. The application of force-producing friction is obtained either by mechanically jamming the friction surfaces together by some toggle-action linkage, or through stiff springs of the coil, leaf, or flat-disk type, as commonly used in automotive clutches. The use of springs has the advantage of providing an automatic wear-compensating feature, and reduces the frequency of necessary clutch adjustment. Friction clutches may be operated by *hand* or by *hydraulic*, *pneumatic*, or *vacuum* methods.

Hand-operated clutches usually have a long lever in order to reduce the necessary effort. Fig. 15–3 shows a hand-operated, double-cone, wet-type clutch. A description of this clutch will give an idea of the operation of a friction clutch in general. The clutch consists of a housing made in two sections, 2 and 7, forming an oil-tight covering for the two opposed cones, 3 and *4*, whose large ends face each other. The housing is secured to the jacking gear *10*, which is bolted to the crankshaft flange by fitted bolts *11*. The weight of both cones is carried on the after or motor shaft. These cones are mounted on hollow steel pins *19*, with a sliding fit. The pins *19* are pressed into the spider 5 and dowelled in place. The spider is secured to the motor shaft by two keys, a taper fit, and locknut *6*.

This clutch has no lining but steel-to-steel friction surfaces. When in the engaged positions, the cones are pushed apart by four heavy, curved, leaf springs *12*, which are connected to cone *4* by pins *8*, and to the sliding sleeve *1* through toggles *13* and rollers *16* with bolts and pins *17* and *18*.

The sleeve *1* is operated by a shifting collar *14*, made in half sections and lined with a white-metal bearing surface. Collar *14* is operated by two arms supported by a shaft, which rests in brackets secured to the engine room floor. To engage the clutch, sleeve *1* is pushed toward the clutch by collar *14;* the rollers *16*, at the end of the sleeve, deflect the inner end of the toggles *13*, and this in turn tends to shorten the springs *12*. This engaging force is at its maximum when the center of pin *18* is in line with the centers

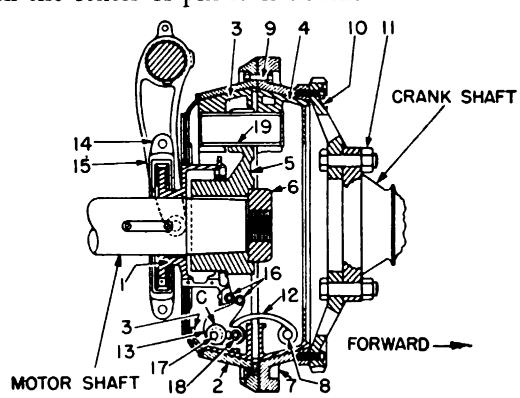


Figure 15–3. Double-cone friction clutch.

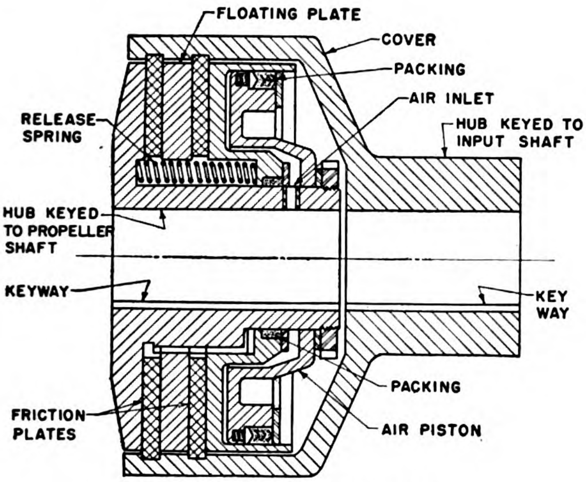


Figure 15–4. Pneumatic clutch.

of pins *17* and *8*. After the center of pin *18* has passed this line, the force that pushes the two cones apart and toward the housing will be slightly relieved but the springs will still press the cones against the housing. Only when an outside force is applied, which will move the center of pin *18* across the line between the pins *17* and *8*, will the two cones be drawn together by the springs *12*, acting the same as straight links. Adjustments for wear are made by turning eccentric bushings which support the pin *17;* this changes the distance between the centers of pins *17* and *8*. Some other friction clutches, used in combination with reverse gears, are shown in Figs. 15–10 and 15–11.

Hydraulic operation. The engaging force in hydraulically operated friction clutches is applied by a piston which travels in a cylinder pushed by oil under pressure. The source of the oil pressure may be either a special pump or the engine-lubricating oil system.

Pneumatic clutches. One type of pneumatic or air-operated clutch is shown in Fig. 15–4. The clutch is of the multi-plate design with four pairs of friction surfaces. The two plates splined to the cover, which is keyed to the engine or input shaft, are made of molded asbestos to obtain a higher coefficient of friction. The floating center plate and the end plate are of cast iron and are splined to the hub which is keyed to the propeller shaft. The air cylinder is part of the end plate and the piston is secured to the clutch hub. When the air pressure is released, four cylindrical springs push the end plate away from the friction plate. Air pressure, 80 to 90 psi., for operation of this clutch is supplied either by an engine-driven compressor or by a separate compressor, and is admitted from the hollow shaft with a hole drilled in it.

In this type of clutch, it is important to have a constant and reliable air pressure. A partial failure of the air supply may cause the clutch to slip and be overheated and damaged.

Another type of pneumatic clutch consists of a tire-like rubber gland, the outside of which is fastened to a drive flange and the inside is bonded to a friction facing, Fig. 15–5. The engine shaft carries a cylindrical drum, not shown on the drawing. When compressed air is admitted inside the rubber gland, the friction surface is pressed against the drum, thus engaging the clutch.

In clutches directly engaged by oil or air pressure as discussed above, no wear compensation is necessary and the engaging force is not changed by wear.

Vacuum-operated clutches are mechanical clutches having an external linkage operated by a vacuum piston and cylinder. The vacuum must be supplied by a separate vacuum pump.

The main advantages of hydraulic, pneumatic, or vacuum control of friction clutches are that they may be operated with a minimum of manual effort and that remote control is greatly simplified. While mechanically operated clutches may also be remotely

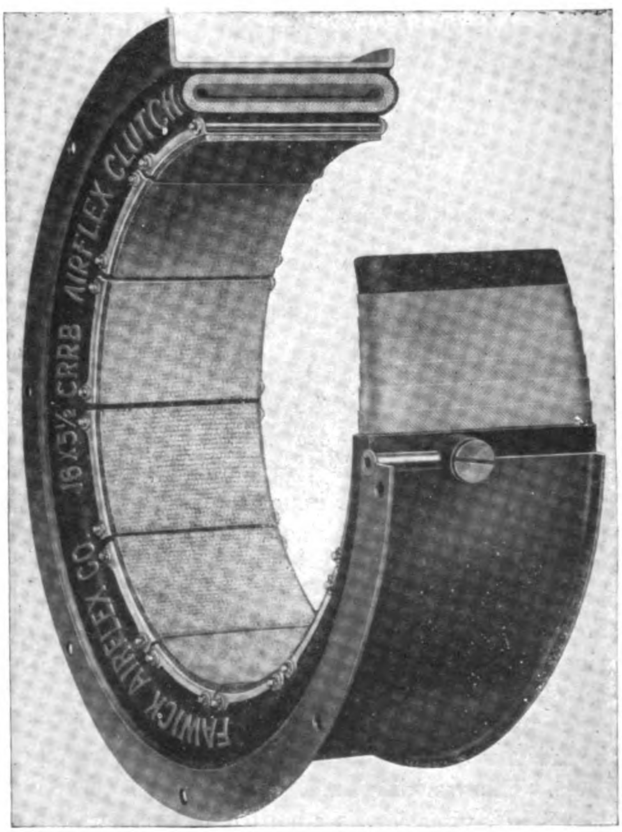


Figure 15–5. Airflex clutch.

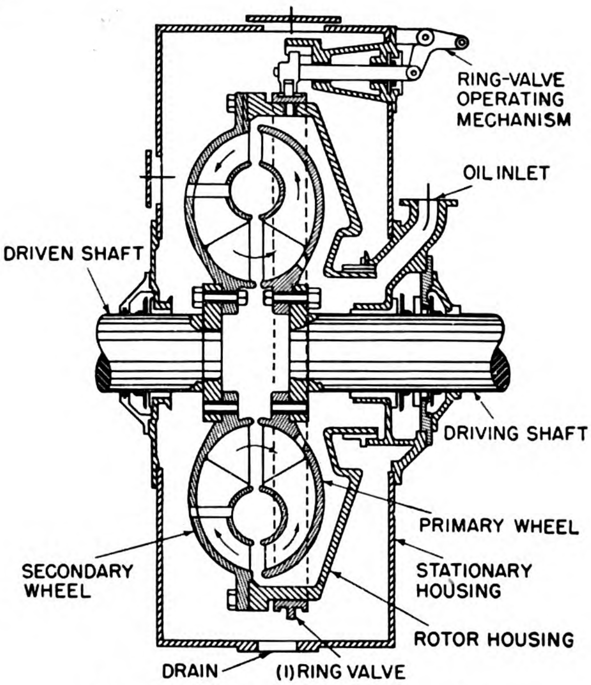


Figure 15–6. Hydraulic clutch-coupling.

controlled by means of suitable linkages, such control is more difficult because of an increase in shifting forces required due to the friction in the linkage, or because of the complication of the required external mechanical linkage.

15–3. Hydraulic clutches. The principle of operation of a hydraulic clutch or coupling may be demonstrated by two electric fans facing each other. If one fan is turned on, the other fan will begin to rotate from the energy contained in the air stream of the first fan and it will finally attain almost the same speed as the first fan.

A sectional view of a typical hydraulic clutch is shown in Fig. 15–6. The unit shown is supplied with lubricating oil from an external pump, and means are provided for a rapid removal of the oil when declutching. The impeller, or driving member is secured to the driving shaft. The runner, or driven member is secured to the driven shaft. A housing is bolted to the runner to enclose the back of the impeller and retain the working fluid. Annular core or guide rings of semicircular section are mounted in the impeller and runner to guide the working fluid. The impeller and runner are provided with a series of radial vanes, as shown in the upper view of Fig. 15–6. The working fluid, a mineral oil of 180 to 200 S.S.U. viscosity, is admitted to the impeller or primary wheel from an external pump. Rotation of the impeller causes the fluid to be thrown outward radially by centrifugal force to its outer rims from where it passes radially inward. The flow path of the fluid then resembles a coil spring bent into a circle. Transmission of power is effected by the release of kinetic energy in the impeller, acquired when the oil flows toward the rim. When the fluid flows radially inward between the vanes of the runner or secondary wheel, the kinetic energy is returned, transformed into work, and turns the runner and its shaft. Rapid declutching may be accomplished by opening the ring valve 1 which covers a series of ports in the runner cover. This permits the fluid contained in the clutch to be thrown out by centrifugal force.

Hydraulic couplings are similar in design to hydraulic clutches. The major difference is that the operating fluid is permanently contained in the unit and there is no provision for declutching while in operation.

Because of the losses due to mechanical and fluid-friction, the runner, or driven member, turns at a slightly lower speed than the impeller, or driving member. The difference in speed is directly proportional to the losses and is therefore a measure of the efficiency of power transmission. The efficiency of hydraulic clutches and couplings, under conditions of rated load and speed, is usually from 95 to 97 per cent.

The main advantages of hydraulic clutches or couplings are:

1. They can be used with engines of any power output.
2. Transmission of torsional vibration between engine and propeller shaft is prevented.
3. They protect the engine and reduction gear from sudden shock loads which may occur either as a result of piston seizure or fouling of the propeller.
4. The relatively large clearance between the rotating members simplifies alignment.

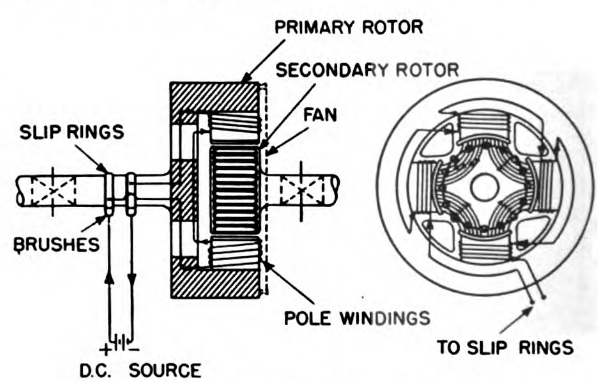


Figure 15–7. Principle of electric coupling.

15–4. Electro-magnetic clutches. The action of an electro-magnetic slip clutch, often referred to simply as either electric or magnetic clutch, is similar to the action of an electric induction motor. It is called a slip clutch because there is always some slipping between the driving and driven members. The speed of the driven shaft is slightly lower than that of the driving shaft. The principal parts consist of an outer member or field. Fig. 15–7, also called the *primary rotor*, surrounding an inner member, called the armature or *secondary rotor*. The magnetic flux between the two provides the force that rotates the secondary rotor. The field is rigidly mounted on the engine crankshaft while the armature is mounted on the driven shaft. An air gap of ample size separates the two parts of the coupling which fit together concentrically. The exciting direct current is introduced to the field through collector rings and brushes. This sets up a magnetic flux which holds the inner member, and with it, the driven shaft in a practically constant position to the primary rotor.

The advantages of a magnetic clutch are similar to those listed for the hydraulic clutch. In addition, by adjusting the strength of the exciting current, very fine speed control of the driven shaft may be obtained and the propeller shaft may be operated at very low speeds, as in maneuvering.

The efficiency of magnetic couplings, including the power required for excitation, ranges from 95 to 98 per cent. The excitation power is about one per cent of the transmitted power.

Remote control of magnetic clutches is particularly simple, regardless of distance, as only fine wires are needed for connections.

15–5. Reduction gears. For minimum weight and size for a given power output, diesel engines must have a relatively high rotative speed. On the other hand, for maximum efficiency, propellers must rotate at a relatively low speed, particularly where high thrust capacity is desired, as in a towing vessel. Reduction gears are used to correlate these two conflicting requirements and thereby, to obtain a low propeller-shaft speed with a high diesel engine speed. Ordinarily reduction gear ratios do not exceed 3:1, although reduction ratios up to 6:1 are sometimes used.

Reduction gears are classified according to the type and arrangement of the gears used. They may be:

1. External-gear type.
2. Internal-gear type.
3. Planetary-gear type.

The *external-gear* type unit, Fig. 15–8a, consists of a bull gear or driving gear, which mates with a pinion gear or pinion mounted on a parallel shaft. The term *pinion* is applied to the smaller of a pair of mating gears, the larger one being called simply the *gear*, regardless of which of the two is the driving and which the driven part. Either straight, helical, or herringbone gear teeth are used. Helical teeth give a smoother engagement, more teeth in engagement at once, and quieter operation, as compared with straight-cut gear teeth. A drawback is an axial thrust developed by the inclination of the teeth. With herringbone gears, which are the same as a pair of helical gears of opposite helix angles placed side by side, all the advantages of helical gearing are retained and the V-shape of the teeth eliminates axial movement and thrust.

In general the *speed ratio* is found by dividing the number of teeth in the *driven* gear *by* the number of teeth in the *driving* gear.

$$\text{Speed ratio} = \frac{\text{teeth number of driven}}{\text{teeth number of driver}} \quad (15\text{–}1)$$

If the number of teeth in the driving gear is smaller than in the driven gear, the quotient will be *greater*

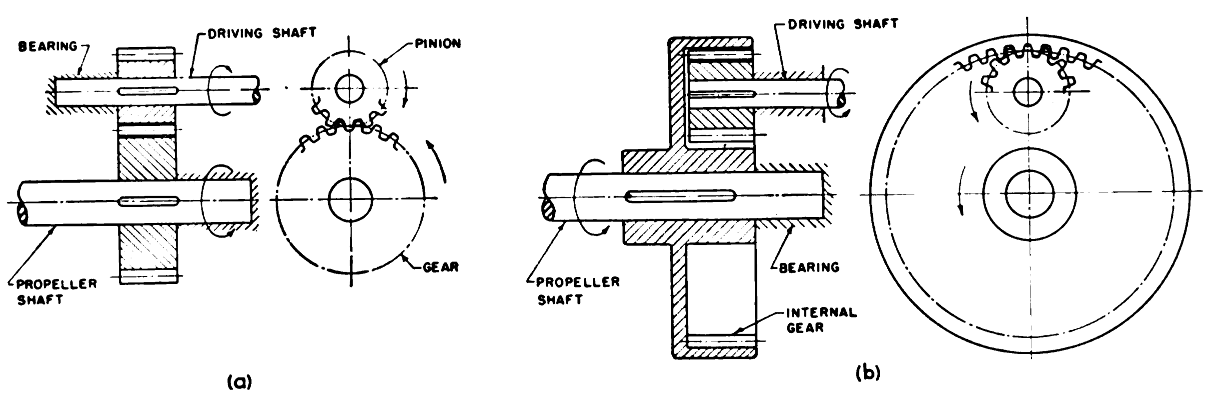


Figure 15–8. Types of reduction gears.

than one and the result will be a *speed reduction.* If the number of teeth in the drive is greater than in the driven gear, the quotient will be *smaller than one* and the result will be a *speed increase*, as found often in centrifugal pump drives. The examples will show the customary way of presenting the results.

> *Example 15–1.* What is the speed ratio if the driving pinion has 12 teeth and the driven gear 33 teeth?
> By definition (15–1): Speed ratio = 33 ÷ 12 = 2.75:1. This is a speed *reduction* of 2.75 to 1.

> *Example 15–2.* What is the speed ratio if the driving gear has 33 teeth and the pinion has 12 teeth?
> By expression (15–1): Speed ratio = 12 ÷ 33 = 1:2.75. In this case the speed is *increased* 2.75 times.

The internal-gear type unit consists of a pinion gear which mates with a gear having teeth on the internal cylindrical surface and mounted on a parallel shaft, as shown in Fig. 15–8b. Either straight- or helical-cut gear teeth are used. This type of reduction gear requires a smaller offset of the propeller-shaft centerline from the input shaft than the external gear units for a given reduction ratio. Also, both shafts turn in the same direction, whereas with an external reduction-gear train they turn in opposite directions, as shown in Fig. 15–8a. However, its design usually requires an *overhung* driving pinion, i.e., one which is supported on only one side by a bearing, thereby permitting greater shaft deflection under load and resulting in more noise and wear. These reduction gears are usually installed only on engines of relatively low power.

One type of *planetary* reduction gear consists of a driving or input gear *1*, Fig. 15–9, which meshes with three identical idler pinions *2;* these in turn mesh with one half of the long idler pinions 2, whose other halves are in mesh with the driven or output gear *4*. The speed reduction is equal to the ratio of the number of teeth in gear *4* to the number of teeth in gear *1;* the number of teeth in the idler pinions do not have any influence. As can be seen, the output gear rotates in the opposite direction to the input gear. The name planetary is used because, as first applied, a similar combination of gears and idler pinions was rotating, and resembled to a certain extent the movements of the planets.

The input and output shafts are on the same centerline, which permits a very compact unit for a specified reduction ratio. However, the disadvantages of this type of gear are its complicated construction, the increased number of operating parts, and the resulting increase in friction loss.

Lubrication of reduction gears is by the splash system in small units, and by oil under pressure from a built in pump in large units. Cooling of the lubricating oil may be either by water jacketing of the gear case, or by use of an oil cooler, or both.

15–6. Reverse gears. Reverse gears are used on marine diesel engines to reverse the direction of rotation of the propeller shaft when maneuvering the vessel, without changing the direction of rotation of the diesel engine. They are used principally on relatively small engines, usually those of not over 500 or 750 hp. If a high-output diesel engine has a reverse gear, the gear is used for low-speed operation only, and does

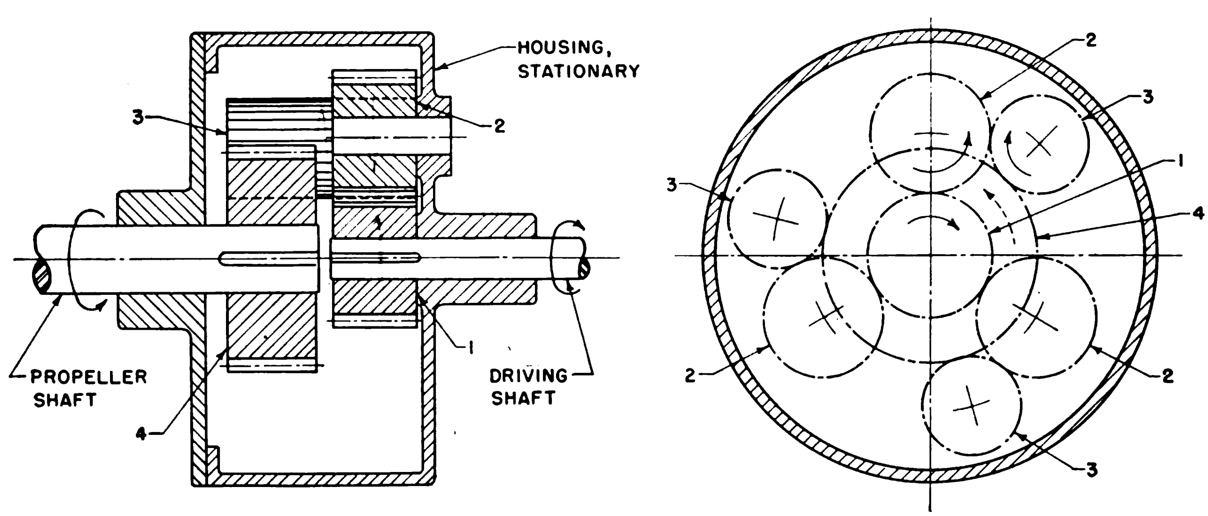


Figure 15–9. Planetary reduction gear.

not have full-load and full-speed capacity. For maneuvering vessels with large direct-propulsion engines, the engines themselves are reversed, as discussed in Sec. 14–5.

There are two general types of reverse gears, *selective*, and *planetary*.

Selective reverse gear. A selective-type reverse gear consists of two separate but similar drives which may be alternatively driven from the engine shaft by engaging one of the two friction clutches. The forward drive consists of a hollow outer shaft, Fig. 15–10, with a clutch plate *h*, which can be engaged with the engine shaft. The power in this case is transmitted to the propeller shaft through gears *1* and 2, and the propeller shaft rotates in the opposite direction to the engine shaft. When the engine shaft is disengaged from clutch *h* and engaged through clutch *s* to the solid inner shaft, the power is transmitted from pinion 3 through idler gear *4* to gear *5* on the propeller shaft. The propeller shaft, therefore, rotates in the same direction as the engine shaft, but in the opposite or reverse direction to which it formerly turned, and ahead drive now becomes an astern drive. This reverse gear is generally used also as a reduction gear by having a greater number of teeth in gear 2 than in gear *1*, and also in gears *5* and 3 respectively. The number of teeth in the idler gear *4* does not influence the speed ratio.

Fig. 15–11 gives one example of such a combined selective reverse and reduction gear. The floating plate is shown in its neutral, disengaged position. When the shifting collar is moved to the right, toward the engine, the lever with the inverted V-ends will pull the neutral plate toward the clutch cover and engage the ahead-drive friction plate secured to the hollow drive shaft. The propeller shaft will then begin to turn in its ahead-drive direction. When the shifting collar is moved away from the engine, it acts on the other end of the V-lever pushing the floating plate against the astern-drive friction plate, engaging the inner solid shaft and thus giving the propeller shaft its astern-drive rotation. The clutch compartment of this gear operates dry and the reduction gear has a self-contained lubricating-oil sump from which the gears and bearings are lubricated by splash.

Planetary reverse gear. A planetary reverse gear consists of: a housing with a planetary reduction gear in it, as shown in Fig. 15–9; a clutch to connect the input and output shafts directly during the ahead drive; and a brake by which the rotation of the gear housing can be stopped after the clutch is disengaged. With the gear housing standing still, the propeller shaft will begin to rotate in the opposite direction, as shown in Fig. 15–9.

A typical planetary reverse gear is shown in ahead-drive position in Fig. 15–12. To obtain the ahead drive, the engine, or input shaft, is engaged with the driven shaft by moving the shifting collar to the right, away from the engine end of the unit. The toggle linkage 33 will push the toggle plunger to the left and press together the plates of the plate clutch. Since alternate plates *13* of this clutch are splined to the driven shaft gear, and plates *11* to the gear housing, engagement of the plate clutch locks the gear housing and gear cage *4* to the propeller shaft. Simultaneously the cone clutch and its flat surface are engaged by the same toggle action. As the cone *5* is splined to the input-shaft gear, the gear cage *4*, which contains the spindles with the idler pinions, thus becomes locked to the input shaft, thereby transmitting the power of the input shaft to the propeller shaft.

In reverse drive, both plate and cone clutches are disengaged. The reverse brake band is clamped

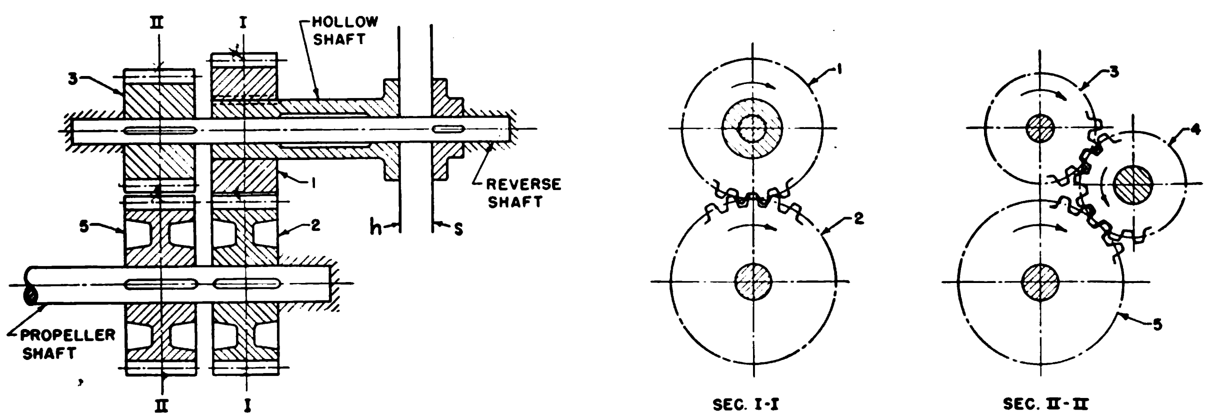


Figure 15–10. Selective reverse and reduction gear.

around the gear cage, preventing its rotation. The input-shaft gear then drives the three short pinions 7. The short pinions drive the long pinions (not shown), which in turn drive the driven-shaft gear. Thus, the drive from the input shaft to the driven shaft is through the idler pinions, giving a reversal of shaft rotation.

In neutral, the plate and cone clutches and the brake band are all disengaged, resulting in free rotation of the gear cage and pinions.

The friction clutches are of the wet type and run in oil which also lubricates all gear teeth and bearings.

15–7. Problems. 1. Find the speed ratio for a gear train in which the driving pinion has 25 teeth and the driven gear 51 teeth. *Ans.* Reduction 2.04 to 1.

2. Find the number of teeth of the gear to be keyed to the shaft of a centrifugal pump to run it at 1,350

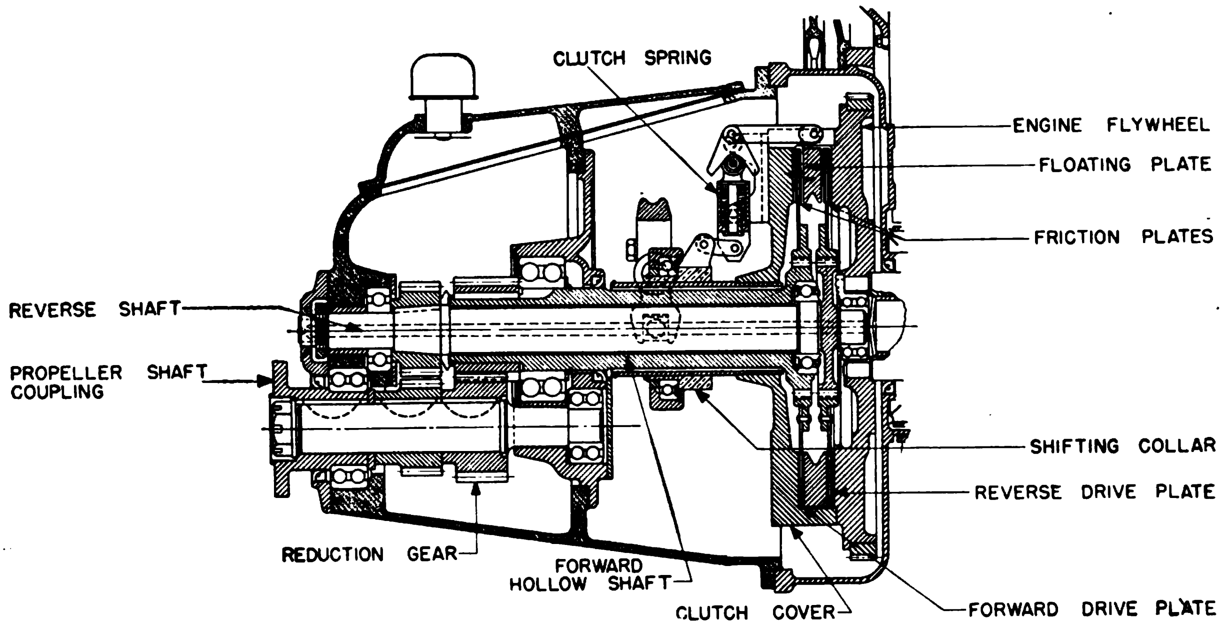


Figure 15–11. Selective-type reverse and reduction gear.

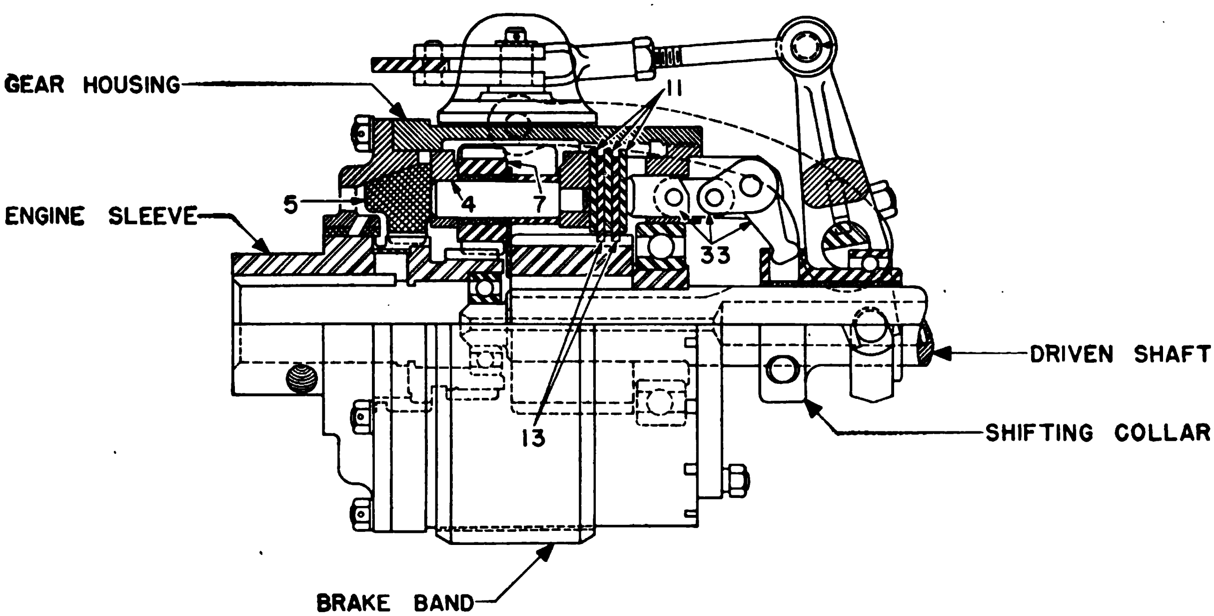


Figure 15–12. Planetary reverse gear.

rpm., if the speed of the drive shaft is 1,200 rpm. and the driving gear has 27 teeth. *Ans.* 24.

3. Find the speed of the propeller shaft driven through a reduction gear, Fig. 15-9, if the speed of the driving shaft is 2,200 rpm., pinion *1* has 24 teeth, pinions 2 have 23 teeth, pinions 3 have 21 teeth, and gear *4* has 48 teeth. *Ans.* 1,100 rpm.

4. Find the speed ratio of the reverse gear, Fig. 15-10, if the driving pinion *1* has 28 teeth, the idler pinions 3 and *4* have 25 teeth each and the gear *5* has 35 teeth. *Ans.* Reduction 1.25 to 1.

15-8. Questions. 1. What types of clutches are used with Navy diesel engines?

2. What are the advantages of wet-type friction surfaces?

3. What methods of operating friction clutches are in use?

4. What are the main advantages of a hydraulic clutch or coupling?

5. What are the advantages of a magnetic clutch?

6. Enumerate the types of reduction gears in use.

7. What are the two main types of reverse gears?

8. What types of friction clutches are used in connection with a planetary-type reverse gear?

CHAPTER 16

ENGINE MECHANICS

16–1. Piston and crank travel. The movements of the piston are transmitted to the crankshaft by means of a *crosshead*, in most cases incorporated in the piston, the *connecting* rod and the *crank*. By these three members the forward and back motion of the piston, called *reciprocating motion*, is transformed into rotary motion. For all practical purposes, the travel of the crankpin can be considered to be a uniform motion along a circle described with the radius R equal to the length of the crank-throw.

For convenience of presentation, Fig. 16–1 shows the motion of a piston along a horizontal line. If the connecting rod, instead of being L inches long, were infinitely long, then, at all positions of the crank, the connecting rod could be considered to remain parallel to the cylinder center line *m-c*, Fig. 16–1. In this case, at any position *1-c* of the crank, the travel of the small end of the connecting rod from top-center position m along the centerline would be equal to the distance *O-s*, cut off by the crank circle and the perpendicular *1-s*, drawn through the position of the crankpin.

The actual position of the small end of the rod, when it is L inches long, and of the piston, can be found by putting one leg of a compass at point *1* and swinging the compass with a radius equal to L to the intersection p with the horizontal centerline. The distance *m-p* will be the piston travel when the crank has traveled the angle a. On the other hand, putting one leg of a compass at point p and swinging the other leg to intersect the centerline at t will give the same distance $O\text{-}t=m\text{-}p$. The additional travel *s-t* is caused by the finite length L, or by the *angularity* of the connecting rod and is due to the connecting rod being inclined at an angle b instead of being parallel when it has an infinite length.

Thus, the angularity of the connecting rod during the first 20° of crank motion from the left or top center, increases the length of travel of the piston as compared with an infinite connecting rod. Conversely the angle of crank travel is smaller than with an infinite connecting rod. Thus, for the mid-position of the piston, the crank angle a_2 is smaller than 90°. Evidently the total piston travel or stroke $\overline{m\text{-}n}=\overline{O\text{-}r}=2R$.

16–2. Piston speed. While the crankpin travel is uniform, and has a constant velocity, the piston travel is not uniform and the piston speed constantly varies. At each dead center, the piston comes to a standstill, its speed becomes zero; as the piston begins to move, the speed gradually increases and reaches a maximum when the angle a, Fig. 16–1, formed by the crank and the cylinder centerline, is equal to 90°. After this position, the piston speed begins to decrease and at the dead center again becomes zero.

For many calculations, it is necessary to know the *average* or *mean* piston speed, or the constant speed with which the piston must move to travel the same distance in the same time as it travels with the actual variable speed. From the definition of velocity, this speed will be expressed as the *distance* traveled *divided by* the *time* in which it is traveled. It is customary to express the piston speed in ft./min. The distance traveled by the piston in one revolution is evidently two strokes. The piston stroke is usually measured

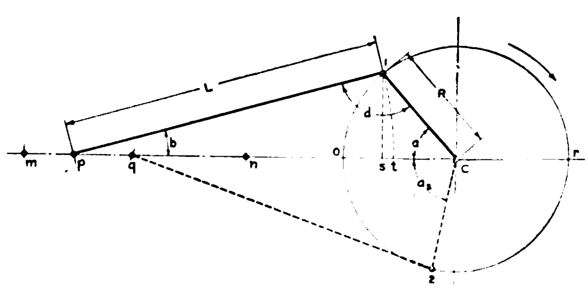


Figure 16–1. Crank and piston travel.

in inches. Dividing the stroke by 12 to change it to feet, and multiplying it by the number of revolutions per minute, or rpm., and cancelling out two, the distance traveled in one min. will be stroke×rpm.÷6; and at the same time, according to the definition of velocity, this will be the mean piston speed, usually called simply *piston speed*, or, briefly,

$$\text{Piston speed} = \text{stroke} \times \text{rpm.} \div 6. \qquad (16\text{–}1)$$

Example 16–1. Find the piston speed of an $8\frac{1}{2} \times 10\frac{1}{2}$ engine running at 750 rpm.
According to expression (16–1):
Piston speed = 10.5×750÷6 = 1,313 ft/min.

In present diesel engines the piston speed varies from 1,000 to about 2,000 ft./min.

16–3. Inertia. Inertia is the resistance of a body to a change of motion. It is the tendency of an object to remain at rest if it is stationary, or to continue to move if it is moving. Inertia as such cannot be measured directly; however, it can be expressed in terms of the force which must be applied to a body in order to change its velocity. As with any force, inertia forces are expressed in pounds. Since the change of velocity is called acceleration, therefore, inertia may be defined also as being equal to that force which must be applied to a body in order to impart to it a certain acceleration, either to speed it up or to slow it down, as the case may be. Numerically, the force of inertia F, is equal to the weight of a body W, divided by the acceleration of the force of gravity, $g = 32.2$, and multiplied by the acceleration a, which is imparted to the body.

$$\text{Inertia force} = (\text{weight} \div 32.2) \times \text{acceleration}$$

or

$$F = (W \div g)a \qquad (16\text{–}2)$$

Example 16–2. Determine the force of inertia of a body which weighs 12 lbs. and moves uniformly with a velocity of 15 ft./sec., if it is desired to stop the body in 2 seconds.
The acceleration, or since in this case it is negative, the deceleration per second must be, $a \div 15 \div 2 = 7.5$ ft./sec., and the negative acceleration force of F will be:
$F = (12 \div 32.2) \times 7.5 = 2.8$ lbs.

This means that if a steady force of 2.8 lbs. is applied to the body against its motion, the body will be brought to rest at the end of two seconds.

Equation (16–2) and the above example show that the force of inertia of a body is not a fixed but a variable quantity. The force of inertia of a body depends upon the acceleration which is applied to the body, or in other words, upon the rate of change of its velocity. The smaller the time during which a change takes place, the higher is the required acceleration and the greater becomes the force of inertia.

16–4. Inertia loads. As stated before, a change of velocity of a body cannot take place unless a certain new force is applied which will produce an acceleration or deceleration of the body, as the case may be. The force which must be applied is necessary to overcome the inertia of the body and can be computed by expression (16–2).

In the case of piston motion, the weight W in expression (16–2), means the weight of all parts connected to the piston and reciprocating with it, including part of the weight of the connecting rod. When the piston starts its travel from the top dead-center, its velocity is zero and the acceleration a in expression (16–2) has a maximum value. The force F must be taken from the uniformly rotating crankshaft through the connecting rod. Normally, the function of a piston is to furnish the force which does the work. Therefore, when the force is furnished from another source, such as the crankshaft, the force in respect to the useful piston forces is negative, or acting in the opposite direction.

As the piston begins to move and its speed increases, this negative force begins to decrease. When the piston speed reaches its maximum, slightly before the mid-position of the piston, this force becomes zero. After that the velocity of the piston begins to decrease and the force of inertia begins to act in the reverse direction, becomes positive, helps the piston to deliver power to the crankshaft, and gradually increases in its value. When the piston reaches the bottom dead-center, the force of inertia reaches another maximum. When the piston stops, the force of inertia becomes zero, and when the piston starts on the return stroke, the force of inertia will have the same value but will act in the opposite direction, thus becoming negative. During the return stroke the force of inertia will undergo the same changes as on the down stroke, but will be in the opposite direction.

The influence of inertia can be presented very clearly

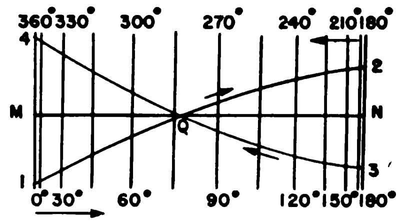


Figure 16–2. Forces of inertia, psi.

using the p-V diagram. Instead of using the total force of inertia in lbs., it is convenient to refer the force to one sq. in. of the piston area by dividing the force by the piston area. The result will be expressed in psi., or in the same units as the pressure in an indicator diagram. The piston travel will be represented by the abscissas, which, as explained in Sec. 6–1, at the same time are proportional to volumes, and the diagram is therefore, a p-V diagram.

Fig. 16–2 gives such a diagram, the curve *1-2* being for the downward stroke and *3-4* for the return stroke. The positive forces, or rather pressures, are shown by ordinates from the zero-pressure line upward, the negative pressures from the same line downward. Although separate areas, such as *1-q-M-1*, or *q-2-N-q* represent work absorbed or done by the piston, the net result must be zero-work and can be neither lost nor created. Therefore, the negative area *1-q-M-1* must be equal to the positive area *q-2-N-q*, and the same is true for the return stroke.

The forces of inertia are applied to the wrist-pin and crankpin bearings, and from the crankpin to the main bearings. The forces applied to bearings are commonly called *bearing loads*, and therefore, the forces of inertia are referred to as *inertia loads.*

16–5. Net-effort diagrams. The forces or loads applied to the engine bearings come from the gas pressures and from the inertia loads. The combining of these forces gives the so-called *net-effort diagram.* The diagram is obtained first by plotting the pressure from an indicator diagram of the *consecutive* strokes of the piston. When the pressures are acting in the *same direction* as the piston motion, they are considered *positive*, and when they are acting *against* the direction of piston motion, they are considered *negative*. Positive pressures are plotted above the zero-pressure line, negative pressures below it, the same as with inertia pressures, Fig. 16–2. In this case, the atmospheric line is used as the zero-pressure line.

The indicator diagram of Fig. 6–7, replotted in this

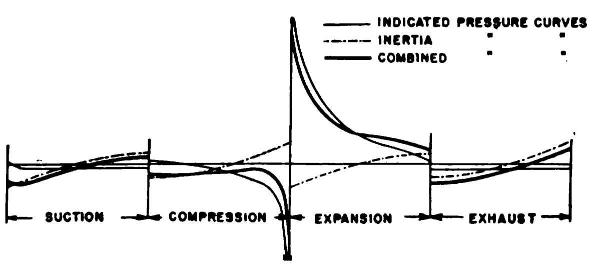


Figure 16–3.
Net-effort diagram for a single-cylinder engine, p-v.

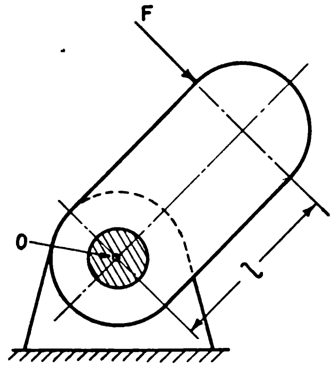


Figure 16–4. Torque.

manner is presented in Fig. 16–3, with light lines. After that, the pressures of inertia, such as shown in Fig. 16–2, computed for the same engine and drawn to the same spring scale as the indicator diagram, are plotted on Fig. 16–3 for the same consecutive strokes, using dot and dash lines. Finally, the pressures for the various consecutive points are combined; when the two pressures for a certain point have the same sign, whether positive or negative, they are added, and when at a certain point they have different signs, the smaller is deducted from the greater and the sign of the greater retained. The combined or resultant pressure lines are entered in Fig. 16–3 as heavy lines. The vertical distance from any point on the final curve to the zero-pressure line, multiplied by the spring scale, will give the net pressure at this point of the stroke in psi., and multiplied by the area of the engine piston, sq. in., will give the net effort or load acting on the piston.

16–6. Torque. Torque is the effect which rotates or tends to rotate a body. In order to produce rotation of a free body, there must be two equal and opposite forces acting along parallel lines but at separate points of the body. These two forces form what is known as a *couple*. The perpendicular or shortest distance between the lines of action of the forces is called the *arm* of the couple. The magnitude of the couple, called the *moment* of the couple, is expressed as the product of one of the forces, multiplied by the length of the arm of the couple.

When a body rotates about the point of application of one of the forces of a couple, as on a fixed pivot, the arm of the couple is known as the *lever*, and the turning moment is called the *torque*. If the lever is fastened to a rotating shaft, it is called a *crank*. Fig. 16–4 shows diagrammatically a force F, acting perpendicularly to a crank having a length l, from the center of rotation *0* of a rotating shaft. The torque T, acting on this shaft is expressed as:

$$\text{Torque} = \text{Force} \times \text{lever}$$

or

$$T = F \times l \qquad (16\text{–}3)$$

The units by which torque is measured are pound-feet (lb.-ft.) or pound-inches (lb.-in.).

16–7. Turning effort. *Torque diagram.* In the same way in which the reciprocating motion of the piston is transformed into rotary motion, so are the net efforts exerted by the piston transformed into *turning efforts* applied to the crank. The turning effort for any crank position is found as follows: The net effort or force F_p acting on the piston, Fig. 16–5, gives the force F_c acting along the connecting rod if the length $\overline{1\text{-}2}$ is made equal to F_p in a certain scale, and a perpendicular to *1-2* drawn at point 2 and extended to point 3, the intersection with the direction of the connecting rod. Length $\overline{1\text{-}3}$ will give the force F_c at the same scale as F_p was drawn. Next the force F_c is transferred to the crankpin, keeping the same direction, and there resolved into two components—one acting radially, from the crankpin toward the shaft center, and designated as F_r, the other at right angles to F_r and designated as F_t. Due to the characteristics of a circle, a straight line forming a right angle with a radius and drawn through its end is always tangent to the circle. In this case, the circle is the path of the crankpin and the tangential force is the turning effort. The product of the turning effort times the crank radius R, is the torque which at this instance is applied to the crankshaft.

$$\text{Torque} = \text{turning effort} \times \text{crank radius}$$

or

$$T = F_t R \qquad (16\text{–}4)$$

If the procedure of finding the turning effort F_t is carried through a complete cycle taking the net efforts F_p from a diagram as shown on Fig. 16–3, for every 10 or 30° of crank travel, and the forces F_t entered as ordinates using as abscissas crank travel taken to a certain scale, and a continuous curve drawn through the ends of the ordinates, a diagram is obtained, such as shown in Fig. 16–6.

This diagram is called a *torque diagram.* Part of the curve is above the zero-line *0-0*, Fig. 16–6, part is below, where the net efforts, Fig. 16–3, are negative. Next, the positive areas, above the zero-line, and the negative areas, below it, are measured separately and the negative values are subtracted from the positive value. The difference is divided by the total length of the diagram. The result will be F_t, Fig. 16–6, the mean turning efforts will do the same work as the actual variable turning effort does per cycle. Since the product of a force times the distance it travels represents work, therefore, the mean turning effort F_t, times the crank travel per cycle, $2\pi R \times 2 = 4\pi R$ where, R is the crank radius, represents the same work as the mean indicated pressure times four strokes. The area $F_t \times 4\pi R$, cross hatched vertically in Fig. 16–6, is equal to the area of the indicator diagram, for which it was drawn, assuming that the scales for forces and travels are the same in both diagrams.

In a four-stroke engine, the travel of the crankpin in one cycle is $4\pi R = 2\pi \times l$, where $l = 2R$ is the piston travel. The travel of the piston in one cycle or two revolutions is $4/l$. Therefore, the travel of the crankpin is $\pi \div 2 = 1.576$ times *greater* than the travel of the piston, and therefore, mean turning effort will be in the same proportion *smaller* than the mean indicated pressure $\times$ piston area.

For an engine with several cylinders, the torque diagram is obtained first by drawing the torque diagram for one cylinder and then superimposing the diagrams for the other cylinders. The torque diagram of each consecutive cylinder is gradually shifted the distance which the corresponding crankpin has traveled to reach the starting point of the first cylinder and the ordinates of the separate diagrams are added in order to obtain the ordinates of the combined diagram.

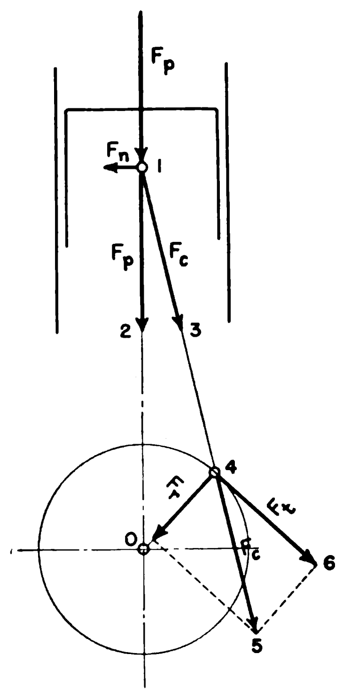


Figure 16–5. Finding of turning effort.

16–8. Flywheels. *Uniformity of rotation.* The torque diagram shows very clearly the flow of energy. Taking as reference the line *m-m*, Fig. 16–6, parallel to the zero-line and drawn a distance F_t above it, it will be readily seen from the former explanations that the sum of areas below this line is equal to the area *a* above it. Area *a* represents the *excess energy*, over the average one, developed during the working stroke, and the areas below represent the gradual consummation of this energy during each cycle.

A flywheel stores up energy which corresponds to the excess area *a* during the working stroke and gives it back during the rest of the cycle. A torque diagram shows that the crankshaft does not rotate uniformly. Whenever the turning effort is greater than the mean value, there is an excessive force acting and therefore, the crankshaft is being *accelerated;* whenever the turning effort is smaller than the mean value, or negative, there is a retarding force acting and the crankshaft is being *decelerated.* A torque diagram serves to determine the weight and diameter of a flywheel required for proper operation of the engine.

The flywheel serves four main purposes:

1. To keep the variations in speed within desired limits at all loads.
2. To limit the instantaneous rise or fall in speed during sudden changes of load.
3. To carry the pistons over the compression pressure when running at low or idling speed.
4. To help bring the engine up to speed when starting.

In multicylinder engines the torque at the end of the crankshaft becomes more uniform, and the required weight of the flywheel becomes very small. The cranks, crankpins, and large ends of the connecting rods have considerable rotating weight and exert the same influence as a flywheel. Therefore, in some multicylinder marine diesel engines, flywheels are not necessary and hence are not used.

Uniformity of rotation. As already stated, the crankshaft of a diesel engine does not turn uniformly, since the speed of rotation increases every time that the crankshaft receives a power impulse from one of the pistons, and after that begins to decrease until another impulse comes from the cylinder which fires next, etc.

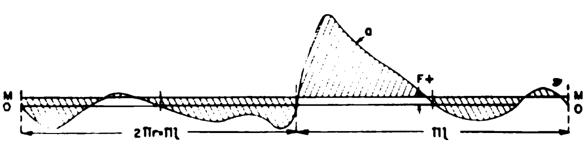


Figure 16–6.
Torque diagram for a four-stroke single-cylinder engine.

The number of speed fluctuations during one revolution is equal to the number of cylinders firing per revolution. In a two-stroke engine it will be equal to the number of cylinders and in a four-stroke engine equal to one-half of the number of cylinders.

If an instrument similar to a tachometer, but more sensitive, were to be used for measuring the speed of rotation of a diesel engine, its pointer would fluctuate between a maximum rpm. and a minimum rpm. reading. On the other hand, a revolution counter gives the engine speed as a mean value of rpm., which is somewhere between the maximum rpm. and the minimum rpm. indicated by the sensitive tachometer and rather close to their average. The ratio between the mean rpm. and the difference between the maximum rpm. and minimum rpm. is a measure of the *uniformity of rotation* of an engine and is called the *coefficient of steadiness.*

$$\text{Coeff. of steadiness} = \frac{\text{mean rpm.}}{\text{max. rpm.} - \text{min. rpm.}} \qquad (16\text{–}5)$$

Uniformity of rotation depends upon the weight of the flywheel and other rotating parts, on the engine speed, and on the number of working strokes per revolution. Thus, all other things being equal, a multicylinder engine has a better uniformity of rotation, i.e., a higher coefficient of steadiness, than an engine with a fewer number of cylinders, and for the same reason a two-stroke engine has a greater uniformity of rotation than a four-stroke engine. The influence of speed is very great and the coefficient of steadiness is proportional to the square of the engine rpm. Therefore, when the speed of an engine is reduced, the coefficient of steadiness drops rapidly. Engines operating with variable speeds, such as boat and ship engines connected to a propeller, usually have a very low uniformity of rotation at idling speed and a very high uniformity at full speed.

In diesel engines used in the Navy, the coefficient of steadiness varies from about 30 in small powerboat engines up to 200 or even higher in multicylinder engines direct-coupled to electric generators.

16–9. Speed factor. Engines are often divided into several classes according to their *speed capacity.* Some are classified as low-speed engines, others as medium-speed, and some as high-speed engines. However, unless a definite yardstick is used, the designations remain vague. There were attempts to use either the engine speed, rpm., or its piston speed, ft./min., as a measure of speed capacity, but neither of these two methods can give correct indications. Rotative speed, as such, is not suitable as a speed characteristic because it does not take into consideration the size of the engine. A 6-cylinder$\times 3\frac{1}{2} \times 4\frac{1}{2} \times$ 900 rpm. engine is not a high-speed engine, as this type of engine normally operates at speeds up to 2,000 rpm. and higher. On the other hand, $8\frac{1}{2}$ in. $\times 10\frac{1}{2}$ in. diesel engines usually operate at speeds not exceeding 750 rpm., and even at this lower speed have many features common with high-speed engines.

The same is true, only in reverse, in respect to piston speeds. In a large engine a relatively high piston speed, 1,800 ft./min. or more, may be obtained with a relatively low rpm.; in a small high-speed engine the piston speed is not high.

A good speed characteristic, called ***speed factor***, is obtained as a product of rpm. and piston speed. For the sake of obtaining smaller, more easily remembered figures, the product is divided by 100,000. Thus:

$$\text{Speed factor} = \frac{\text{rpm.} \times \text{piston speed (ft./min.)}}{100{,}000} \qquad (16\text{–}6)$$

The figures obtained for various existing diesel engines lie between the limits of 1 and slightly less than 81. According to this data, all engines can be divided into four classes, in each class the high limit being obtained by multiplying the low limit by 3:

1. Low-speed engines with a speed factor of 1 to 3.
2. Medium-speed engines with a speed factor of 3 to 9.
3. High-speed engines with a speed factor of 9 to 27.
4. Super-high-speed engines with a speed factor of 27 to 81.

Example 16–3. Find the speed factor and speed classification of a $16 \times 8\frac{1}{2} \times 10\frac{1}{2} \times 750$ engine.

As found in Example 16–1, the piston speed is 1,313 ft./min., and therefore, by expression (16–6):

$$\text{Speed factor} = \frac{750 \times 1{,}313}{100{,}000} = 9.85$$

According to the table given above, the engine, having a speed factor over 9 but under 27 should be classified as a *high-speed* engine.

Practical meaning. Classification of an engine in one of the above-named groups according to its speed factor has a particular value for purposes of designing the engine. The knowledge of what speed group an engine belongs in is of value also to the engine operator: the higher the speed classification of an engine, the more attention the operator should pay to keeping the engine in its best possible running condition, by observing every detail given in the manufacturer's instruction book, and the more careful should he be when inspecting or overhauling the engine.

16–8. Problems. 1. Find the piston speed of a $6 \times 5 \times 6 \times 1{,}500$, four-stroke engine. *Ans.* 1,500 ft./min.

2. Find the torque developed by a $6 \times 4 \times 5\frac{1}{2} \times 1{,}700$, Navy engine, if the mean turning effort on each crank is 134 lbs. *Ans.* 184.2 lb.-ft.

3. Determine the coefficient of steadiness for an engine running at 2,000 rpm., if the speed fluctuates during each cycle, between 2,016 and 1,984 rpm. *Ans.* 62.5.

4. Find the speed factor of a $6 \times 4 \times 5\frac{1}{2} \times 1{,}700$ engine. *Ans.* 26.5.

16–9. Questions. 1. How does the length of the connecting rod influence the piston travel as compared with an infinitely long connecting rod?

2. What is piston speed and how is it determined?
3. What is meant by inertia loads in a diesel engine?
4. What is a net-effort diagram?
5. What is a torque diagram?
6. What are the main purposes of an engine flywheel?
7. What is coefficient of steadiness?
8. What is meant by the term the speed factor in Navy diesel engines?
9. What is the practical meaning of the speed factor?

CHAPTER 17

VIBRATIONS

17–1. Introduction. If an elastic body is displaced from a position of equilibrium, it will develop a restoring force which tends to return it to that state. Therefore, when the displacing force is removed, the body will move toward its original position. Due to its inertia, the body, on its return movement, will not stop at its previous position of equilibrium, but will be carried beyond, causing a displacement in the opposite direction. The restoring force thus developed in the opposite direction will reverse the action and the process will continue until these consecutive movements called *oscillations* or *vibrations* are dissipated or *damped out* by friction or other resisting forces. The number of vibrations per second, or the rate at which they occur, is known as the *frequency* of vibration. If an elastic body is allowed to vibrate freely, it will do so at a certain rate, known as its *natural frequency* of vibration, which depends upon its shape and the material of which it is made.

When displacing forces occur repeatedly on an elastic body, they cause vibrations known as *forced* vibrations. When these forced vibrations occur at the same rate as the natural frequency of vibration of the body, or some multiple of it, called a *harmonic*, then the free vibrations will be reinforced or amplified by the forced vibrations. Under this condition, known as *resonance*, the displacement or amplitude of the resultant vibrations will be greatly magnified and may produce excessive stresses in the body.

Vibrations in engines occur due to displacing forces, resulting from various unbalanced forces acting in the engine. If all the forces in an engine were constant in magnitude and direction, they could easily be balanced. Actually, however, the forces within an engine are constantly changing both in magnitude and direction, and are therefore, difficult to balance. The problem of balancing these changing forces is made more difficult by the reciprocating motion of certain parts. Engine vibrations may thus occur due to unbalanced rotating forces, unbalanced reciprocating forces, and variations in gas pressure, inertia forces, and torque.

If these fluctuating forces in an engine occur at the same rate as the natural frequency of vibration of the engine structure or one of its parts, the resulting condition of resonance may increase the amplitude of the vibrations to such an extent that serious damage will result. Generally, the natural frequencies of the engine structure and its parts, are considerably higher than the frequency at which the unbalanced forces in the engine are likely to occur under normal operating conditions. However, if these forced vibrations occur so that they reinforce every second, or third, or fourth natural vibration and so forth, of a particular engine part, a condition of resonance may occur at one of

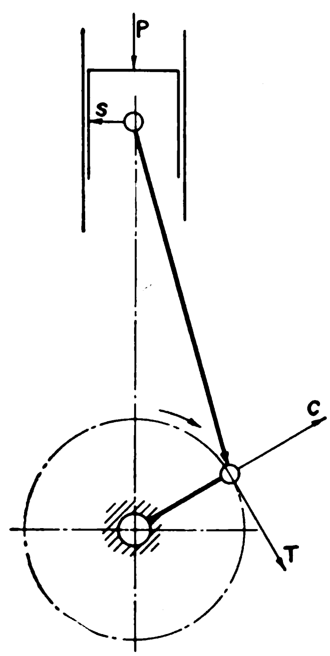


Figure 17–1. Acting engine forces.

these higher harmonics of the natural frequency of vibration. While resonance with these harmonics of higher order will not produce as great magnification of the vibrations as with low-order harmonics, they are more likely to occur and are responsible for the so-called *vibration points* or *critical speeds* within the normal operating speed range of the engine.

The gas pressure *P*, Fig. 17–1, acting on the piston of a diesel engine, is resolved into two types of forces: (1) a tangential force *T* turning the crankshaft, and (2) the side thrust *S* on the cylinder walls. To this is added the centrifugal force *C*, created by the rotation of the crankshaft with various parts attached to it. All these forces vary in size, direction, or point of application, and all can cause some kind of vibration.

17–2. Engine vibrations. Engine vibrations may be classified by the type of displacements which they cause (see Fig. 17–2):

Shaking: caused by fluctuating vertical or horizontal forces which tend to move the engine up-and-down or sideways, respectively.

Rocking: caused by fluctuating horizontal forces acting above the center of gravity of the engine and tending to rock the engine about a line passing through its center of gravity.

Pitching: caused by fluctuating vertical couples which tend to make the ends of the engine rise and fall.

Yawing: caused by fluctuating horizontal couples which tend to turn the engine crossways or move the ends to the left and right.

Torsional: caused by fluctuating torque reactions which tend to twist the crankshaft as it rotates.

From the above definitions it will be seen that *shaking* is due to the unbalanced reciprocating forces and vertical or horizontal components of the centrifugal forces, while *pitching* is the result of the unbalanced couples produced by these forces. *Rocking* is caused primarily by variations in the horizontal component of the piston reaction or side thrust, as shown in Fig. 17–1, due to changes in gas pressures, inertia forces, and load reaction. *Yawing* occurs chiefly as a result of the unbalanced couples produced by the horizontal components of the centrifugal forces in a vertical engine, and by the horizontal components of the reciprocating forces in a V-type engine. *Torsional* vibrations are due primarily to the variations in torque which are caused by changes in gas pressure, inertia forces, and torque load reactions.

While all these vibrations are to some extent interrelated, the gas pressures and load reaction will have little effect on the engine vibrations other than torsional and rocking. Shaking, pitching, and yawing vibrations are due to the unbalanced reciprocating and rotating forces and couples which occur in all engine loads unless the engine is equipped with means to balance them.

In addition, there are internal vibrations in the engine structure itself due to fluctuations in the gas pressures and inertia forces. This is evidenced in engine roughness, which occurs at certain vibration points when the frequency of the fluctuating forces coincides with the natural frequency of the engine structure or some multiple of it. For the purpose of preventing resonance with these vibrations, the engine frame is made as rigid as possible to increase its natural frequency of vibration.

17–3. Unbalanced engine forces. As was explained in Sec. 16–4, the inertia forces become considerably greater as the speed of the engine increases. These reciprocating forces are considered as acting only along the line parallel to the axis of the cylinder.

In order to determine the magnitude of unbalance in a reciprocating engine, the moving parts may be considered as divided into those which are reciprocating and those which are rotating. The small end and the adjacent part of the connecting rod are considered as reciprocating, while the large end and the rest of the connecting rod are considered as rotating. The reciprocating weight thus consists of the piston, piston rings, piston pin, and the weight of the upper end of the connecting rod. The rotating weight consists of a weight, assumed to be concentrated at the radius of the crank, which will produce a centrifugal force equivalent to that of the entire crank structure plus the weight of the lower end of the connecting rod.

The magnitude of the inertia force of the reciprocating parts varies throughout each stroke, as shown

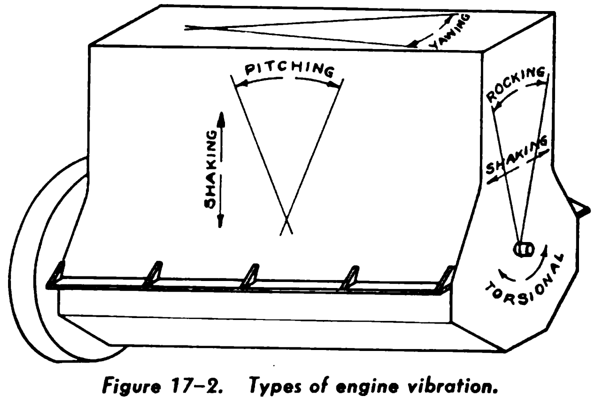


Figure 17–2. Types of engine vibration.

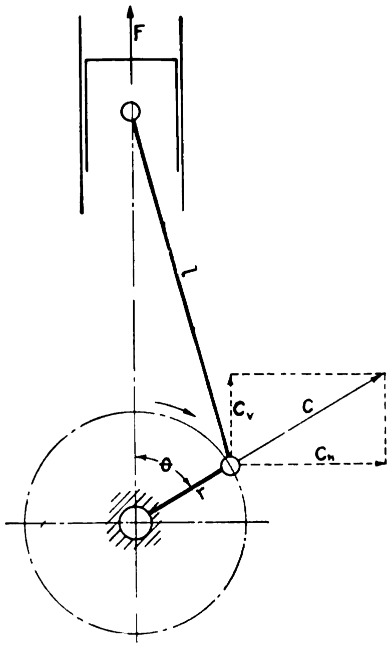


Figure 17–3. Action of centrifugal force.

in Fig. 16–2. This force can be considered as consisting of the sum of two separate forces, a *primary* force and a *secondary* force. The primary force depends upon the weight of the reciprocating parts, as enumerated above, and upon the crank radius and engine speed in rpm. The primary force is equal to the inertia force which would be produced if the connecting rod were infinitely long. The secondary force represents the influence of the angularity of the connecting rod, as explained in Sec. 16–1. It depends upon the same factors as the primary force and in addition, upon the ratio of the length of the connecting rod to the crank radius. Due to their nature, the primary force alternates in magnitude and direction once per revolution, while the secondary force alternates twice per revolution.

The centrifugal force of the rotating parts always acts radially outward along the line of the crank, and thus moves around the shaft at crankshaft speed. As indicated in Fig. 17–3, this centrifugal force may be resolved into two perpendicular components: one acting along the cylinder axis, and the other acting across the cylinder axis. As the crank rotates, the magnitude of these two component forces changes with each position of the crank, even though the resultant centrifugal force remains constant at a given speed. Since these forces alternate in magnitude and direction once per revolution, they also are considered as part of the primary forces. By combining the components of the primary rotating and reciprocating forces, the resultant unbalanced primary forces are obtained.

Crankshaft balance. A crankshaft whose center of gravity coincides with its centerline is said to be *statically* balanced. With its ends supported on two horizontal knife edges, the shaft will be in a stable position at any position of the crank, and will have no tendency to roll. The centrifugal forces from the rotating parts of each crank structure of a statically balanced crankshaft, if considered to act in the same plane, will all balance each other. Actually, however, the centrifugal forces of the rotating parts of a crankshaft do not act in the same plane but in different planes, at various distances from the middle point of the shaft. These forces therefore, will form *couples* (Sec. 16–6) which, unless balanced, will tend to cause pitching and yawing as the crankshaft rotates. When these *couples* are completely balanced, the crankshaft is said to be in *dynamic balance.*

Before the other moving parts of an engine can be balanced, its crankshaft must be in dynamic balance. Some crankshafts, by virtue of their crank arrangement, are always in dynamic balance. Others, however, must be dynamically balanced by means of weights, known as *counterbalances*, or *counterweights* placed opposite their cranks. By the proper choice of the weight of the counterbalance and the point at which it is placed, its centrifugal force may be made exactly equal to and acting in the opposite direction from the centrifugal force of the other rotating parts of the crankshaft. A crankshaft constructed with counterbalances opposite every crank-throw, to balance all the centrifugal forces, will thus also be in dynamic balance.

Counterbalances, however, are not always necessary, and since they add extra weight to the crankshaft, are not always used. Thus, crankshafts which have their cranks arranged in symmetrical pairs, each of which acts at the same angle and distance on either side of the middle point of the shaft, will be in dynamic balance, because the couples acting about the center are equal and opposite.

In the case of multi-throw crankshafts, in two-stroke-cycle engines, the centrifugal forces of the symmetrical cranks on each side of the center of the shaft act at equal distances but in different directions. These forces will form couples which are not balanced, and consequently the shaft is not in dynamic balance. In order to balance such a shaft dynamically, counterweights must be added to produce couples which are exactly equal and opposite to the resultant unbalanced couples. While this could be accomplished with counterbalances opposite each crank-throw, some crankshafts are made with counterweights only opposite the outer cranks as shown in Fig. 7–6. By placing these counterweights at a certain angle with the outer crank-throw instead of directly opposite it, the unbalanced couples from the inner cranks will also be balanced. In this way, while the centrifugal forces from all cranks are not individually balanced, the resultant couples produced by them, which would tend to cause pitching or yawing, are balanced by the counterbalances at each end and the crankshaft is in dynamic balance.

17–4. Balancers and vibration dampers. As stated in the above discussion concerning the components of centrifugal force, the primary reciprocating force has the same nature as the vertical component of the centrifugal force. Thus, if a counterbalance is given an excess weight equal to that of the reciprocating parts and placed at a radius equal and opposite to the crank, the primary reciprocating forces will be balanced. The horizontal components of this added centrifugal force, however, will be unbalanced.

This principle is employed in single-cylinder engines to balance part of the reciprocating forces. A weight which produces a centrifugal force equivalent to one-half the reciprocating forces is added to the counterbalance. At both the top and bottom of the stroke, this excess centrifugal force opposes the reciprocating force and thus reduces its magnitude by one-half. At the position midway in the stroke, however, since there is no reciprocating force to balance it, the excess centrifugal force acts across the axis of the cylinder. The net result is to balance one-half of the reciprocating forces, but causing an equivalent unbalancing of the horizontal components of the rotating forces.

In engines with two or more cylinders, all primary forces are generally balanced by the use of symmetrically spaced cranks, or by counterbalances as previously explained.

Primary couple balancers. In engines which have unbalanced primary couples, particularly two-stroke engines of 2,3,4 and 6 cylinders, an equal and opposite couple must be provided which has the same frequency as the crankshaft. One method of providing this type of balance, is the use of two balance shafts driven in opposite directions at the same speed as the crankshaft, and provided with counterweights at each end, as shown in Fig. 17–4. As will be seen from the illustration, the centrifugal forces due to the counterweights when they are in a horizontal position, are always balanced. When the weights are in the vertical position, the centrifugal forces will act up at one end and down at the other end of the engine and will produce a definite couple. The weights of these counterbalances on the balance shafts must be such that the resulting couple will be equal and opposite to the unbalanced couple of the crank mechanism, and will thus balance it. One of the counterbalance shafts also acts as the valve camshaft, and the counterweights at one end are built into their drive gears *6, 6*. This arrangement will balance the primary couple which would normally tend to cause pitching vibrations.

Vibration dampers. In spite of all precautions used, it is not possible to balance all forces and couples which occur in a reciprocating engine. In order to absorb the vibrations which cannot be balanced, the engine must be provided with dampers which prevent injury to the engine or its mount. Engine-mount vibration dampers consist essentially of a flexible engine-support made of rubber, cork, or springs. In order to absorb all types of vibration, this support should allow a small deflection of the engine in any direction, and absorb the vibrations without transmitting them to the foundation. An example of an engine vibration damper with coil springs, also called vibration isolator, is shown in Fig. 17–5. However, the fitted pins in this damper restrict its usefulness to absorption of vibrations due only to vertical shaking, pitching, and rocking. Fig. 17–6 shows a vibration damper in which heavy rubber pads are interposed between the engine-bed plate and the floor plate of the boat which allows a small displacement in all directions.

17–5. Torsional vibration. If a long steel shaft with a flywheel fastened at one end is clamped in a vise at the other end, and the flywheel is given a slight turn, the shaft will be twisted. When the flywheel is released, the shaft will untwist itself, thus turning the flywheel back in the opposite direction.

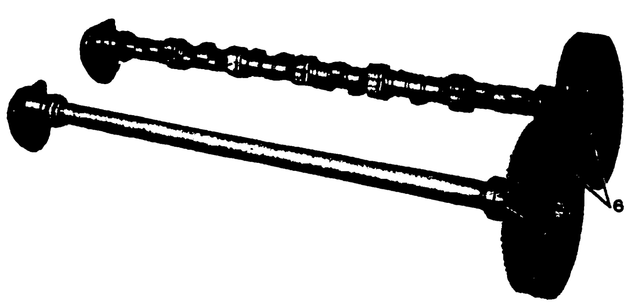


Figure 17–4. Cam and balance-shaft assembly.

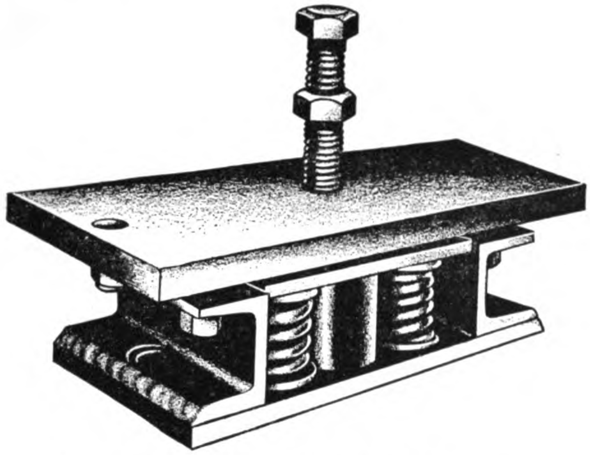


Figure 17–5. Vibration damper with coil springs.

Because of its inertia, however, the flywheel will not stop turning when the shaft is fully untwisted, but will continue to turn until the shaft is twisted up in the opposite direction. Since steel is elastic, this shaft will continue to twist and untwist in opposite directions until the internal friction between the steel fibers slows it down, or damps out the angular vibration.

The twisting and untwisting of a shaft is known as *torsional* vibration. Every shaft has a certain natural frequency of torsional vibration depending on its size, shape, elasticity, and attached weights. For instance, a long, thin, steel shaft with one end firmly secured and the other end fastened to a heavy flywheel will vibrate, or twist and untwist, much more slowly than will a short thick steel shaft to the free end of which the same flywheel is attached. The long thin shaft, together with the flywheel, is thus said to have a lower natural frequency of torsional vibration. The shorter and thicker an engine shaft is, the higher will be the natural frequency of its torsional vibration.

In an engine crankshaft, the forces exerted by firing in the cylinders tend to twist the crankshaft while it is rotating. This gives rise to periodic vibrations which in normal operations are so small that they are not evident and are not ordinarily harmful. If, however, these periodic impulses happen to coincide with the natural frequency of torsional vibration of the engine crankshaft and attachments, then these impulses reinforce each natural vibration, and the resulting vibrations may become so great that the stresses in the crankshaft will exceed the safe value and the shaft finally will break. However, even before the crankshaft breaks, the excessive torsional vibration may cause other serious damage, such as pounding or breaking teeth in the gears that are driven by the crankshaft.

The speed of crankshaft rotation at which these twisting impulses occur in the same rate as the natural frequency of torsional vibrations, is called the *critical speed of the first order*. The crankshafts on all properly designed engines have natural frequencies so high that the critical speed of the crankshaft system can never be reached in normal operation.

If an engine is run at a speed at which the twisting impulses occur at a rate which is half of the crankshaft's natural frequency of torsional vibration, the twisting impulses will reinforce every second natural vibration. The speed at which this vibration point occurs, is called the *critical speed of the second order*. Similarly, depending upon the engine characteristics, critical speeds may occur at which these impulses reinforce every third or fourth natural vibration, and so on. This is called the critical speed of the third or the fourth order, etc. When, in bringing an engine up to speed, a crankshaft vibration point is reached, it is usually due to a critical speed of higher order, which, although not so destructive as the first-order critical speed, should be run past as quickly as possible to avoid any possible detrimental effects. Where an engine is to be run at a constant speed, other than the rated speed for which it was designed, it must not be at or near any critical speed.

High-speed, multi-cylinder engines are more likely to run through critical speeds than low-speed engines, because the size and weight limitations prevent the use of large-diameter crankshafts, and yet, a long shaft is necessary. A large number of cylinders and high operating speed result in a great number of firing impulses tending to twist the crankshaft. These factors tend to bring the low-order critical speeds within the operating speed range of the engine.

In order to obtain high natural frequencies, crankshafts are made as short and as large in diameter as

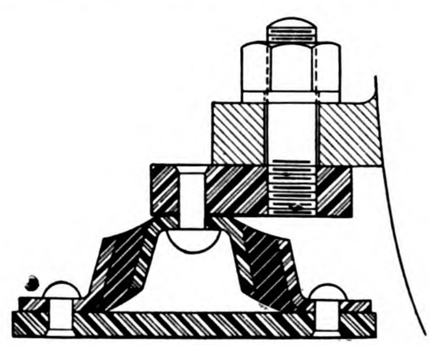


Figure 17–6. Vibration damper with rubber.

possible. In order to keep the weight down, and yet assure sufficient rigidity to avoid critical speeds within the operating range of the engine, some crankshafts are made large in diameter but hollow. For the same reason, the flywheels are made as light in weight as possible.

Torsional vibration dampers. It is not always possible to make a crankshaft so rigid that resonance with some of the higher-order harmonics of its natural frequency will not occur within the operating speed range. Therefore, crankshafts on multi-cylinder, high-speed engines, are generally equipped with torsional vibration dampers which prevent vibrations from building up to dangerous amplitudes when operating at or near a critical speed.

Torsional vibration dampers limit the amplitudes of the vibrations by providing frictional resisting forces which oppose them. The vibrating energy is thus converted into heat which is dissipated without harmful affects on the engine.

Torsional vibration dampers usually consist of a small, flexibly driven flywheel mounted at the opposite end of the crankshaft from the main flywheel. In some cases, the damper flywheel is driven by a spring-loaded friction clutch which tends to keep it rotating at crankshaft speed. If the crankshaft starts vibrating torsionally, it will tend to turn the damper flywheel either faster or slower as it twists first one way and then the other. But the inertia of the damper flywheel will tend to keep it rotating at constant speed and will thus oppose the vibrating forces. By adjusting the tension of the springs on the friction-driven clutch, it can be made to slip when the vibrating forces reach a certain magnitude. The vibrating energy will then be converted into heat by doing work against the clutch friction, and this heat is dissipated to the surroundings.

A similar effect is obtained by driving the damping flywheel through a rubber coupling. In this case, the vibrating forces will tend to deflect the rubber and thus expend their energy against the internal friction of the rubber which is dissipated in heat as before. In order to take care of vibrations occurring at different speed ranges, some engines are equipped with a vibration damper consisting of two small flywheels of unequal weights so that their resisting forces will be most effective at different speeds.

Another type of torsional vibration damper, sometimes called a *harmonic balancer*, consists of an auxiliary flywheel driven by flexible leaf springs. The vibrating energy in this case is dissipated by the friction between the leaves of the springs sliding over each other. The same principle is also employed in elastic couplings between the crankshaft and the main flywheel, which serve to damp out some of the torsional vibrations. This type of damper usually operates in lubricating oil which assists in damping the vibrations by its viscous friction and also carries away the heat generated. In a similar manner, hydraulic couplings or fluid drives also damp out and prevent the transmission of torsional vibrations between the crankshaft and the driven load.

Pendulum-type damper. This device consists of two or more symmetrically located, heavy steel segments or balances, Fig. 17-7, suspended so that they can swing in the plane of rotation, similar to a pendulum. The weight of the segments and length of the links are so selected that the natural frequency of swinging of the segments is equal to the frequency of vibrations of the shaft system which must be damped out. During undisturbed rotations of the shaft, the centrifugal force keeps the pendulum weights at the greatest distance from the axis of rotation. When vibration of the shaft starts, the weights will begin to swing and thus will be periodically drawn nearer to the shaft axis, Fig. 17-7. The work of bringing the pendulums closer to the axis against the resistance of the centrifugal force, is derived from the energy put into the shaft when it starts to vibrate, and thus reduces this energy and damps out, or at least, considerably reduces the angle of torsional vibration of the shaft. In actual construction, the pendulums are suspended, not on links but on *pins* inserted through holes, which are

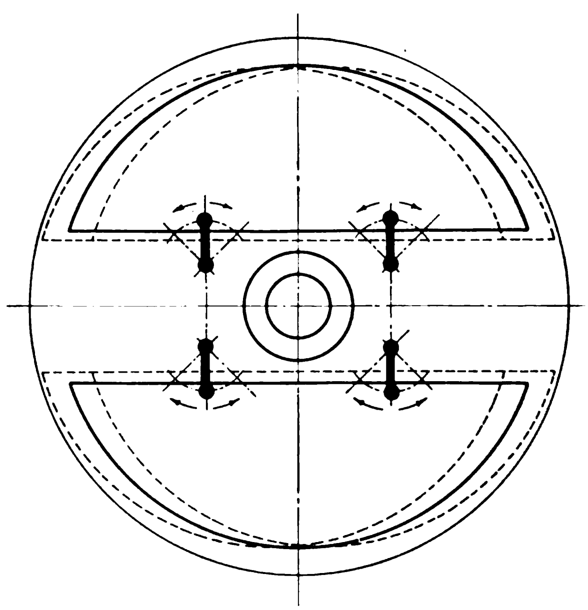


Figure 17-7. Action of a pendulum vibration damper.

drilled in the sectors and in the housing and are slightly larger than the pin diameter. However, their action is the same as with the use of the links.

17–6. Valve spring vibration. Vibration also may influence the operation of the valve springs of an engine. Whenever a sudden blow is struck on one end of a coil spring, as by the action of the valve-operating cam, it will tend to start the coils vibrating. These vibrations are transmitted through the coils at the rate of its natural frequency of vibration. When repeated blows are struck on the end of the valve spring at a rate which is the same as the natural frequency of the spring, or some multiple or harmonic of it, then the vibrations will be amplified by each impulse. Under this condition, known as resonance, the amplitude of vibration of the coils will be greatly magnified and spring surge may result.

Spring surge. When surge occurs in a spring, the coils are compressed by the compressive wave of vibration as it travels from one end of the spring to the other. Since the end coils of a valve spring are fixed, most of the surge occurs in the center coils. Under unfavorable conditions, these coils are compressed sufficiently to reduce the effective force of the entire spring, and may permit the valve gear to bounce. Another serious result of surge is breakage of the valve springs due to the extreme compression of a few coils at the end of the spring when they come together with a shock instead of having an equal compression of all coils as in normal operation.

Valve-spring surge will occur at certain engine speeds when the impact of the cams on the valve gear happens to be in resonance with some harmonic of the natural frequency of vibration of the spring. In order to eliminate surge, the natural frequency of vibration of valve springs is made as high as possible, usually several times as great as the maximum camshaft speed.

Where a single spring cannot be designed with a sufficiently high natural frequency, two or more smaller springs are used, since smaller springs have a higher frequency of vibration. Multiple-valve springs usually have different natural frequencies so that surge will not affect all of them at the same time, and if one spring should break, the engine could still operate temporarily with the remaining spring or springs.

17–7. Questions. 1. What is meant by the natural frequency of vibration of an engine part?

2. What is meant by the critical speed of an engine?

3. Enumerate the types of vibrations which may be produced in a diesel engine.

4. What may cause vibrations in a diesel engine?

5. What is the main object of counterweights on a diesel engine crankshaft?

6. What is a primary-couple balancer?

7. What is the object of using vibration-damping mountings?

8. What is torsional vibration?

9. What methods are used to damp out torsional vibration of a diesel engine crankshaft?

10. What is spring surge?

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