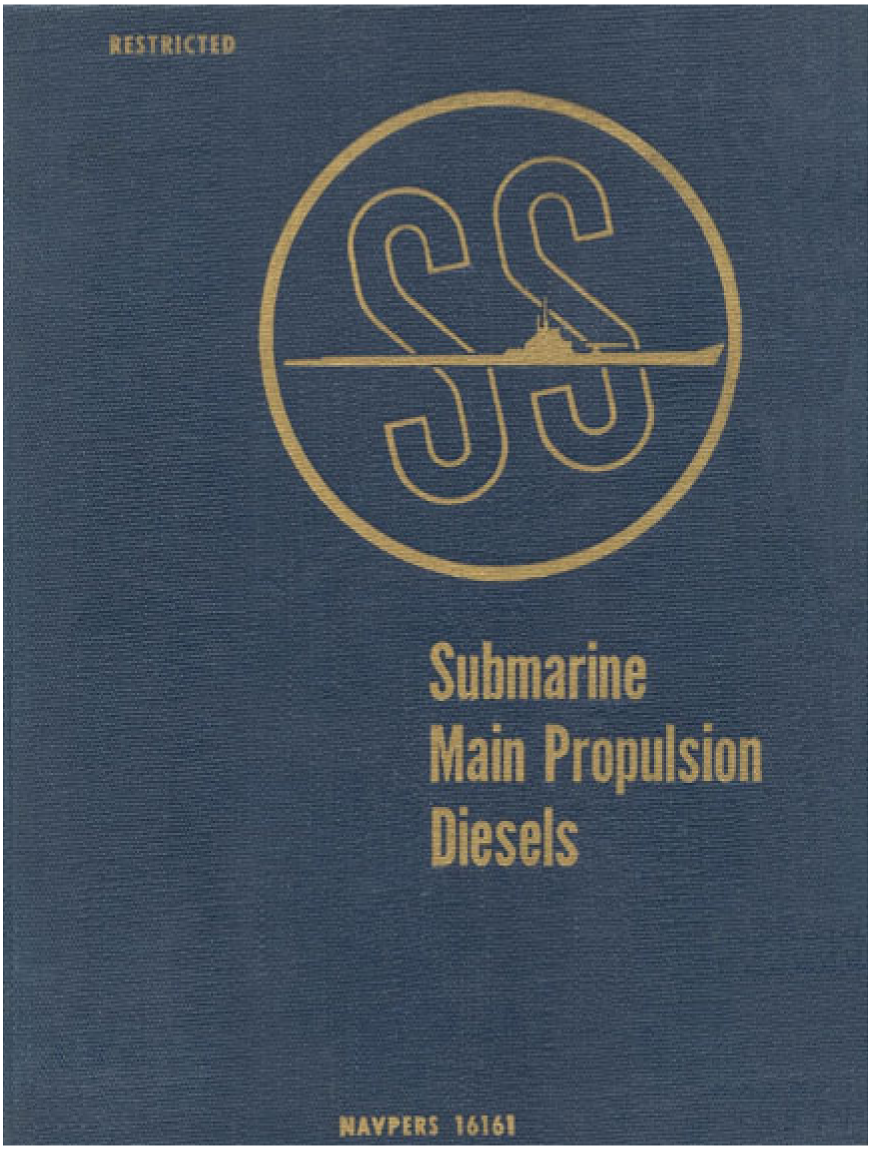
The Fleet Type Submarine Online
Main Propulsion Diesels



Folks,

Main Propulsion Diesels, Navpers 16161, is one of a series of submarine training manuals that was completed just after WW II. The series describes the peak of WW II US submarine technology.

In this online version of the manual we have attempted to keep the flavor of the original layout while taking advantage of the Web's universal accessibility. Different browsers and fonts will cause the text to move, but the text will remain roughly where it is in the original manual. In addition to errors we have attempted to preserve from the original (for example, it was H.L. Hunley, not CS Huntley), this text was captured by optical character recognition. This process creates errors that are compounded while encoding for the Web. Please

report any typos, or particularly annoying layout issues with the Mail Feedback Form for correction.

Our thanks to Shelly Shelstad, creator of History on CD ROM) for permitting us to use images he has scanned, particularly the oversized images that were meticulously pieced together. History on CD ROM sells a very nice CD or thumb drive version of this manual in PDF format for easy access off the web and for a printing. Thanks also to IKON Office Solutions (now Ricoh USA http://www.ricoh-usa.com) for scanning services.

Richard Pekelney
Webmaster

Search MARITIME.ORG

NavPers 16161

Produced for ComSubLant by
Standards and Curriculum Division,
Training, Bureau of Naval Personnel

SUBMARINE

MAIN PROPULSION DIESELS

June 1946

~~RESTRICTED~~

This is one of a series of Submarine
Training Manuals. The series includes:

1.	The Fleet Type Submarine	NavPers 16160
2.	Submarine Main Propulsion Diesels	NavPers 16161
3.	Submarine Electrical Installations	NavPers 16162
4.	Submarine Refrigerating and Air-Conditioning Systems	NavPers 16163
5.	Submarine Distilling Systems	Navpers 16163A
6.	Submarine Air Systems	NavPers

PREFACE

The Submarine School, Submarine Base, New London, Connecticut, and other activities of Submarines, Atlantic Fleet, have collaborated in the preparation of this manual. It is designed as both an instruction and a service manual. Included in the text are detailed descriptions of all engines used on modern fleet type submarines, their various auxiliaries, and the submarine systems used in connection with these installations. In addition, general engineering design principles and diesel engine operating principles are used where necessary to supplement the detailed descriptions.

The various classes of submarines in service today have many different types of engine installations and variations in design and operating procedures. No attempt has been made in this book to cover all such variations, but typical installations have been chosen and described in detail, while major variations have been adequately covered. At all times, however, strict adherence to established operating and maintenance principles has been maintained.

CONTENTS

ILLUSTRATIONS

Fleetsub Home Page   Next chapter

1
DIESEL ENGINE PRINCIPLES

A. DEVELOPMENT

1A1. General. In order that the function and operation of submarine diesel engines may be thoroughly understood, it is necessary to describe briefly the history and development leading to modern design.

It is significant that the diesel engine is an outgrowth of the early struggle to improve the efficiency of existing types of other internal combustion engines. Today's fleet type submarine diesel engines are indirectly the result of widespread experimentation in both the Otto (gasoline) engine field and the more recently developed diesel engine field. Basically, however, the principles of operation have not changed materially since the first practical models of the early designs.

Among the contributors to progress in the development of diesel engines has been the Submarine Service of the United States Navy. Keen interest and untiring effort, not to mention risk in experimentation, testing, and correcting design, have given unparalleled impetus toward improved design.

1A2. History of diesel engine development. The reciprocating internal combustion engine was

The next notable achievement in improving the efficiency of the internal combustion engine was the Hornsby-Ackroyd engine produced in England a short while later. It was among the first early designed engines that used a liquid fuel derived from crude oil. This engine employed the Brayton principle of controlled fuel injection and compressed the air in the cylinder prior to ignition. The compression heat thus generated, plus the use of a hot surface, induced ignition. Since this engine employed hydraulic force to inject the fuel, it is now considered the first example of an engine using mechanical or solid injection.

In 1893, Dr. Rudolf Diesel, a Bavarian scientist, patented a design for an internal combustion engine which was termed a Diesel engine. He considered previous failures and applied himself to designing an engine to operate on an entirely different thermodynamic principle.

Using the mechanics of the 4-stroke cycle, Dr. Diesel proposed that only air be drawn into the cylinder during the suction or intake stroke. The compression stroke was to compress the air in the cylinder to a sufficiently high temperature to induce ignition

introduced in theory as far back as 1862 by Beau de Roches in France. A few years later, Otto, of Germany, made the first practical application of Beau de Roches's theory in an actual working model. Otto's engine was practicable and fairly reliable compared to other earlier attempts. It employed a 4-stroke cycle of operation using gas as a fuel. Thus, the 4-stroke cycle of a gas engine became popularly known as an Otto cycle.

George Brayton, an American, introduced a new principle of fuel injection in 1872. Brayton used an internal combustion gas engine in his experiments. He demonstrated that prolonging the combustion phase of the cycle, by injecting fuel at a controlled rate, produced more power per unit of fuel consumed. However, much of the efficiency gained by this method was lost due to the lack of an adequate method of compressing the fuel mixture prior to ignition.

and combustion without the use of added heat. Like Brayton's engine, this engine was to inject fuel at a controlled rate. It was Dr. Diesel's theory that if the rate of injection were properly controlled during the combustion phase, combustion could be made to occur at a constant temperature. Since fuel would have to be injected against high compression pressures in the cylinder, Dr. Diesel's design called for fuel injection to be accomplished by a blast of highly compressed air. Essentially, this was air injection. Dr. Diesel further theorized that the temperature drop during the expansion phase of the cycle would be efficient to make external cooling of the combustion chamber unnecessary.

A single-cylinder working model was constructed and first experiments were conducted using coal dust as a fuel. All efforts to operate

1

a working model on the cycle proposed by Dr. Diesel resulted in explosions and failure. Further attempts to experiment along this same line were abandoned. Consequently, an engine operating entirely on the theoretical cycle proposed by Dr. Diesel was never produced. This cycle subsequently became known as the diesel cycle.

Many designers realized the value of the practical elements in the cycle of operation outlined by Dr. Diesel. Subsequently,

and, in addition, many of these engines gave off considerable carbon monoxide fumes, creating a menace to personnel.

In the meantime, MAN built and experimented with 2-stroke cycle diesel engines for submarine propulsion. However, insufficient progress had been made in metallurgy to provide metals capable of withstanding the greater heat and stress inherent in engines of this type. MAN then turned its efforts toward production of a 4-stroke cycle

experimenters began to achieve favorable results by eliminating the impractical elements and by altering the cycle of operation. Successful experiments were conducted by the Machinen-fabrik-Augsburg-Nurnberg (commonly called MAN) concern in Germany.

By this time the more volatile petroleum fuels were in common use and diesel engines utilizing liquid fuel were designed. These engines operated on a cycle in which the combustion phase occurred at constant pressure rather than at constant temperature. Experience also disclosed that it was essential to cool the combustion chamber externally. Early diesel engines operating on the constant pressure cycle, were efficient enough to make commercial production feasible.

Progress in diesel engine design has been rapid since the early models were introduced. The impetus of war demands, progress in metallurgy, fabrication, and engineering, and refinements in fuels and lubricants have all served to produce modern, high-speed diesel engines of exceptional efficiency.

1A3. History of submarine engine development. The first United States submarines utilizing internal combustion engines for propulsion were powered by 45-horsepower, 2-cylinder, 4-stroke cycle gasoline engines produced by the Otto Company of Philadelphia. Meanwhile, the English Submarine Service made use of diesel engine capable of developing 1000 hp. While fairly successful, these engines eventually developed structural weaknesses at the crankcase.

By 1914 the MAN 4-stroke cycle diesel had been partially redesigned and strengthened, producing the SV45/42, 1200-hp engine used in the majority of German submarines during World War I. Following World War I, the United States Navy acquired a number of these engines for use in the earlier S-class boats. A copy of this engine was produced by the New York Navy Yard and used in other early S-class submarines.

The Electric Boat Company, which was formerly the Holland Torpedo Boat Company, became licensee in the United States for the MAN Company of Germany. Later, the Electric Boat Company consolidated with the New London Ship and Engine Company. Shortly before World War I, the Electric Boat Company developed the well-known NELSECO engine. During, and subsequent to World War I, a number of United States submarines of the O, R, and S classes were equipped with these NELSECO engines. In fact, the principal installations in United States submarines were 6- and 8-cylinder NELSECO's until about 1934.

Prior to 1930 the engines used in most submarines of all the larger naval powers, with the exception of Great Britain, were 4-stroke cycle diesel engines. The United States Navy, however, experimented with a 2-stroke cycle Busch-Sulzer engine and equipped a number of boats with this type

12- and 16-cylinder gasoline engines in their earlier submarines.

The inherent hazards accompanying the use of such a highly volatile fuel as gasoline were quickly realized. Stowage was a constant problem and handling of the fuel was extremely dangerous. Internal explosions were frequent of engine. Since then, the majority of engines designed for United States submarine use have been of the 2-stroke cycle type.

2

Prior to 1929, all engines in the United States Submarine Service were of the air injection type. Shortly after 1929, mechanical or solid type injection was employed on MAN engines. The advantages to be obtained with this type of injection were immediately apparent. By using solid type injection, the weight of the engines could be considerably reduced. The elimination of the air compressor alone accounted for a saving in weight of approximately 14 percent.

The advantages derived from the use of mechanical injection were numerous and included:

1. simplification of design
2. reduction in length of the engine
3. greatly reduced weight per horsepower
4. reduced fuel consumption
5. improved load balance in the engine
6. far greater reliability
7. less maintenance

The need for more powerful engines became apparent with

1A4. How submarine requirements affect engine design. The fact that submarines are both subsurface and surface vessels places definite restrictions upon size, hull design, and shape. Total weight, too, is a factor having considerable bearing on underwater operations. Hull characteristics restrict engine size and location of the engine compartments. Engine weight must bear a proportionate relationship to the weight and displacement of the vessel as well as to power requirements.

In the first engine-powered submarines, the engines were mechanically connected directly to the propeller shafting. This design, known as direct drive, developed immediate operational problems. The hull characteristics definitely fixed the angle of the propeller shafting. This restriction also determined engine position and location. Also, the most efficient propeller speeds did not correspond with the most efficient engine speeds. In direct drive installations, critical speeds (or synchronous torsional vibrations) which were inherent in the early

the development of the fleet type submarine. The three engines that seemed to fulfill submarine requirements were the Winton V-type, now known as the General Motors engine; the Fairbanks-Morse opposed piston type; and the Hooven-Owen-Rentschler double-acting type engine. Of these, the HOR was later removed from submarines in favor of the General Motors and Fairbanks-Morse engines which are now the two standard submarine engines.

At the present time, the General Motors Corporation manufactures 16-cylinder, single-acting engines rated at 1600 brake horsepower (bhp) for main engine installations, and 8-cylinder engines for auxiliary installations. Fairbanks Morse and Company manufactures 9- and 10-cylinder, opposed piston engines rated at 1600 bhp for main engine installations, and 7-cylinder, opposed piston engines for auxiliary installations. These engines have proved most efficient. They weigh as little as 15 to 20 pounds per bhp including auxiliary equipment. Standardizing on only two designs has also made it possible to mass produce engines with a minimum amount of delay and difficulty.

model engines, were transferred through the direct drive into shafting and propellers. At times, the exact cruising speed desired could not be obtained, as it was necessary to pass the engines through critical speeds in the desired operating range as rapidly as possible. Two major problems were brought to the foreground by these early models:

1. How to power the propellers and yet separate engines and propeller shafting so that no mechanical unity existed.

2. How to design a drive in which different and varied rotative speeds could be selected for both engines and propellers.

Various types and combinations of drives were designed and tested. Over a period of time it became apparent that the electric drive installations (commonly referred to as diesel-electric drive) were the practical solution. This type of design solved both of the major problems. The engines were coupled only to the generators that supplied power to the electric motors. The propeller shafting was driven by the motors through reduction gears or directly

3

---

by slow-speed electric motors. The only connections between engine power and propeller shafting were electrical. Hence, vibrations developed by the engines could not be conducted

1. The engine should furnish maximum amount of power with minimum weight and space requirement.

to the propeller shafting and propellers, and the various stresses encountered by the propellers could not be transmitted directly to the engines as was the case with mechanical couplings.

In addition, the rotative speed of the engine was no longer limited by the rpm of the propellers. Consequently, the engines could be designed for any desired speed within a selected range. Likewise, the propellers could be operated independently of engine speed within the speed limits of their design. The diesel-electric drive gave greater latitude to designers with respect to operating speed, size, and location of engines. It also gave the boat designers greater freedom in placement of engine compartments.

There are eight major requirements that a submarine diesel engine should fulfill:

2. The engine should possess the ability to develop occasionally more than full load rating.

3. The engine should have the ability to run continuously at slightly less than full load rating.

4. The engine should operate with small fuel consumption per unit of horsepower.

5. The engine should have a small lubricating oil consumption.

6. All wearing parts should be readily accessible for quick replacement.

7. There should be perfect balance with respect to primary and secondary forces and couples.

8. Major critical speeds within the operating ranges of the engine should be eliminated.

B. PRINCIPLES Of DESIGN AND OPERATION

1B1. Reciprocating internal combustion engines. An engine that converts heat energy into work by burning fuel in a confined chamber is called an internal combustion engine. Such an engine employing back-and-forth motion of the pistons is called a reciprocating type internal combustion engine. The diesel engine and the gasoline engine are the most familiar examples of reciprocating internal combustion engines.

The basic principle of operation of an internal combustion engine

charge of fuel and air is admitted, and the process is repeated. The above sequence of events is called a cycle of operation.

1B2. Cycles of operation. The word cycle enters into the description of the operation of any internal combustion engine. As applied to internal combustion engines, it may be defined as the complete sequence of events that occur in the cylinder of an engine for each power stroke or impulse delivered to the crankshaft. Those events always occur in the same

is relatively simple. The space in the cylinder in which the fuel is burned is called the combustion chamber. Fuel and air are admitted to the combustion chamber and ignited. The resulting combustion increases the temperature within the combustion chamber. Gases, released by combustion, plus the increase in temperature, raise the pressure which acts on the piston crown, forcing the piston to move. Movement of the piston is transmitted through other parts to the crankshaft whose rotary motion is utilized for work. The expended gases are ejected from the cylinder, a new order each time the cycle is repeated.

Each cycle of operation is closely related to piston position and movement in the cylinder. Regardless of the number of piston strokes involved in a cycle, there are four definite events or phases that must occur in the cylinders.

1. Either air or a mixture of air and fuel must be taken into the cylinder and compressed.

2. The fuel and air mixture must be ignited, or fuel must be injected into the hot compressed air to cause ignition.

3. The heat and expansion of gases resulting from combustion must perform work on the piston to produce motion.

4. The residual or exhaust gases must be discharged from the cylinder when expansion work is completed.

The cycles of operation in each type of internal combustion engine are characterized both by the mechanics of operation and the thermodynamic processes. The three most commonly known cycles are the Otto cycle, the diesel cycle, and the modified diesel cycle.

1B3 Thermodynamics. To explain thermodynamics as used in an engineering sense, it is first necessary to define the term and the related terms used with it.

Matter is anything having weight and occupying space. Solids, liquids, and gases are matter.

A molecule is the smallest division of a given matter, which, when taken alone, still retains all the properties and characteristics of the matter.

Heat is a form of energy caused by the molecular activity of a substance. Increasing the velocity of molecular activity in a substance increases the amount of heat the substance contains. Decreasing the velocity of molecular activity in a substance decreases the amount of heat the substance contains.

Temperature is a measure of the intensity of heat and is recorded in degrees by a thermometer. The two temperature scales most commonly used are the Fahrenheit and centigrade scales.

Thermodynamics is the science that deals with the transformation of energy from one form to another. A basic law of thermodynamics is that energy can neither be created nor destroyed but may be changed from one form to another. In diesel engineering, we are concerned primarily with the means by which heat energy is transformed into mechanical energy or work.

Force is that push or pull which tends to give motion to a body at rest. A unit of force is the pound.

Pressure is force per unit area acting against a body. It is generally expressed in pounds per square inch (psi).

Work is the movement of force through a certain distance. It is measured by multiplying force by distance. The product is usually expressed in foot-pounds.

Power is the rate of doing work, or the amount of work done in unit time. The unit of power used by engineers is the horse power (hp). One horsepower is equivalent to 33,000 foot-pounds of work per minute or 33,000/60 = 550 foot-pounds per second.

Energy is the ability to perform work. Energy is of two types: kinetic, which is energy in motion, and potential, which is energy stored up.

Volume may be described as the amount of space displaced by a quantity of matter.

1B4. The mechanical equivalent of heat energy. The function of an internal combustion engine is to transform heat energy into mechanical energy. Recalling the basic law of thermodynamics we know that energy cannot be destroyed. It is possible to convert mechanical energy to heat completely, and by delicate physical experiments it has been found that for every 778 foot-pounds of mechanical energy so converted, one Btu of heat will be obtained. Because of fundamental limitations, it is usually not possible to convert heat completely to work, but for every Btu that is converted, 778 foot-pounds will be realized. This important constant is known as the mechanical equivalent of heat.

1B5. Relationship of pressure, temperature, and volume. Figure 1-1A illustrates a simple cylinder with a reciprocating piston. A dial pressure gage at the top of the cylinder registers pressure inside the cylinder. Temperature inside the cylinder is recorded by a thermometer. The thermometer at the side registers room temperature. The piston is at outer dead center in its stroke. At this stage, the pressure inside

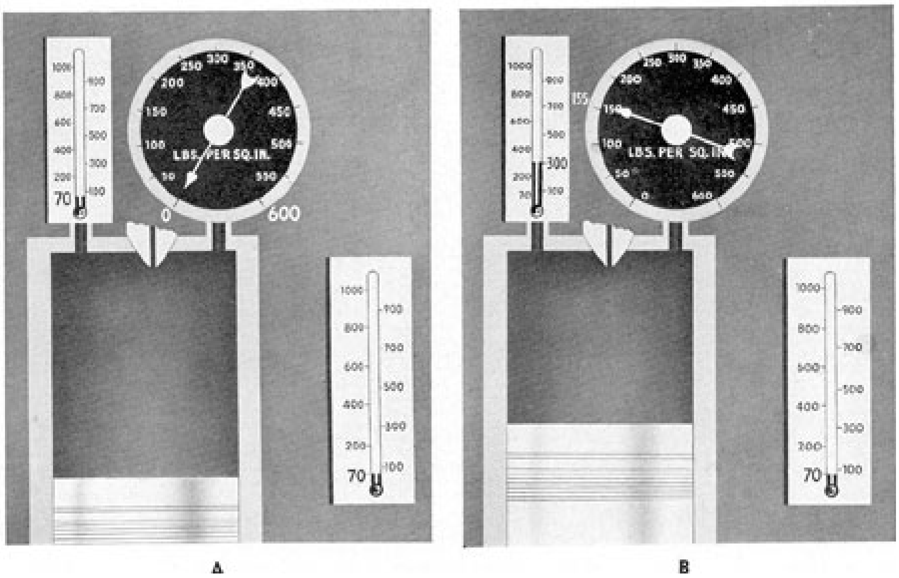


Figure 1-1. Pressure, temperature, and volume relationship in a cylinder.

the cylinder is the same as atmospheric pressure outside, and the dial of the pressure gage registers 0. Also, the temperature inside the cylinder is the same as room temperature, or approximately 70 degrees F.

In Figure 1-1B, force has been applied to the piston, moving it about a third of the distance of its compression stroke. Air trapped in the cylinder is compressed. As the volume of this air is decreased, the pressure is increased to about 155 psi. The temperature rises from 70 degrees F to about 300 degrees, indicating that heat has been added to the air in the cylinder. This shows that mechanical energy, in the form of force supplied to the piston, has been transformed into heat energy in the compressed air.

In Figure 1-1C, more force has been applied to the piston, raising the pressure in the cylinder to about 300 psi, and the temperature to nearly 700 degrees F.

Figure 1-1D shows the final stage of the compression stroke conditions found in the compression stroke of a modern submarine diesel engine. The temperature of the compressed air within the cylinder has been raised to a sufficient degree to cause automatic ignition on the injection of fuel oil into the cylinder.

Thus, in summation, we see that during a cycle of operation, volume is constantly changing due to piston travel. As the piston travels toward the inner dead center during the compression stroke, the air in the cylinder is reduced in volume. Physically, this amounts to reducing the space occupied by the molecules of air. Thus, the pressure of the air working against the piston crown and walls of the cylinder is increased and the temperature rises as a result of the increased molecular activity. As the piston nears inner dead center, the volume is reduced rapidly and the temperature increases to a point sufficient to support the automatic ignition of any fuel injected.

Combustion changes the injected fuel to gases. After combustion, the liberation of the gases with a very slight increase in volume

as the piston arrives at inner dead center. Pressure is in the neighborhood of 470 psi and the temperature is about 1000 degrees F. This illustration closely approximates the causes a sharp increase in pressure and

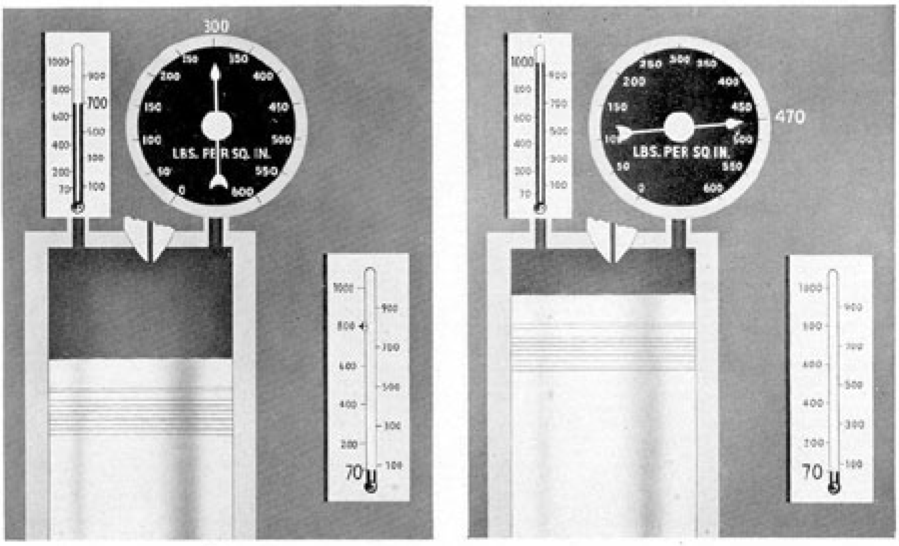


Figure 1-1. Pressure, temperature, and volume relationship in a cylinder.

temperature. During the power stroke, volume increases rapidly, and toward the end of the stroke, pressure and temperature decrease rapidly.

1B6. Pressure-volume diagrams. Various methods and devices are used for measuring and recording the pressures at various piston positions during a cycle of operation in an engine cylinder. The result may be graphically illustrated by a diagram such as that shown in Figure 1-2. Such diagrams are known as pressure-volume diagrams. In practice, they are referred to as indicator cards.

Pressure-volume diagrams give the relationship between pressures and piston positions, and may be used to measure the work done in the cylinder. Also, if

1B7. Pressure-volume diagrams for the Otto cycle, diesel cycle, and modified diesel cycle. Figure 1-2 shows typical pressure-volume diagrams for the three types of engine cycles. Each pressure-volume diagram is a graphic representation of cylinder pressure as related to cylinder volume. In the diagrams the ordinate represents pressure and the abscissa represents volume. In actual practice, when an indicator card is taken on an engine, the vertical plane is calibrated in pressure units and the volume plane is calibrated in inches. The volume ordinate of the diagram then shows the length of stroke of the piston which is proportional to the volume.

Letters are located on each of the figures in the diagrams. The distance between two adjacent

the speed of the engine and the time involved in completing one cycle are known, the indicated horsepower may be computed by taking pressure-volume diagrams on each cylinder and converting the foot-pounds per unit of time into horsepower. This method of determining horsepower, however, is not practicable on modern fleet type submarine engines.

letters on the figures is representative of a phase of the cycle. Comparing the diagrams provides a visible means of comparing the variation in the phases between the three cycles.

1B8. The Otto cycle. The Otto cycle (Figure 1-2) is more commonly known as the constant volume cycle and its principles form

7

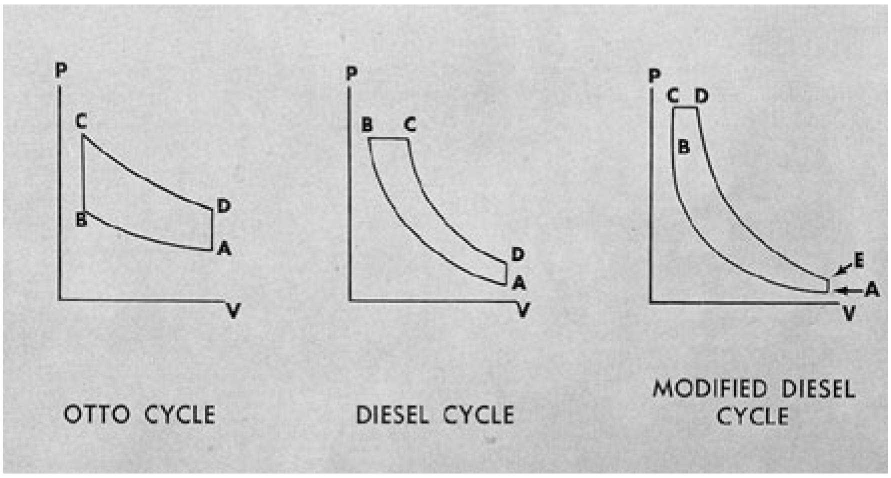


Figure 1-2. Pressure-volume diagrams.

the basis for all modern automobile gasoline engine designs. In this cycle, combustion is timed to occur theoretically just as the piston arrives at top dead center. Ignition is accomplished by a spark, and, due to the volatility of the fuel-air mixture, combustion practically amounts to an explosion. Combustion is completed with virtually no piston travel and hence, little, if any, change in volume of the gas in the combustion chamber. This gives rise to the description constant-volume cycle. During combustion there is a quick rise of the temperature in the cylinder, immediately followed by a pressure rise which

oils will ignite automatically with sufficient air at a temperature of about 480 degrees F, ignition occurs as soon as the fuel oil spray reaches the hot air. This is called compression ignition.

This combustion process (or burning of the fuel and compressed air) is a relatively slow process compared with the quick, explosion type combustion process of the Otto cycle. The fuel spray penetrates the compressed air, some of the fuel ignites, then the rest of the fuel charge burns. In the true diesel cycle, the expansion of gases keeps pace with the change in volume occasioned by piston travel during the combustion phase. Thus

performs the work during the power stroke.

The Otto cycle may be defined as a cycle in which combustion induced by spark ignition theoretically occurs at constant volume.

1B9. The diesel cycle. In the true diesel cycle, only air is compressed in the cylinder prior to ignition. This normally produces a final compression pressure of about 500 psi. At such a pressure the temperature of the compressed air may range from 900 degrees to 1050 degrees F. Since most fuel

combustion is said to occur at constant pressure.

The diesel cycle may be defined as a cycle in which combustion induced by compression ignition theoretically occurs at a constant pressure.

1B10. Modified diesel cycle. We have previously described the Otto cycle as one in which combustion occurs theoretically at constant volume, and the diesel cycle as one in which

8

combustion occurs theoretically at constant pressure. In actual operation, a gasoline engine does not follow the true Otto cycle, nor does the diesel engine follow the true diesel cycle. In fact, the operation of a medium- or high-speed diesel engine follows the modified diesel cycle (Figure 1-2). This cycle involves phases of both the Otto cycle and the diesel cycle in that the combustion phase takes place at both constant volume and constant pressure.

The modified diesel cycle, as applied to diesel engines, may be defined as a cycle of operation in which the combustion phase, induced by compression ignition, begins on a constant-volume basis and ends on a constant pressure basis.

All submarine main and auxiliary engines used today employ the

After the fuel is injected into the cylinder, combustion converts it into gases. This conversion is a thermodynamic change. A thermodynamic change during which the temperature remains constant is called an isothermal process. A thermodynamic change during which the temperature may vary but during which heat is neither received nor rejected is called an adiabatic process.

In a strict sense the thermodynamic cycles outlined below are not true thermodynamic cycles. In a true cycle the process is reversible. The working substance is heated, does work, is cooled, and is heated again. In the cycle of an actual engine, the residue of the combustion process is exhausted at the end of the expansion stroke and a new charge is taken into the cylinder for the next cycle of events. However, the true thermodynamic

modified diesel cycle. The fundamental differences between the Otto and the modified diesel cycles are:

1. The methods of mixing fuel and air. This is accomplished before and during compression in the Otto cycle and usually near the end of the compression phase in the modified diesel cycle.

2. The methods of ignition. Spark ignition is used in the Otto cycle and compression ignition is used in the modified diesel cycle.

The term diesel cycle has become popularly associated with all compression-ignition or diesel engines. In actual practice, this is a misnomer when applied to modern, medium-speed or high-speed diesel engines, because practically all diesel or compression-ignition engines in this category operate on the modified diesel cycle.

1B11. Thermodynamics of the Otto cycle, every diesel cycle, and modified diesel cycle. In every thermodynamic cycle there must be a working substance. With internal combustion engines, some form of substance must undergo a change in the cylinder in order to convert heat energy into mechanical energy. The working substance in the cylinder of a compression-ignition engine is fuel oil. cycle is useful for studying the thermodynamic processes in actual engine operation.

a. The Otto cycle. This is the thermodynamic cycle used as a basis for the operation of all modern gasoline engines. The cycle (Figure 1-2) consists of the adiabatic compression of the charge in the cylinder along the line AB, the constant-volume combustion and heating of the charge from B to C, the adiabatic expansion of the gases from C to D, and the constant-volume rejection of gases from the cylinder along DA.

b. The diesel cycle. In the original diesel cycle proposed by Dr. Diesel, the combustion phase of the thermodynamic cycle was to be a constant-temperature or isothermal process. However, no engine was ever operated on this cycle. As a result of his experimentation, however, a constant-pressure thermodynamic cycle was developed. All early type, slow-speed diesel engines approximated this cycle, although it is in little use today.

In this cycle (Figure 1-2), adiabatic compression occurred along AB, to provide the temperature necessary for the ignition of the fuel. Fuel injection and combustion were so controlled as to give constant-pressure combustion

9

---

along BC. This was followed by adiabatic expansion from C to D. Rejection of the gases from the

2. With the intake valve closed, the piston makes an upward stroke, compressing the air. Pressure is generally around 500 psi with

cylinder was constant volume from D to A.

c. The modified diesel cycle. This is the cycle (Figure 1-2) used in all fleet type submarine diesel engines and in practically all modern diesel engines. In this thermodynamic cycle, compression is adiabatic from A to B. Combustion is partly constant volume from B to C and partly constant pressure from C to D. Expansion is adiabatic from D to E. Rejection of gases from the cylinder is constant volume along EA.

1B12. Thermal efficiency. The thermal efficiency of an internal combustion engine may be considered the percentage of efficiency, in converting the total potential heat energy available in the fuel into mechanical energy. We have already stated that the mechanical equivalent of heat energy is 778 foot-pounds for one Btu of heat. By this equation, it is a simple matter to figure how much work should be delivered on an ideal basis from a given quantity of fuel. An engine operating on this- basis would be 100 percent efficient. No internal combustion engine, however, is 100 percent efficient, because heat losses, conducted through the cooling and exhaust systems, and friction losses make the thermal efficiency of any internal combustion engine relatively low.

1B13. The 4-stroke diesel cycle. In the 4-stroke diesel cycle, the piston makes four strokes to complete the cycle. There is one power stroke or power impulse for every four piston strokes, or resultant temperatures as high as 900 degrees to 1050 degrees F, depending on the design of the engine. At about the end of this stroke, the fuel is injected into the hot compressed air, and ignition and combustion occur over a relatively short period of piston travel.

3. The expansion of combustion gases forces the piston downward through one stroke. This is called the power stroke. As the piston nears the end of this stroke, the exhaust valve opens, permitting some of the burned gases to escape.

4. The piston makes another upward stroke in which the remaining exhaust gases are forced out of the cylinder. This completes the cycle.

1B14. The 2-stroke diesel cycle. In this cycle (Figure 1-4) the piston makes two strokes to complete the cycle. There is one power stroke for every two piston strokes or for each revolution of the crankshaft. An engine employing this cycle requires a scavenging air blower to assist in clearing the exhaust gases from the cylinder, to replenish the cylinder with the necessary volume of fresh air, and to make possible a slight supercharging effect.

Figure, 1-4 shows the two strokes and the sequence of events that occur in the 2-stroke diesel cycle as follows:

1. Start of compression. The piston has just passed bottom dead center, the cylinder is charged with fresh air, and both the intake ports and the exhaust valve are closed.

two complete revolutions of the crankshaft.

Figure 1-3 shows the four strokes and the sequence of events that occur in the 4-stroke diesel cycle.

1. The intake valve opens and a supply of fresh air is drawn into the cylinder while the piston makes a downward stroke. The fresh air is trapped and compressed in the cylinder.

2. Injection. At about the end of the compression stroke, the fuel is injected and combustion occurs.

3. Expansion. Expansion of gases from combustion forces the piston downward through one stroke. As the piston nears the end of this stroke, the exhaust valve is opened slightly in

advance of the uncovering of the intake ports. This permits some of the burned gases to escape.

4. Exhaust. As the intake ports are uncovered, the scavenging air which is under pressure, rushes into the cylinder. This drives out the remaining exhaust gases and completes the cycle.

C. DIESEL ENGINE TYPES

1C1. Single-acting diesel engine. Both the 4-stroke and the 2-stroke cycle diesel engines illustrated and described in the previous section were of the single-acting type (Figure 1-5). In all single-acting engines the pistons used are usually of the trunk type, that is, pistons whose length is greater than their diameter. One end of the trunk type piston is closed; this end is called the crown. The opposite or skirt end of the piston is open. The connecting rod extends through the open end of the piston and is attached to the piston by means of the piston pin.

The term single-acting is used to describe these engines because the pressure of the gases of combustion acts only on one end of the piston. The piston rod extends through the cylinder head of the lower combustion chamber and passes through a stuffing box to prevent leakage of pressure. The piston rod is attached to a crosshead, and the connecting rod is attached to the crosshead so that it may turn freely on the crosshead pin. The crosshead has a flat bearing surface that moves up and down on a crosshead guide to steady the piston rod and piston and prevent uneven wear.

Combustion occurs in the upper combustion chamber, and the pressure of the gases of combustion is applied to the top end of the piston during the downward stroke. At the completion of this stroke, combustion occurs in the bottom combustion chamber and

side (the crown) of the pistons. In the 4-stroke cycle, single-acting engines, the power stroke occurs only once in every two revolutions of the crankshaft. In the 2-stroke cycle, single-acting engines, the power stroke occurs once in every revolution of the crankshaft. All of the main and auxiliary diesel engines currently installed in fleet type submarines are of the single-acting type.

1C2. Double-acting diesel engine. A considerable number of double-acting diesel engines (Figure 1-6), namely the HOR and MAN engines, were used in installations for fleet type submarines until recent years. Lately, however, most of these double-acting engines have been removed and replaced with 2-stroke cycle, single-acting engines. While double-acting engines have no place in current installations, it is well for the student to be familiar with their general design and operation.

In double-acting diesel engines, the piston proper is usually shorter and is described as the crosshead type. The piston is closed at both ends and has a rigid piston rod extending from the lower end. Both ends of the cylinder are closed to form a combustion chamber at each expansion pressure is applied to the bottom end of the piston during the upward stroke. The downward power stroke serves as the compression stroke for the lower combustion chamber and the upward power stroke serves as the compression stroke for the top combustion chamber. Thus the power strokes are double that of a single acting engine and the engine is referred to as a double-acting type.

The 2-stroke cycle, double-acting engine has a distinct advantage in power output compared with the single-acting type. With twice as many power strokes as a comparable single acting engine and, with other conditions being equal, it develops practically twice as much power per cylinder. In addition, the operation is smoother due to the fact that the expansion stroke in one combustion chamber of the cylinder is balanced or cushioned by the compression stroke in the opposite combustion chamber.

There are two principal difficulties encountered in adapting double-acting engines to submarine use. First, the crosshead type of piston construction requires considerably more length than that of single-acting engine types. As a

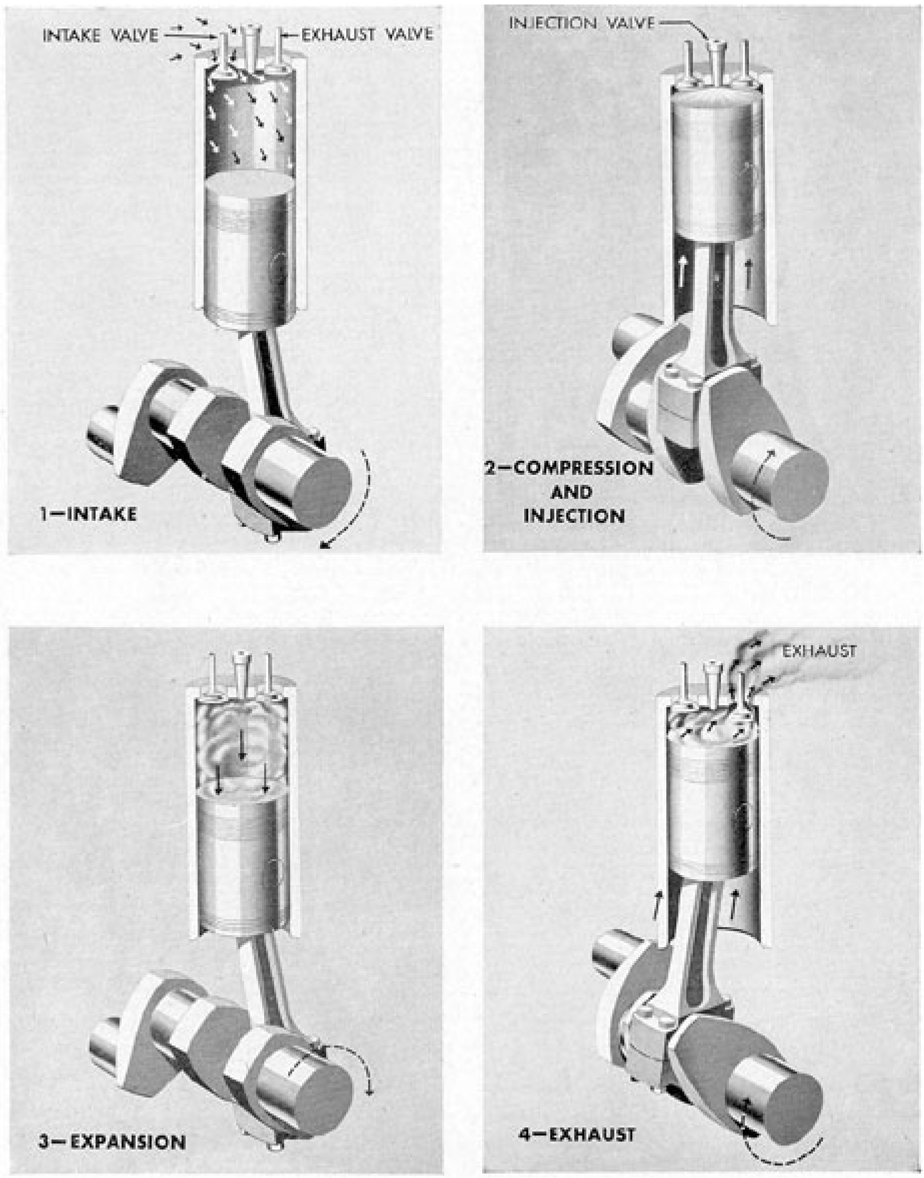


Figure 1-3. The 4-stroke diesel cycle.

12

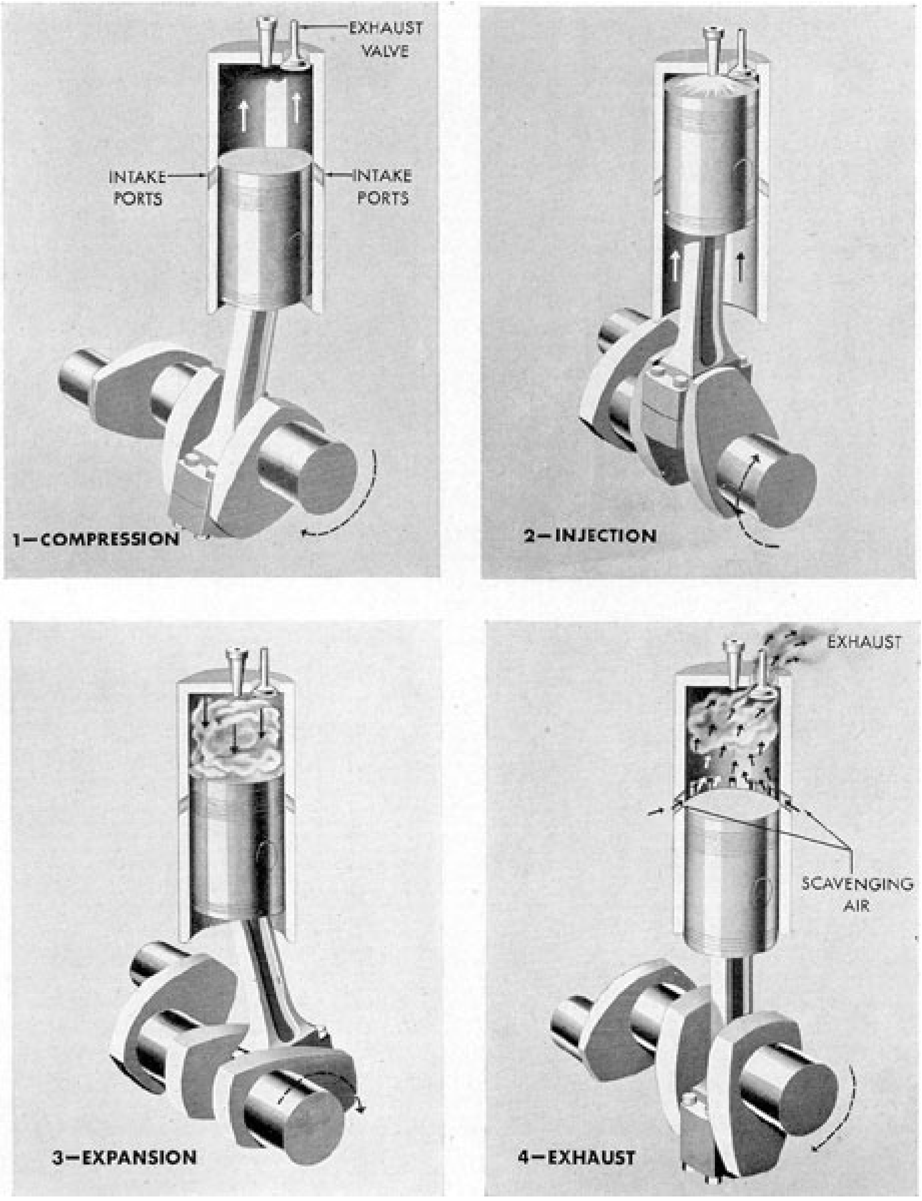


Figure 1-4. The 2-stroke diesel cycle.

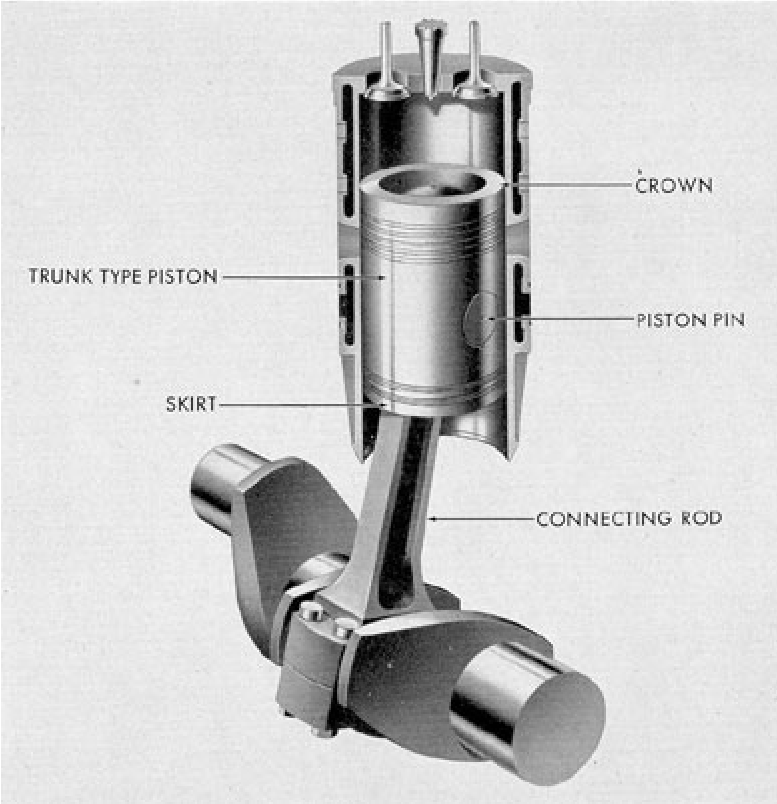


Figure 1-5. Single-acting diesel principle.

consequence, the engines must be built too high and bulky for practical use in the confined spaces available aboard submarines. Secondly, many difficulties are encountered in effecting a tight seal where the piston rod passes through the stuffing box.

1C3. Opposed piston engine. The opposed piston engine (Figure 1-7) is designed with two pistons in each cylinder. The pistons are arranged in opposed positions in the cylinder. Piston action is so timed that at one point of travel the two pistons come into close proximity to each other near the, center of the cylinder.

As the pistons travel together they compress air between them. The space between the two pistons thus becomes the combustion chamber. The point at which the two pistons come into closest proximity is called combustion dead center. Just prior to combustion dead center, fuel is injected and the resultant expansion caused by combustion drives the pistons apart.

The scavenging air ports are located in the cylinder walls at the top of the cylinder and are opened and closed by the movement of the upper piston. The exhaust ports are located near the bottom of the cylinder and are opened and closed by the movement of the lower piston.

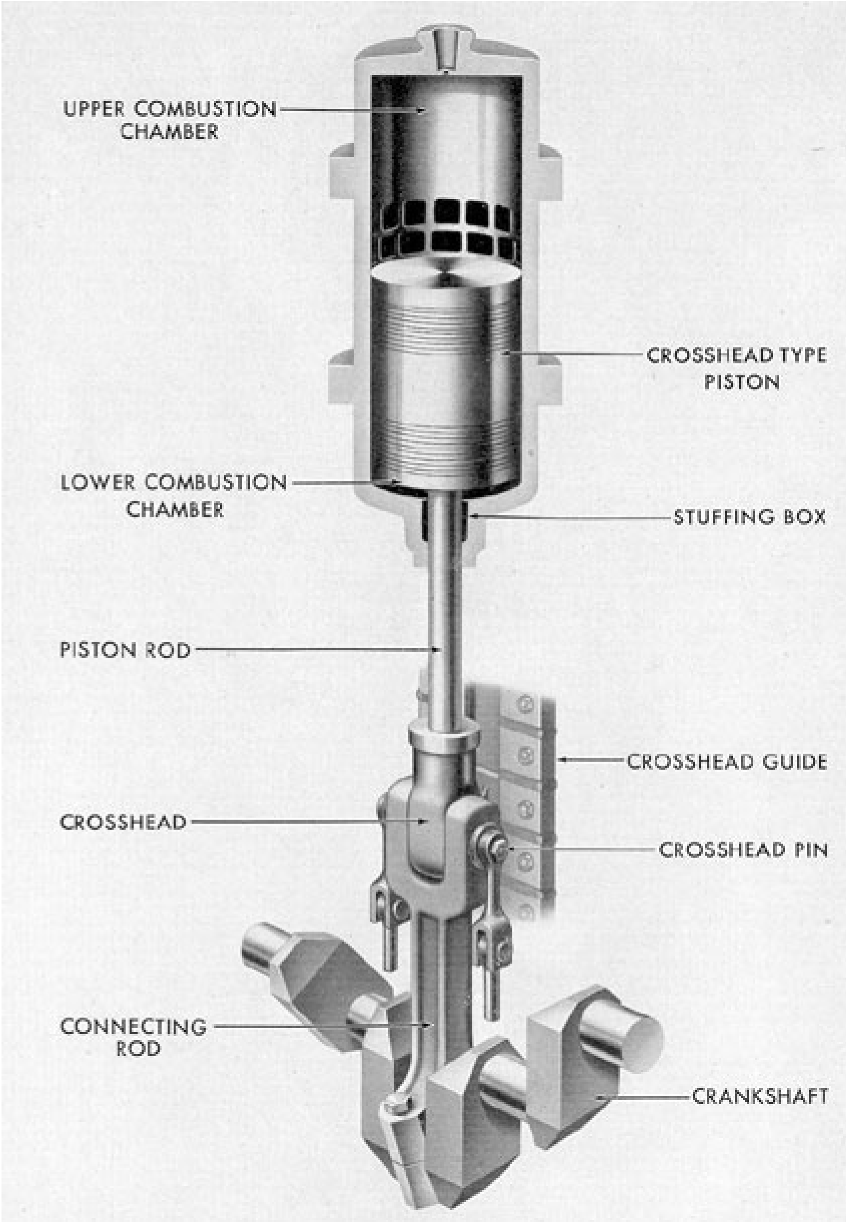


Figure 1-6. Double-acting diesel principle.

15

---

1. Both pistons are on the return travel from outer dead center, the upper piston has covered the scavenging air ports, the lower piston has covered the exhaust ports, and compression has begun.

2. Just as both pistons approach combustion dead center, fuel is injected.

3. Injection has been completed, expansion has begun, and both pistons are moving toward outer dead center.

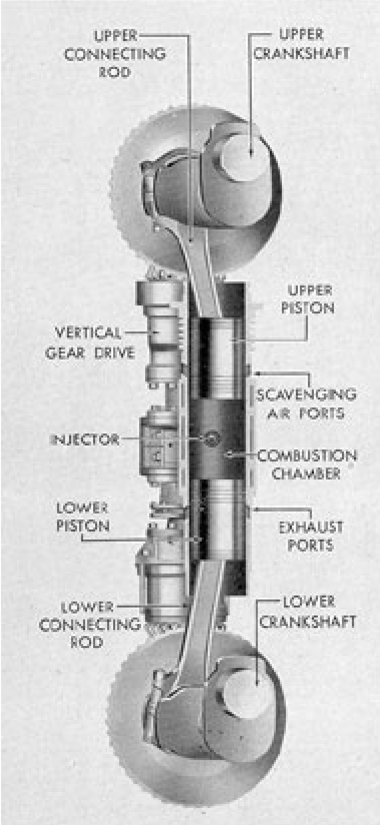


Figure 1-7. Opposed piston principle.

All the upper pistons are connected by connecting rods to the upper crankshaft. All the lower pistons are connected by connecting rods to the lower crankshaft. In Fairbanks Morse, opposed piston, submarine engines, the upper and lower crankshafts are connected by a vertical gear drive. The power from the upper crankshaft not used to drive auxiliaries is transmitted through this drive to the lower crankshaft and ultimately to the engine final drive.

Figure 1-8 shows the various phases in a 2-stroke cycle of operation in an opposed piston engine.

4. Expansion of gases from combustion drives the pistons apart, causing the crankshafts to turn. This is the power stroke of the cycle.

5. As the pistons approach outer dead center, the lower piston uncovers the exhaust ports and most of the expanded gases escape. Just before reaching outer dead center, the upper piston uncovers the scavenging air ports and scavenging air rushes into the cylinder, cleaning out the remaining exhaust gases.

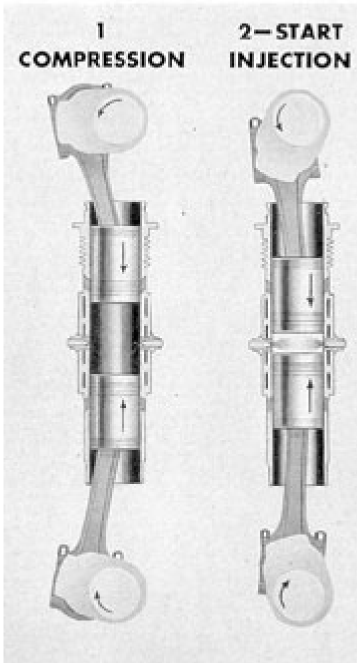


Figure 1-8. Opposed piston cycle.

16

6. The lower piston has covered the exhaust ports and

With the 12-degree lower crankshaft lead, the lower piston

scavenging air supercharges the cylinder until the upper piston covers the scavenging air ports.

Figure 1-13 shows how the lower crankshaft leads the upper crankshaft by 12 degrees in the Fairbanks-Morse submarine diesel engine. This lower crankshaft lead has a definite effect both upon scavenging and power output.

Since the lower crankshaft leads the upper, the exhaust ports at the lower end of the cylinder are covered slightly before upper piston travel covers the intake ports. Thus, for a brief interval, the exhaust ports are closed while the intake parts are open. By the time the intake port is covered, the cylinder has been charged with fresh air well above atmospheric pressure. Thus, through the lower crankshaft lead and scavenging action, a supercharging effect is achieved in this engine.

has advanced the crankshaft through a 12-degree arc of travel in the expansion phase of the cycle by the time the upper piston has reached inner dead center. This causes the lower piston to receive, at full engine load, the greater part of the expansion work, with the result that about 70 percent of the total power is delivered by the lower crankshaft.

For submarine use, the opposed piston engine has three distinct advantages.

1. It has higher thermal efficiency than engines of comparable ratings.

2. It eliminates the necessity of cylinder heads and intricate valve mechanisms with their cooling and lubricating problems.

3. There are fewer moving parts.

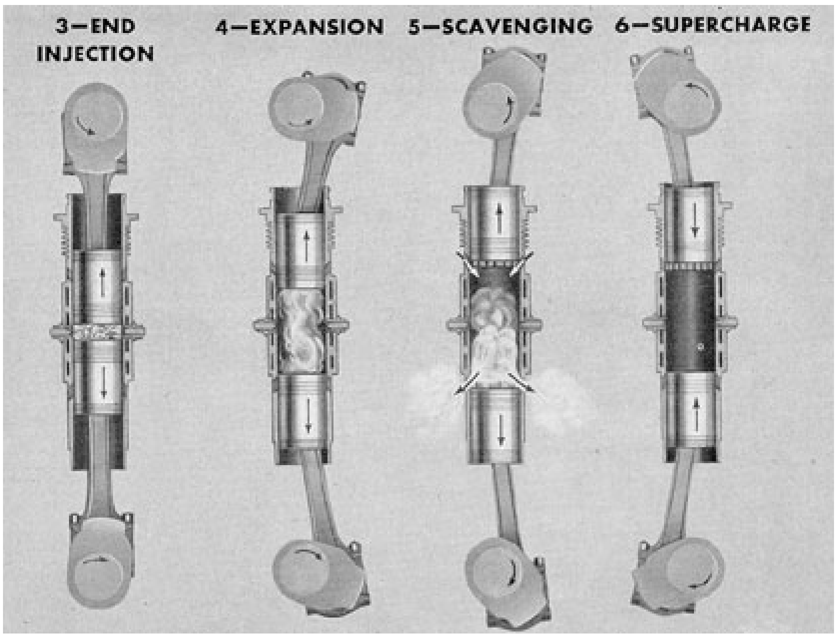


Figure 1-8. Opposed piston cycle.

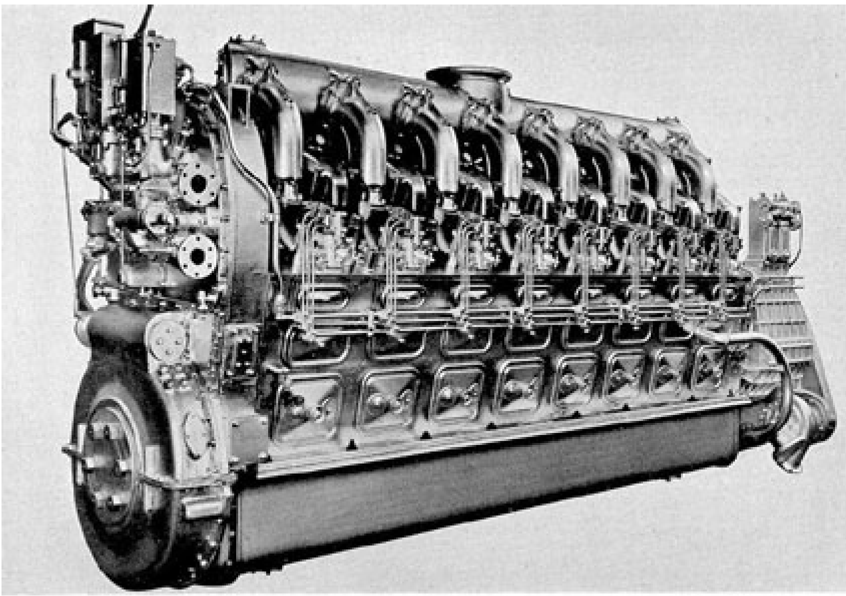


Figure 1-9. GM 16-278A, outboard side, control end, right-hand engine.

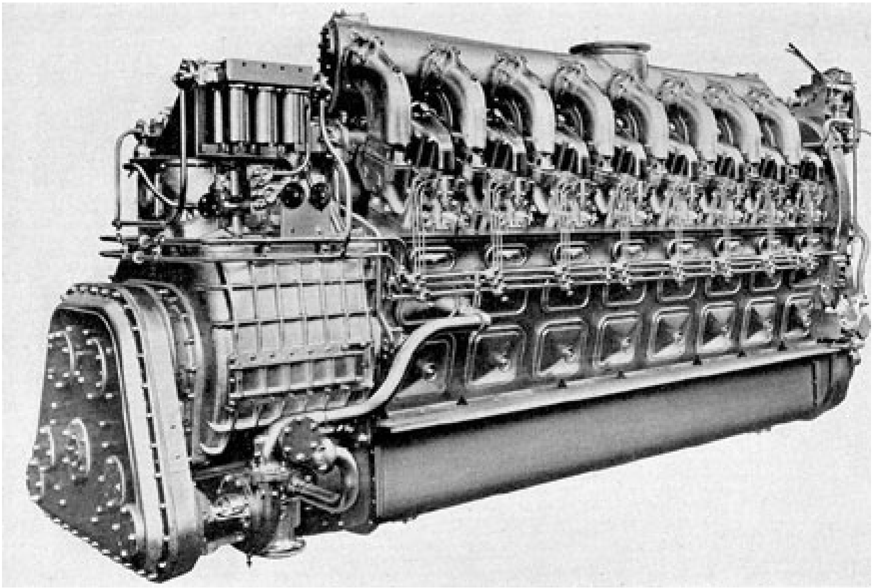


Figure 1-10. GM 16-278A, inboard side, blower end, right-hand engine.

18

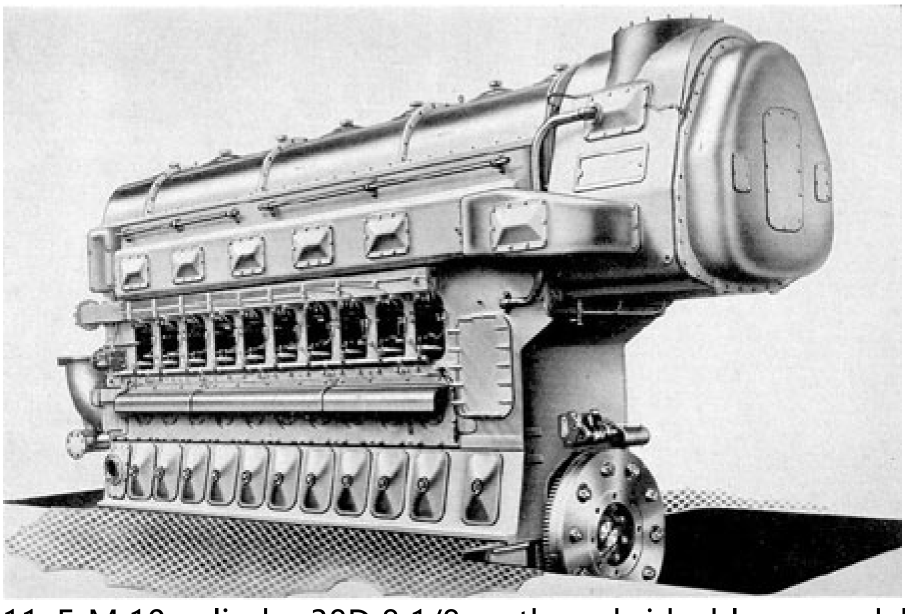


Figure 1-11. F-M 10-cylinder 38D 8 1/8, outboard side, blower end, left-hand

engine.

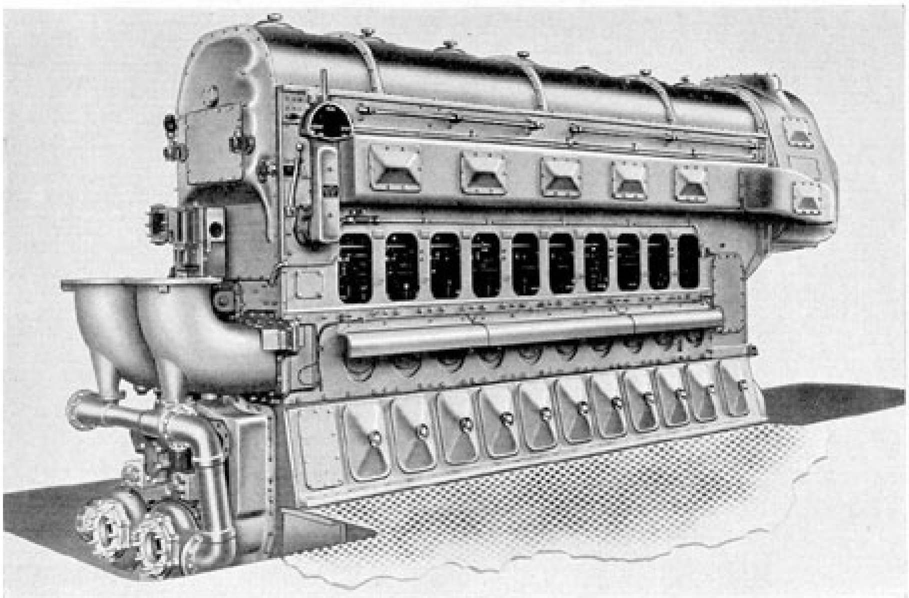


Figure 1-12. F-M 10-cylinder 38D 8 1/8, Inboard side, control end, right-hand engine.

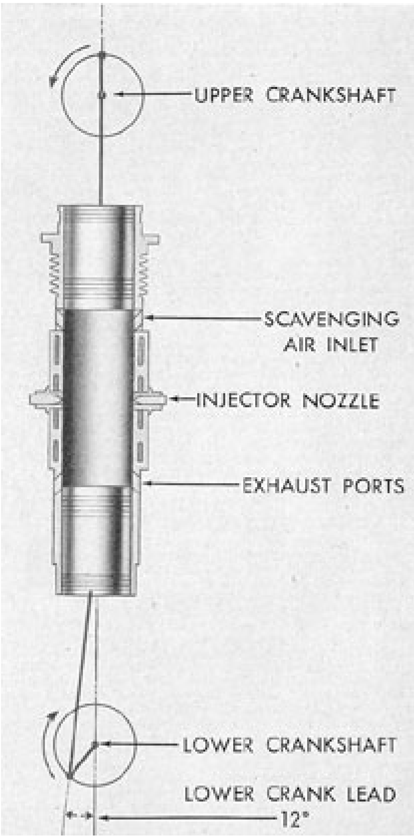


Figure 1-13. Lower crank lead.

1C4. Modern fleet type submarine diesel engines. Modern diesel engines currently used in fleet type submarine installations vary in design but all are of the 2-stroke cycle type. Following is a list of engines normally found on fleet type submarines:

a. Main engines.

1. General Motors V-16 type. There are two engine designs in this category, the 16-278A and 16-248. Each engine has two banks of 8 cylinders, each arranged in a V-design with 40 degrees between banks. Each engine is rated at 1600 bhp at 750 rpm. Both engines are equipped with mechanical or solid type injection and have a uniflow valve and port system of scavenging.

2. Fairbanks-Morse opposed piston type, Model 38D 8 1/8. This model number includes two engines, one a 10-cylinder and the other a 9-cylinder engine. Both engines are rated at 1600 bhp at

720 rpm. Both engines are equipped with mechanical or solid type injection and have a uniflow port system of scavenging.

b. Auxiliary engines.

1. General Motors, Model 8-268. This engine is an 8-cylinder, in-line type. When operated in a generator set at 1200 rpm, it has a power output of 300 kilowatts. This engine is equipped with mechanical or solid type injection and has a uniflow valve and port system of scavenging.

2. Fairbanks-Morse opposed piston type, Model 38E 5 1/4. This is a 7-cylinder, opposed piston type engine. When operated in a generator set at 1200 rpm, it has a power output of 300 kilowatts. This engine is equipped with mechanical or solid type injection and has a uniflow port system of scavenging.

D. SUBMARINE DIESEL ENGINE INSTALLATIONS

1D1. Submarine diesel engine installations. Figure 1-14 shows a typical main and auxiliary engine installation aboard a modern, diesel-electric drive, fleet type submarine. Each engine is coupled with a generator to form a generator set. Through the main control cubicle, the current supplied by main generator sets may be directed to charging the batteries or powering the main motors. The auxiliary generator set may be used directly either to charge the batteries or to power the auxiliary equipment. It may also be used indirectly for powering the main motors. Main motors are used for propulsion and may be powered either by the batteries or by the main generator sets.

20

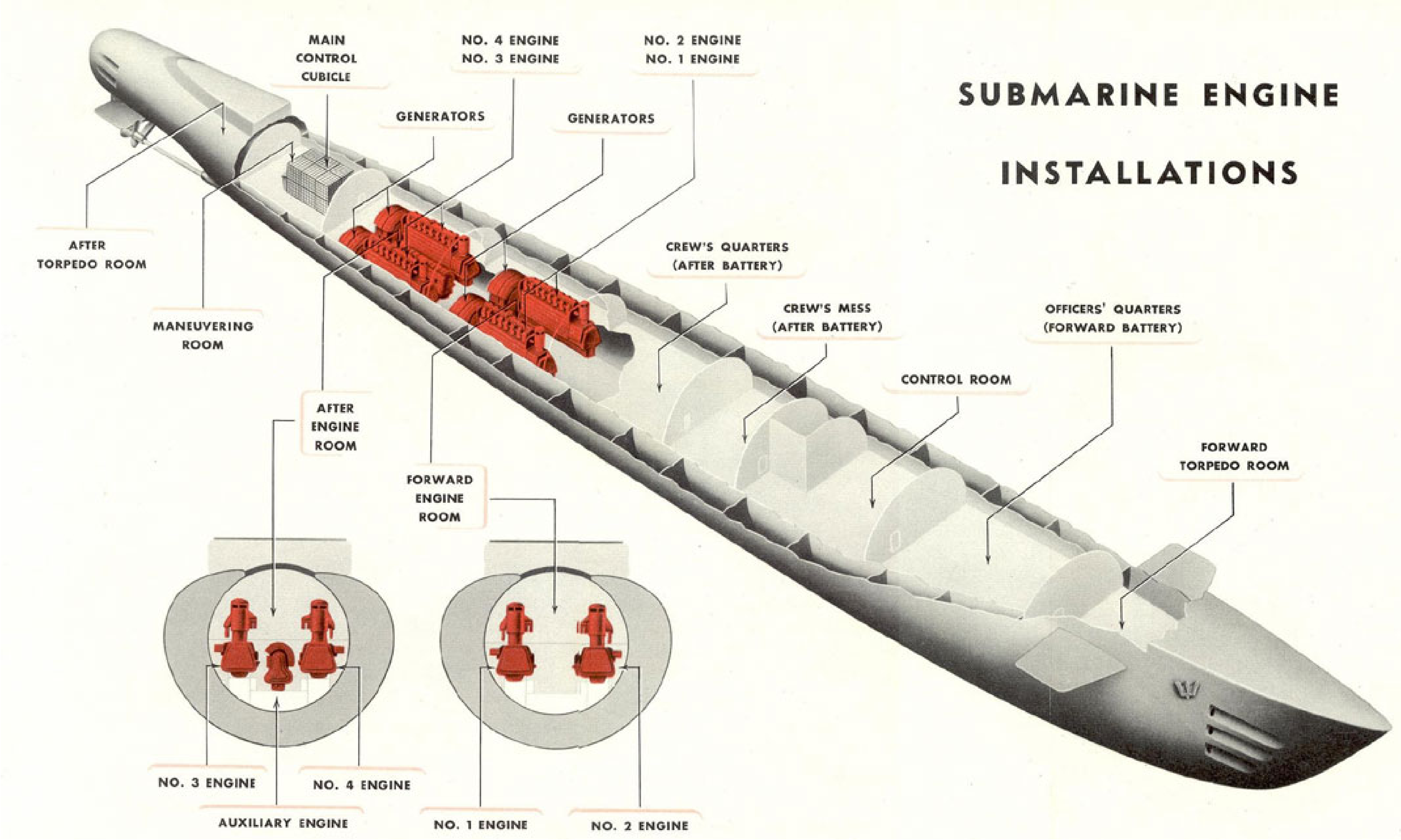
Figure 1-14. CUTAWAY OF FLEET TYPE SUBMARINE SHOWING ENGINE INSTALLATIONS.

Sub Diesel Next chapter
Home Page

Version 1.10, 22 Oct 04

Figure 1-14. CUTAWAY OF FLEET TYPE SUBMARINE SHOWING ENGINE INSTALLATIONS.

Sub Diesel Home Page



SUBMARINE ENGINE
INSTALLATIONS
MAIN CONTROL CUBICLE
NO. 4 ENGINE
NO. 3 ENGINE
NO. 2 ENGINE
NO. 1 ENGINE
GENERATORS
GENERATORS
AFTER TORPEDO ROOM
MANEUVERING ROOM
CREW'S QUARTERS (AFTER BATTERY)
CREW'S MESS (AFTER BATTERY)
OFFICERS' QUARTERS (FORWARD BATTERY)
CONTROL ROOM
AFTER ENGINE ROOM
FORWARD ENGINE ROOM
FORWARD TORPEDO ROOM
NO. 3 ENGINE
NO. 4 ENGINE
AUXILIARY ENGINE
NO. 1 ENGINE
NO. 2 ENGINE

2
MEASUREMENTS AND INSTRUMENTS

A. MEASUREMENTS

2A1. Fundamental and standard units. In order to understand and operate an engine efficiently it is necessary for the operator to be familiar with various units of measurement and the instruments by which they are recorded. As soon as any branch of science is developed to any extent, attempts are made to measure and evaluate the quantities and conditions found to exist. To do this a unit must be selected for each measurable quantity. These units are derived from a set of basic units known as fundamental units. The fundamental units are units of force, length, and time.

Fundamental units should not be confused with standard units. Standard units of measurement are units that are established and legalized by the government of a country. Whenever standardized units are established, the fundamental units are expressed in terms of the standard units to secure uniformity of procedure and comparison.

2A2. The metric system of measurement. The metric system of measurement is used generally throughout the world, particularly in Europe. It is not in general use in the United States. Such units as length, volume, and mass are easily converted to the next higher denomination by using the simple multiplier, 10. For example:

Units of Length

10 millimeters = 1 centimeter
10 centimeters = 1 decimeter
10 decimeters = 1 meter
1000 meters = 1 kilometer

Units of Weight

10 milligrams = 1 centigram
10 centigrams = 1 decigram
10 decigrams = 1 gram
1000 grams = 1 kilogram
1000 kilograms = 1 metric ton

The metric system has been legalized for use in the United States and is frequently used in scientific and laboratory work, because the smaller units facilitate work of extreme accuracy and the use of the simple multiplier, 10, makes computation of work quick and easy.

2A3. The English system of measurement. The English system of measurement is by far the most commonly used in engineering work in the United States. The system is given wide usage primarily because of precedent rather than because of any recommending features such as

Because the metric system is a decimal system, it is less subject to arithmetical error than the other common system, the English system of measurement. Since the metric system uses the simple multiplier, 10, it is easy to establish the value of the unit of measure as denoted by the prefix in the name of the unit. The table below explains how the prefix denotes the value of the unit of measure and gives examples of the use of the prefix.

Prefix	Example
micro (meaning millionth)	micron, micrometer
milli (meaning thousandth)	millimeter, milligram
centi (meaning hundredth)	centimeter, centigram
deci (meaning tenth)	decimeter, decigram
deka (meaning ten)	dekameter
hecto (meaning hundred)	hectometer
kilo (meaning thousand)	kilometer

In the metric system the fundamental units of force, length, and time are expressed in the standard units of kilograms, meters, and seconds. Although the units of the English system are used in some instances in this manual, it is necessary to be able to convert from one system to another; however, the units most often used are those encountered in the metric system.

In the English system the fundamental units of force, length, and time are expressed in the standard units of foot, pound, and second. Unlike the metric system, the English system has no common multiplier and the subdivisions of the units of measurement bear no common relation to each other. For example, below are given the units of length and weight and the relationship of the various subdivisions of each.

Units of Length

12 inches = 1 foot

3 feet 1 yard

5 1/2 yards = 1 rod (16 1/2 feet)

Units of Weight

16 ounces = 1 pound

2000 pounds = 1 ton (short)

2240 pounds = 1 ton (long)

Since all forms of matter are measurable in terms of the fundamental units of force, length, and time, it is possible to combine the units of measurement to express measurement of quantities encountered in various engineering and scientific work. by 2.54, and centimeters converted to inches by dividing centimeters by 2.54,

2A6. Unit of force. Force is the push, pull, or action upon a body or matter at rest which tends to give it motion. In the English system, the unit of force is the

In the following sections, the English and metric units of measurement in engineering work are discussed. In the description of each, it is easy to see how each of these units of measurement may be basically reduced to fundamental units.

2A4. Unit of length. Length is usually defined as the distance between two points. In the English system it is expressed in inches, feet, yards, rods, miles, or fractions thereof. The accuracy required in engineering work makes it a general practice for engineers to measure length in thousandths of an inch. Thus, various tolerances, clearances, and minute measurements are expressed by decimal divisions of an inch in thousandths, such as .125 (one hundred twenty five thousandths).

In a problem involving measurement of area, the area of a regular shape may be expressed by the product of two measurements of length. Thus, a square 3 feet by 3 feet has 9 square feet of area. Likewise, a problem of measuring volume, where the shape is adaptable to linear measurement, may be expressed by the product of three measurements of length. Thus, a cube 3 feet by 3 feet by 3 feet has 27 cubic feet of volume.

2A5. Conversion factors of length. Often when using the English system in engineering work it is necessary to convert measurements to the metric system and vice versa. To change units of one system to those of another it is necessary to have a conversion factor that

pound. In the metric system, the unit of force is the kilogram.

2A7. Unit of work. The work done upon a body is equal to the average force acting upon the body multiplied by the distance through which the body is moved as a result of the force. In the English system, the unit of work is the foot-pound. For example, if a force of 500 pounds acts upon a body to move it 10 feet, 5000 foot-pounds of work have been done upon this body.

2A8. Units of mass and weight. The mass of a body may be defined as the quantity of matter in a body without regard to its volume or the pull of gravity upon it. The term mass must be distinguished from the term weight which is the measurement of the force of gravity acting upon body at any given point upon the earth's surface. Weight varies with locality, but mass is considered constant. The student must not confuse mass with weight although the units are the same for both. The standard kilogram is defined as the mass of a certain piece of platinum iridium in possession of the International Bureau of Weights and Measures. The fundamental unit of mass, the gram, is one one-thousandth of the standard kilogram.

English System	Metric System
1 ounce	= 26.35 grams
1 pound	= 0.454 kilograms
1 gram	= 0.0353 ounces
1 kilogram	= 2.205 pounds

establishes the relation between the two systems for the same quantity. The most commonly used conversion factors between the English and metric systems are:

English System		Metric System
1 inch	=	2.54 centimeters
39.37 inches	=	1 meter

All English system measurements of length may be reduced to inches and all metric system measurements of length to centimeters. Knowing the basic conversion factor, inches can be converted to centimeters by multiplying inches

Kilograms are converted into pounds by multiplying the number of kilograms by 2.205, and conversely pounds are converted into kilograms by multiplying the number of pounds by 0.454. For example, 1 metric ton (1000 kilograms) equals 1000 x 2.205 or 2205 pounds.

2A9. Unit of pressure. Pressure is defined as force per unit area acting against a body. In the English system, the unit of pressure may be expressed as pounds per square inch or pounds per square foot.

Since all forms of matter have weight, the air of the earth's atmosphere has weight. At sea

level, the weight of air exerts a pressure of 14.7 pounds per square inch and has a weight of approximately 0.08 pounds per cubic foot. At higher altitudes, the pressure, and therefore the weight, becomes less.

Gage pressure. Pressure gages are commonly used to determine the pressure existing or to record the peak pressure attained within a container. Most pressure gages make no allowance for atmospheric pressure and normally register zero at existing atmospheric pressure.

Absolute pressure. In practically all pressure problems, atmospheric pressure is present and must be accounted for. When atmospheric pressure is added to the gage or indicated pressure, the total obtained is

of 180 degrees or graduations between the freezing point and the boiling point of pure water at sea level. On the Fahrenheit scale the freezing point of water is fixed at 32 degrees and the boiling point of water at 212 degrees. The centigrade scale is established with a range of 100 degrees or graduations between the freezing point and the boiling point of water at sea level. On the centigrade scale the freezing point of water is fixed at 0 degrees and the boiling point of water at 100 degrees.

a. Absolute zero temperature. Absolute zero temperature is theoretically the lowest temperature that can be obtained. It is that temperature at which all molecular motion ceases entirely and at which point the given matter possesses no heat

the absolute pressure. Thus, absolute pressure is the total pressure recorded from a zero point. For example, the scavenging air pressure in a cylinder is 4 psi. If the cylinder is at sea level, the atmospheric pressure of 14.7 psi must be added, making the total 18.7 psi absolute pressure.

2A10. Unit of power. Work has been defined as force acting through a given distance. Power may be defined as the amount of work performed during a unit period of time. The unit of power used by engineers is the horsepower. One horsepower (hp) equals the amount of work necessary to raise 33,000 pounds through a distance of 1 foot in 1 minute. One horsepower also equals the amount of work necessary to raise 550 pounds through a distance of 1 foot in 1 second.

Example: How many horsepower are required to raise a weight of 12,000 pounds through a distance of 22 feet in 2 minutes?
Solution: (12,000 x 22)/(2 x 33,000) = 4 horsepower

2A11. Unit of temperature. Temperature may be defined as the measure of intensity of heat. In simple language, temperature is the measure of hotness (usually referred to as high temperature) or coldness (usually referred to as low temperature) of a body or matter.

Temperature is measured and expressed in degrees according to established standard scales known as the Fahrenheit and whatsoever. Absolute zero temperature has been determined to be -273 degrees C and -459.6 degrees F. From a practical standpoint, absolute zero is unattainable.

b. Conversion factors of temperature. Since the centigrade scale covers the same temperature range (freezing to boiling points of water) in 100 degrees that the Fahrenheit scale covers in 180 degrees, a centigrade degree equals 9/5 of a Fahrenheit degree. Hence, a centigrade reading may be converted to a Fahrenheit reading by multiplying the centigrade reading by 9/5 and adding 32 degrees. And, conversely, a Fahrenheit reading may be converted to a centigrade reading by subtracting 32 degrees and multiplying by 5/9.

Expressed as a simple equation, the conversion factor is:

$F = 9/5\ C + 32$
$C = 5/9\ (F - 32)$

Example: How many degrees centigrade are 86 degrees Fahrenheit?
Solution: $C = 5/9 \times (86 - 32) = 30$ degrees C.

Example: How many degrees Fahrenheit are 35 degrees centigrade?
Solution: $F = 9/5 \times 35 + 32 = 95$ degrees F.

2A12. Unit of heat. Heat is a form of energy, and the English system unit of heat is the mean British thermal unit (Btu). The British thermal unit is the amount of heat necessary to raise the temperature of 1 pound of water 1 degree F at sea level atmospheric pressure.

centigrade scales. The Fahrenheit scale is established with a range

When 1 pound of fuel oil is completely burned, a certain number of Btu of heat are given off. The quantity of heat liberated by the complete combustion of 1 pound of fuel oil is known as the fuel oils heating value.

Since heat is a form of energy, it cannot be destroyed but may be converted into mechanical energy. One Btu of heat is equivalent to 778 foot-pounds of work. Thus, the conversion factor for power to heat is:

1 hp = 33,000 / 778 = 42.42 Btu per minute

2A13. Unit of time. The standard unit of time in both the English system and the metric system is the second. The second is defined as 1/86,400 part of a mean solar day. The mean solar day is obtained by taking the average length of all the days of the year, a day being measured from the noon of one day to the noon of the next.

The multiples of the units of time are:

60 seconds = 1 minute

60 minutes = 1 hour

24 hours = 1 day

2A14. Units of velocity. Velocity may be defined as the rate of movement of a body. If a body moves a specified distance during a specified time at a uniform speed, the velocity may be determined by dividing the distance by the time. There are two types of velocity normally encountered, linear and angular. If the velocity is linear, the movement is in a straight line and the velocity may be expressed in terms such as feet per second, feet per minute, or miles per hour. If the velocity is angular, the movement of the body is rotary or about a central axis, and the velocity may be expressed in revolutions per minute or revolutions per second. In engineering work it is common practice to rate the velocity of shafts, wheels, gears, and other rotating parts in revolutions per minute (rpm).

B. INSTRUMENTS

2B1. General. In the previous section we have defined and explained the fundamental units of measurement and the standard units of measurement for both the English and the metric systems. It is the purpose

expansion or contraction of the instrument from changes in temperature can be considerable.

c. Calipers. Engineers and machinists frequently use calipers to secure accurate measurements of inside and outside diameters.

of this section to enumerate and describe the various instruments by which these measurements are computed and recorded.

2B2. Instruments for measuring length. a. General. In engineering and machine work there are several instruments for measuring length, area, and volume. Since the measurement of area and volume often can be obtained by compounding simple measurements of length, instruments used for computing area and volume are also described here.

b. Rulers and tapes. The most common method of obtaining simple measurements of length is by the ruler or tape (Figure 2-1). A ruler may be graduated into feet, inches, or fractions thereof. Rulers and tapes used in engineering work are most frequently made of metal and the fractions of inches may be graduated to subdivisions as small as 1/64 or 1/100 of an inch. Care should be exercised in using metal rulers and tapes, especially if extreme accuracy is required. The margin of error due to

Figure 2-2 shows how various caliper settings may be taken and how the registered setting of the calipers may be measured by a ruler or by a micrometer.

d. Micrometer calipers. Engineers frequently rely on the micrometer caliper (Figure 2-3) to obtain measurements accurate to 1/1000 of an inch. This instrument is particularly useful for measuring relatively short lengths and the diameter of journals or cylinders. The common commercial micrometer consists of a frame; an anvil, or fixed measuring point; a spindle; a sleeve, or barrel; and a thimble. The spindle has threads cut 40 to the inch on the portion that fits inside the sleeve. The thimble fits over the end of the sleeve, and rotating the thimble turns the spindle.

Rotating the thimble until the spindle has made one complete turn moves the spindle 1/40 of an inch, which is equal to 0.025 inch. The number of turns the spindle makes is indicated by graduations on the sleeve. Each graduation

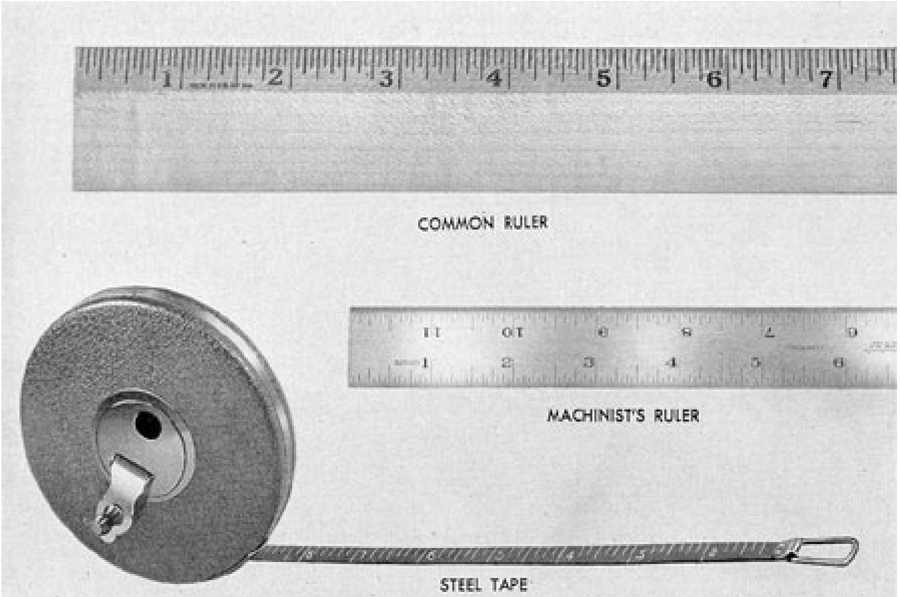


Figure 2-1. Common ruler, machinist's ruler, and steel tape.

represents one complete turn and every fourth graduation is marked 1, 2, 3, and so on, to represent 1/10 of an inch. Thus, each number is equivalent to the sum of four graduations, or 4 x 0.025, which equals 0.100 inch.

The thimble has a beveled edge divided into 25 parts and numbered 0, 5, 10, 15, 20, and back to again. Each of these marks represents 1/25 of a turn or 1/25 of 0.025 which is 1/1000 (0.001) of an inch. A final reading of the micrometer is obtained by multiplying the number of graduations on the sleeve by 25 and adding the number of marks indicated on the beveled edge of the thimble. This gives the reading in thousandths.

For example, in Figure 2-3 the graduations on the sleeve show the spindle has turned 7 revolutions which is equivalent to 7 x 0.025, or 0.175 inch. The thimble has been turned 3 marks, or 0.003 inch. The total reading then is 0.175 plus 0.003, or 0.178 inch.

e. Feeler gages. The feeler gage (Figure 2-4) comes into frequent machine work. Such a gage consists of thin blades of metal of various thicknesses. There is generally a blade or strip for each of the most commonly used thicknesses such as 0.002 inch, 0.010 inch, and .015 inch. The thickness of each blade is generally etched on the blade.

Feeler gages are principally used in determining clearances between various parts of machinery. Probably the most common use is determining valve clearance. Various blades are inserted between the tappet and the push rod until a blade of the feeler gage is found that will just slide between the two surfaces without too much friction or sticking. The thickness of the blade then determines the clearance. Or, a particular feeler of proper thickness may be selected and the tappet adjusted until the feeler will just slide between the tappet and push rod with out catching.

f. Bridge gages. Bridge gages are used to measure the amount an engine main bearing has dropped due to wear. Figure 2-5 shows

use in engineering and

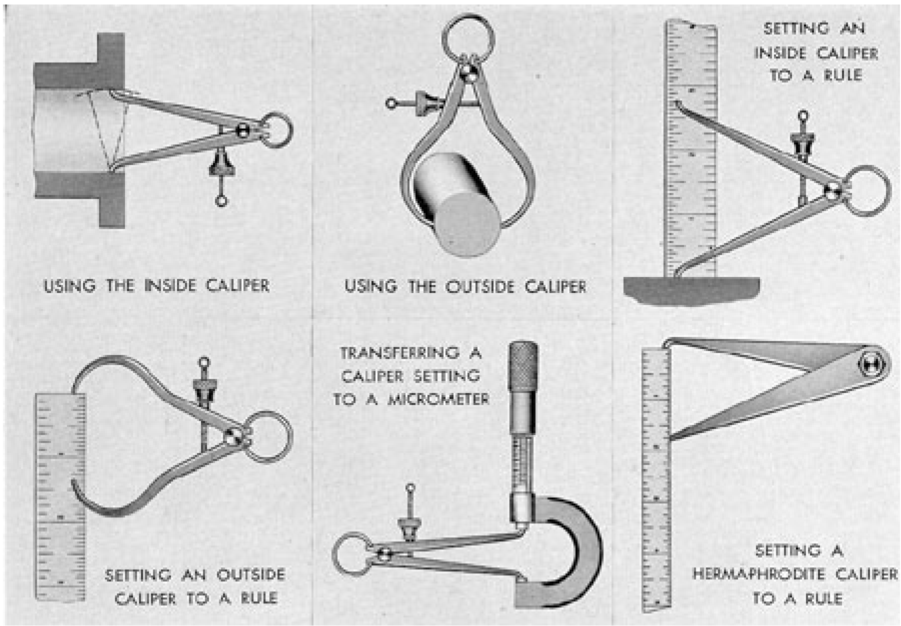


Figure 2-2. Types of calipers and methods of measurement.

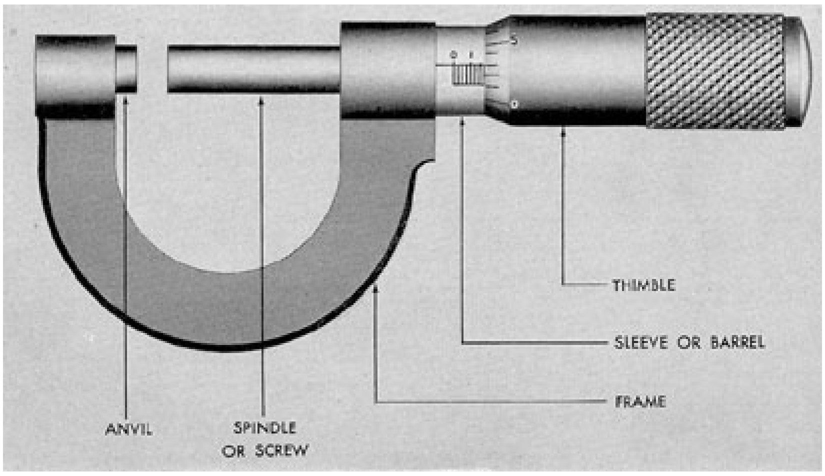


Figure 2-3. Micrometer.

a bridge gage in use. The upper cap of the main bearing has been removed and the bridge gage has been placed over the journal as shown. A feeler gage is then inserted between the tip of the bridge gage and the journal. The measurement recorded by the feeler gage is then compared to the original measurement taken at the time the engine was installed or with

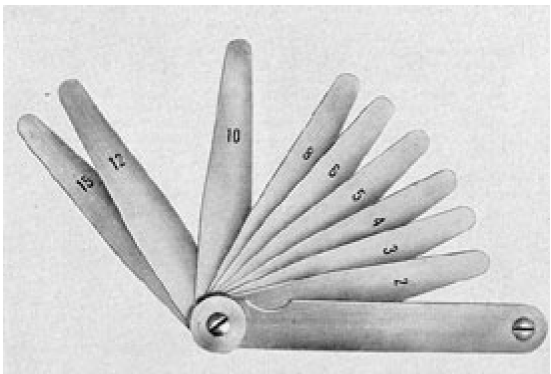


Figure 2-4. Feeler gage.

previous bridge gage readings. Thus, the amount of bearing wear can be determined.

Bridge gages must be handled with great care. If the tip on the gage or the supporting surfaces becomes burred, worn, or distorted, the gage will give an inaccurate reading.

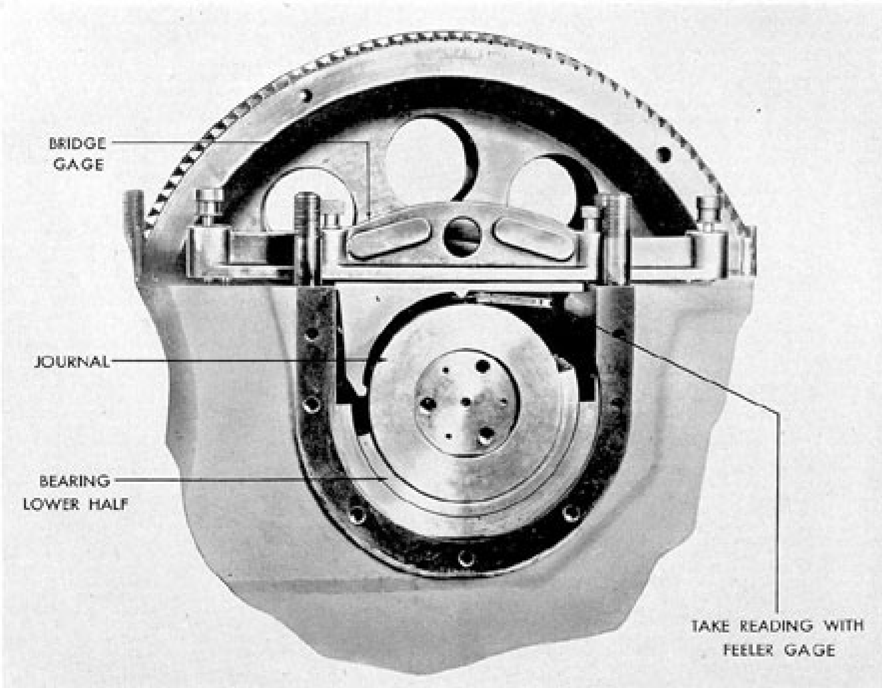


Figure 2-5. Using bridge gage and feeler gage to determine clearance.

2B3. Instruments for measuring temperature. a. General. As previously stated, temperature is a measure of the intensity of heat, and the measurements may be made with one of several instruments. The instruments most commonly used for measuring temperatures below 1000 degrees F are the mercury thermometer, the thermocouple pyrometer, and the electrical resistance thermometer. For taking temperature measurements above 1000 degrees F, the most commonly used instrument is the thermocouple pyrometer. percentage of error present.

b. Liquid-in-glass thermometers. In the type of thermometer in which a hollow glass stem is filled with a liquid (Figure 2-6) the liquid most commonly used is mercury, although some thermometers are filled with alcohol or pentane. In some cases, where extremely low temperatures are to be recorded, a gas may be used. In the construction of the common mercury thermometer, care is used in sealing the stem to insure that a vacuum exists above the column of mercury in the stem. Otherwise, the mercury would have to

In taking measurements with thermometers and pyrometers, the operator should bear in mind the possibility of errors in measurement and what effect they may have on his particular problem. An error is the difference between the observed value and the true value and may be expressed as a percentage. Some errors inherent in an instrument may be avoided by periodically checking the calibration of an instrument with one of known accuracy. Sometimes, errors due to the aging or failure of materials in the instrument are unavoidable, such as the deterioration of glass due to aging and repeated stress. A check of the instrument will indicate the compress the air in the stem, and a false reading would result.

To graduate a thermometer (Figure 2-7) the bulb and a portion of the stem holding the mercury are submerged in melting ice and the point at which the mercury stands in the tube is marked 32 degrees if the thermometer is Fahrenheit, or 0 degrees if the thermometer is centigrade. Next, the bulb and stem are placed in a closure in which they are surrounded by steam rising off boiling water at sea level atmospheric pressure. The position of the top of the column of mercury is then marked at 212 degrees if the thermometer is Fahrenheit, or at 100 degrees if the

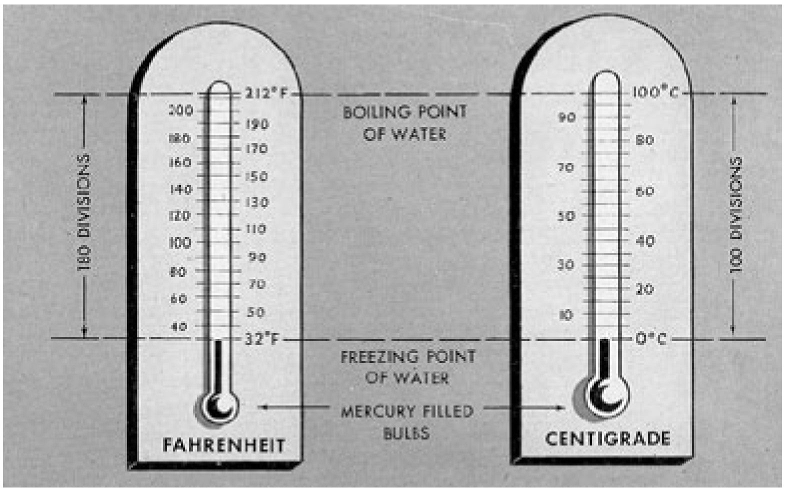


Figure 2-6. Fahrenheit and centigrade thermometers.

thermometer is centigrade.

On Fahrenheit thermometers the distance between the 32 degrees and the 212 degrees marks is graduated and marked into 180 equal parts, each space or subdivision representing 1 degrees F. On centigrade thermometers the distance between the 0 degrees and 100

cause the galvanometer pointer to move across its scale accordingly. Metals commonly used in the thermometer bulb are platinum and nickel.

degrees marks is graduated and marked into 100 equal parts, each space representing 1 degree C. The space above and below these markings is calibrated into the same graduations for the entire temperature range of the thermometer.

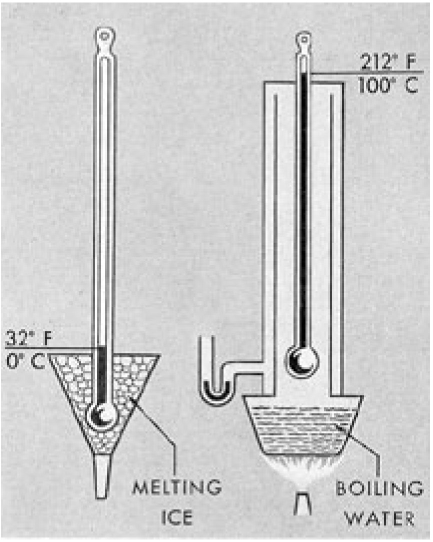


Figure 2-7. Method of graduating thermometers.

c. Electrical resistance thermometers. Electrical resistance thermometers (Figure 2-8) make use of the principle that the electrical resistance of various metals varies with their temperature. The resistance is measured by a Wheatstone bridge which is connected to a galvanometer calibrated to read in degrees of temperature. One leg of the balanced bridge circuit is led to the thermometer bulb which is inserted at the point where the temperature is to be measured. A temperature change at the thermometer bulb will change the resistance with regard to the circuit, causing an electrical unbalance in the entire bridge. This unbalance will

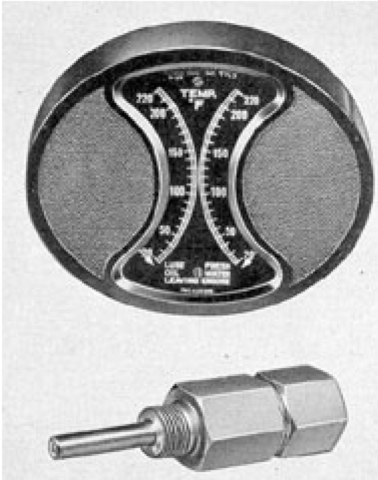


Figure 2-8. Electrical resistance thermometer dial and bulb.

d. Thermocouple pyrometers. The thermocouple unit of the pyrometer (Figure 2-9). is made of two wires or strips of dissimilar metals connected at one end and having an electrical connection at the other end. When the two ends or junctions are subjected to different temperatures, an electrical current is generated. This current is measured to give an indication of the differences in temperatures between the two junctions. In submarines the most common application of this instrument is for measuring the exhaust temperature in the exhaust elbows of the engine. One of the two thermocouple wires is made of pure iron and the other is made of constantan, a nickel copper alloy. The wires are twisted together and welded at the tip of the thermocouple and mounted in the closed end of the protecting tube made of pure nickel. The protecting tube is fitted with a terminal head in which the connections are made between the extension leads and the thermocouple wires. These connections between the thermocouple and the

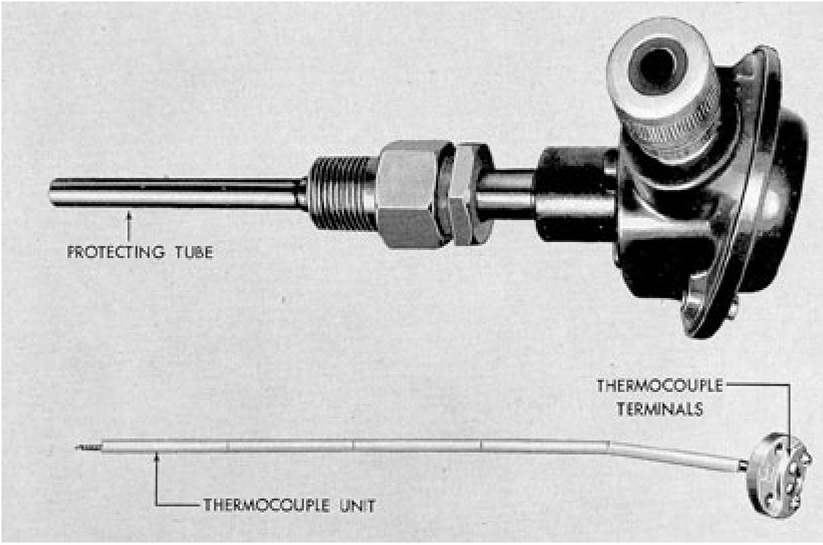


Figure 2-9. Thermocouple pyrometer and thermocouple unit.

indicating instrument are made with wires of the same material as the thermocouple and cause the cold junction to be extended from the thermocouple terminals back to the indicator. Other types of wires are never used for this purpose.

2B4. Instruments for measuring pressure. a. Barometers. The most common instrument in use for measuring atmospheric pressure is the mercury barometer (Figure 2-10). This instrument consists of a long, hollow, glass tube, sealed at one end and with the open end of the tube submerged beneath the surface of an open container of mercury. An increased air pressure acting upon the surface of the mercury in the open container causes the mercury to rise in the tube. The space between the mercury and the sealed end inside the tube is a vacuum so that air will not be compressed in the tube and counteract the pressure exerted outside. The tube containing the column of mercury is calibrated in inches and subdivisions of 1/100 of an inch. As atmospheric pressure acting upon the surface of the mercury in the open container varies, the column of mercury in the tube rises and falls and the amount can be measured by the calibrations on the tube. When the column of mercury stands at 29.92 inches at 32 degrees F and at sea level, standard atmospheric pressure is registered.

Another type of barometer is the aneroid barometer (Figure 2-10). The aneroid barometer consists of an exhausted chamber with corrugated diaphragm walls. Atmospheric pressure causes the diaphragm walls to deflect against the resistance of a spring. The deflections of the diaphragm walls against the spring are recorded by a lever or indicator upon a calibrated face through a delicate system of levers. Some aneroid barometers are so sensitive that they will register a change when raised or lowered only a few feet. Due to the effect of aging and fatigue of the diaphragm construction, aneroid barometers should have their calibrations

frequently checked against mercury barometer readings.

b. Pressure gages. Pressure gages (Figure 2-11) of the diaphragm or tube type are generally used for determining the pressure of steam, water, air, and other mediums. The aneroid barometer described above is an example of the diaphragm type pressure gage. However, the tube type gage is considered more accurate. Such a gage is called a Bourdon gage. The simplex pressure gage illustrated in Figure 2-11 is a Bourdon type gage. This gage consists of an elastic metal tube of oval cross section, bent into an arc. The two metals commonly used in making the tube are brass and steel. In low-pressure gages, brass is normally used but if the pressures to be measured exceed 100 psi, the tubes are always constructed of steel. One end of the tube is fixed and the other end is movable. The free end of the tube is connected to a spring-loaded needle through a gear and system of levers. Pressure exerted on the inside walls of the oval tube tends to make the tube straighten out. The free end of the tube pulls on the end of the lever, the motion of which is transmitted to the needle. The needle registers across the face of the dial, and the gage is calibrated so that it will indicate the pressure in pounds per square inch.

2B5. Instruments for measuring volume. a. Sounding. One of the most common measuring problems in diesel engineering is determining the volume of fluid remaining in fuel oil and lubricating oil tanks. The simplest and most accurate method of determining the volume of fluid in a tank is by sounding. In submarine fuel systems, as fuel is withdrawn from a tank, it is replaced by compensating water. Small sounding tubes of various lengths are installed in the tanks to determine whether there is oil or water at various levels.

b. Fuel oil meters. Fuel oil meters are also used in submarine fuel systems to indicate the amount of fuel withdrawn from the main fuel tanks. Fuel oil meters should be checked frequently for accuracy. Strainers should be

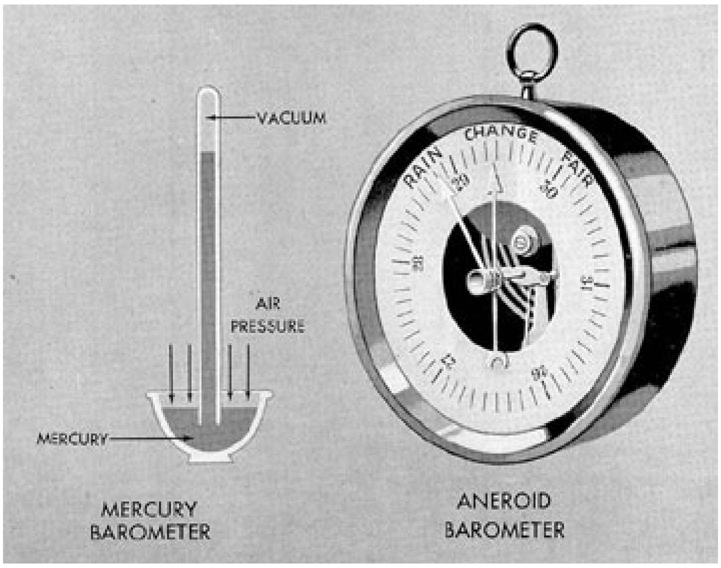


Figure 2-10. Mercury and aneroid barometers.

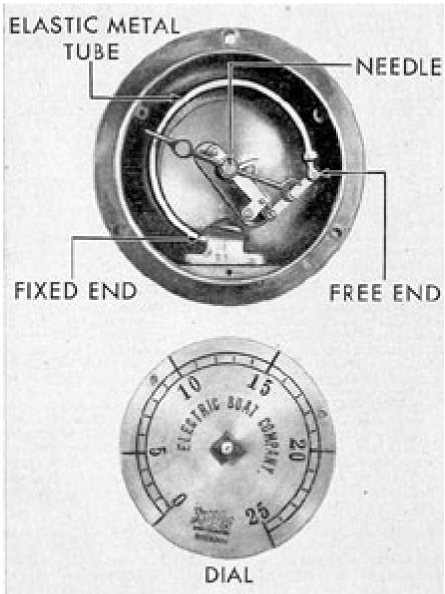


Figure 2-11. Simplex tube type pressure gage and dial.

installed in the line to the fuel oil meter to prevent any foreign substance from getting into the meter mechanism and affecting the accuracy of its registration.

c. Liquidometers. In submarines, liquidometers are frequently used to determine: 1) the level of the liquid in a partially filled tank, and 2) the level between two dissimilar liquids in a completely filled tank.

The liquidometer is equipped with a float mechanism, the

b. Revolution counters. Revolution counters (Figure 2-12) used aboard ship are principally of three types: mechanical, electrical, and electro-mechanical. The mechanical type may be either of the rotating type or the oscillating ratchet type. Probably the most accurate of the common counter devices is the rotating counter with a magnetic clutch connector and a synchronous electric timer operated by the same switch. It is frequently used for calibrating other counters.

The rotating continuous counter may have direct-reading wheels of the cyclometer type or may operate dials or pointers through a gear train. The oscillating or stroke counter is adapted for low speeds only. Rotating counters may be obtained for high-speed work, up to 5000 rpm. It is important that a counter not be used for speeds higher than the speed limits recommended by the manufacturer.

movement of which actuates a double-acting opposed hydraulic mechanism which registers upon a calibrated dial the volume of the desired liquid.

2B6. Instruments for measuring rotational speed. a. General. Aboard ship it is often imperative to know the rotational speed of an engine or piece of machinery which is generally measured in rpm. Various instruments such as revolution counters, mechanical tachometers, and electrical tachometers, are available for securing this measurement.

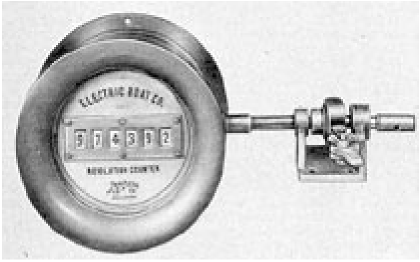


Figure 2-12. Mechanical revolution counter.

c. Mechanical tachometers. Tachometers (Figure 2-13) are measuring instruments that give a direct and continuous indication of rotary speed in rpm. For submarine diesel engines, the mechanical tachometers are usually permanently mounted on a gage board. They are generally driven from the engine camshaft through a gearing and a flexible shaft. In operation, the force produced by the rotation is balanced against a calibrated spring or against the force of gravity. Those used in submarines are usually of the indicator type in which the pointer registers the rpm at the moment, rising and falling with the fluctuations in engine speed.

Hand type tachometers have frequent use in engineering work. This type of tachometer is generally held in the hand and pressed firmly against the end of a rotating shaft to register the rpm directly. Some types of hand tachometers have several sets of change gears so that a wide range of rotary speeds may be accurately read with a single instrument.

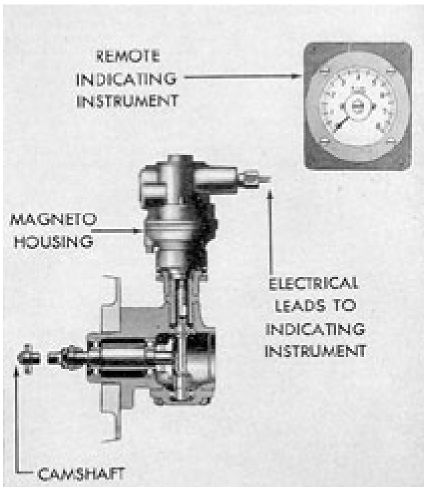


Figure 2-14. Electrical tachometer.

the electric current generated actuates an indicator which is

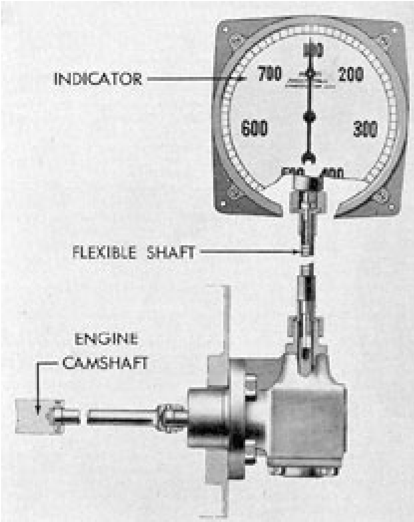


Figure 2-13. Mechanical tachometer.

d. Electrical tachometers. Electrical tachometers (Figure 2-14) of the indicating type are used with submarine diesel engines. The drive mechanism for the electrical tachometer is actuated by the engine camshaft. The drive in turn powers a tachometer magneto and calibrated to register engine revolutions per minute. The electrical tachometer possesses the distinct advantage that the indicating instrument may be mounted at a distance from the drive mechanism.

All tachometers should be checked frequently for accuracy. This check can be made by using a mechanical revolution counter which is 100 percent accurate. The tachometer is checked against the counter for several minutes with a stop watch and then the reading on the counter is divided by the number of minutes to check the number of rpm.

3

ENGINES AND ENGINE COMPONENTS

3A1. Introduction. All of the present fleet type submarines are equipped with engines manufactured either by the Cleveland Diesel Engine Division, General Motors Corporation, Cleveland, Ohio, or by Fairbanks, Morse and Company, Beloit, Wisconsin. These engines have been in the process of development for the past several years, and the latest models proved highly dependable under wartime operating conditions.

Before World War II, these engines were used almost exclusively on submarines. With the expansion of the Navy, however, these engines have also been used on destroyer escorts, amphibious craft, escort type patrol vessels, and various auxiliary craft.

The following sections are devoted to the discussion of basic diesel engine construction and the application of these basic principles to the General Motors and Fairbanks-Morse engines.

3A2. General Motors engines. Two models of GM main engines are found in fleet type submarines today, Model 16-248 and Model 16-278A. The former was installed exclusively in General Motors engine equipped vessels until early in 1943 when supplies a Model 8-268 auxiliary engine for fleet type submarines. This is an 8-cylinder, in-line, 2-cycle, air starting engine, rated at 300 kw generator output at 1200 rpm. The size of the bore and stroke is 6 3/8 inches and 7 inches respectively.

The tables at the end of this chapter, pages 78 and 79, contain engine data, ratings, and clearances for General Motors main engines and auxiliaries.

3A3. Fairbanks-Morse engines. There are two types of F-M main engines in use in modern submarines (Figures 1-12 and 1-13). The model number for each is 38D 8 1/8. The basic difference between them is the number of cylinders, one being a 9-cylinder and the other a 10-cylinder engine. Both engines have the same bore and stroke and in most respects are similar in principle, design, and operation.

The F-M 38D 8 1/8 model is an opposed piston, in-line, 2-cycle, 9- or 10-cylinder engine employing air starting and rated at 1600 bhp at 720 rpm. Bore and stroke are 8 1/8 and 10 inches respectively.

An auxiliary engine, Model 38E 5 1/4, is also supplied by Fairbanks, Morse and Company. This is a 7-cylinder, opposed piston, 2 cycle, air starting engine rated at 300 kw

Model 16-278A was introduced. All General Motors installations since that time have been Model 16-278A engines (Figures 1-10 and 1-11). Basically the two models are similar. The principal differences are in the size and design of the parts, methods of construction, and type of metals used. In the following chapters all references are based on the current Model 16-278A. Important differences between the two models, however, will be noted.

The GM engine is a 16-cylinder V-type engine with 2 banks of 8 cylinders each The engine operates on the 2-stroke cycle principle, is air started, and is rated at 1600 bhp at 750 rpm. The size of the bore and stroke of the 16-248 engine is 8 1/2 inches and 10 1/2 inches respectively as compared to 8 3/4 inches and 10 1/2 inches for Model 16-278A.

The General Motors Corporation also generator output at 1200 rpm. The bore is 5 1/4 inches and the stroke 7 1/4 inches.

The tables at the end of this chapter, page 80, contain engine data, ratings and clearances for Fairbanks-Morse main engines and auxiliaries.

3A4. Classification of engine components. To simplify the study of the design, construction, and operation of the component parts of the diesel engines in the following sections of this chapter, the parts have been classified under three subjects as follows: 1) main stationary parts, 2) main moving parts, and 3) valves and valve actuating gear.

Section 3B deals with engine components as listed above, in general. Sections 3C and 3D deal with the same components as applied to the GM and F-M engines respectively. In all

instances the ends of the engines will be referred to as the blower and the control ends. It should be noted that the blower end of the F-M engines is also the generator coupling end, whereas the blower end of the GM engines is opposite the generator coupling end.

B. GENERAL DESCRIPTION OF ENGINE COMPONENTS

3B1. Main stationary parts.a. Frame. The framework of the diesel engine is the load carrying part of the machinery. The design of diesel engine frames has undergone numerous changes in recent years. Some of other parts for inspection and repair. The doors are usually secured with handwheel or nut operated clamps and are fitted with gaskets to keep dirt and foreign material out of the interior. Some of these access doors or

the earlier types of framework which were eventually abandoned were: 1) A-frame type, 2) crankcase type, 3) trestle type, 4) stay-bolt or tie rod type.

The framework used in most modern engines is usually a combination of these types and is commonly designated as a welded steel frame. A frame of this type possesses the advantages of combining greatest possible strength, lightest possible weight, and greatest stress resisting qualities.

The welded steel type of construction is made possible by the use of recent developments in superior quality steel. For diesel engine frame construction, steel is generally used in thick rolled plates which have good welding quality. In this type of construction, deckplates are generally fashioned to house and hold the cylinders, and the uprights and other members are welded, with the deckplates, into one rigid unit.

b. Oil drain pan. The oil drain pan is attached to the bottom of the cylinder block and serves to collect and drain oil from the lubricated moving parts of the engine. The bottom of the oil pan is provided with a drain hole at each end through which oil runs to the sump tank. In some installations the bottom of the pan slopes toward one end or the other of the engine.

Oil drain pans require little maintenance. They should be cleaned and flushed of any residual dirt during major overhaul periods. New gaskets inspection covers may be constructed to serve as safety covers. A safety cover is equipped with a spring-loaded pressure plate. The spring maintains a pressure which keeps the cover sealed under normal operating conditions. An explosion or extreme pressure within the crankcase overcomes the spring tension and the safety cover acts as an escape vent, thus reducing crankcase pressure.

d. Cylinder and cylinder liners. The cylinder is the enclosed space in which the mixture of air and fuel is burned. A cylinder may be constructed of a varying number of parts among which the essentials are the cylinder jacket, the cylinder liner, and in most cases the cylinder head. In most designs the space between the cylinder jacket and the liner is cored to carry circulating water for cooling purposes.

There are two general types of cylinder liners. One, the wet type, is a replaceable liner that makes direct contact with the cooling water; the other, the dry type, is a replaceable liner that fits into a water-cooled jacket without making direct contact with cooling water. All submarine diesel engines under consideration here use the wet type cylinder liners.

e. Cylinder head. The cylinder head seals the end of the cylinder and usually carries the valves. Heads must be strong enough to withstand the maximum pressures developed in the cylinders. Also, the joint between the cylinder and the head must be gastight. Due to the high temperatures encountered, cylinder heads must

should be installed at these times to assure an oiltight seal.

c. Access doors and inspection covers. The cylinder block walls are equipped with access doors or handhole covers. With the doors or covers removed, the openings furnish access to cylinder liners, main and connecting rod bearings, injector control shafts, and various

be water cooled. To accomplish this, water passages are cored in the head during the casting process. Valves usually found in the head are the exhaust valves, injection valves, and air starting valves.

3B2. Main moving parts.a. General. The main moving parts of a diesel engine are those

that convert the power developed in the cylinders by combustion to mechanical energy, that is delivered to the shaft. These parts are used to change the reciprocating motion of the pistons in the cylinders to rotary motion at the engine final drive, and may be divided into three major groups:

1. Those parts having rotary motion, such as crankshafts and camshafts.

2. Those parts having reciprocating motion, as, for example, the pistons and piston rings.

3. Those parts having both reciprocating and rotary motion, such as the connecting rods.

b. Crankshaft. The crankshaft transforms the reciprocating motion of the pistons into rotary motion of the output shaft. It is one of the largest and most important moving parts of a diesel engine.

The materials used in the construction of crankshafts vary greatly, depending on the size of

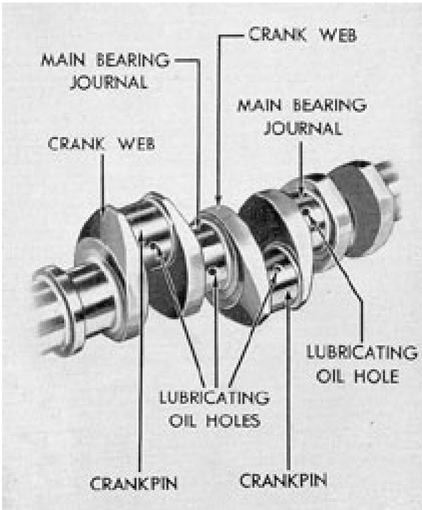


Figure 3-1. Nomenclature of crankshaft parts.

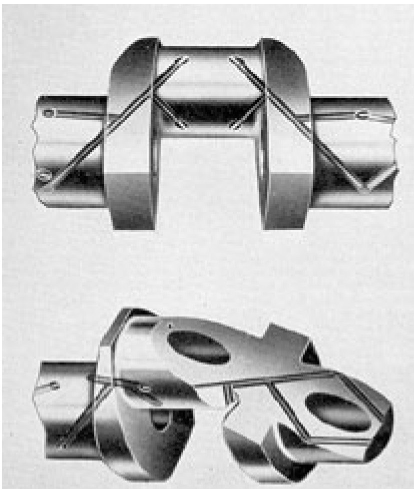


Figure 3-2. Sections of crankshaft showing oil passages and hollow construction.

the shaft, speed of the engine, horsepower of engine, and number of main bearings. Regardless of materials used, crankshafts are always heat treated. This is necessary in order to give uniform grain structure, which increases ductility and capacity for resisting shock. The tensile strength of crankshaft materials varies from 60,000 psi to as much as 100,000 psi. Crankshafts may be either forged or cast. They may be either made up in one section, or in two or more with the sections interchangeable for economy in construction and replacement. Crankshafts are machined to very close limits with a high finish and are balanced both statically and dynamically.

The crankshaft consists essentially of a number of cranks placed at equal angular intervals around the axis of the shaft. Between the cranks are the crankshaft supports commonly referred to as the journals. Each crank on a crankshaft is made up of the crankpin, which is the journal for the connecting rod bearing, and two crank webs (Figure 3-1).

Journals, crankpins, and webs are drilled for the passage of lubricating oil (Figure 3-2). All such holes are usually straight to facilitate construction and cleaning of the passages. In larger engines, crankshafts are practically always constructed with hollow main bearing journals and crankpins. This construction is

much lighter than a solid shaft and is better adapted for carrying the lubricating oil to various bearings in the engine. In large engines, the crankshaft is sometimes built up by pressing the journals into the webs. In this type, generally, the crankpin and its two adjacent webs are forged or cast in one piece, this unit then being joined to other cranks by hydraulically pressing them onto the main bearing journals. The cranks are held at the proper angles during this process, after which the assembled shaft is put in a lathe and finished to size.

c. Main bearings. The function of the main bearings is to provide supports in which the crankshaft main bearing journals may revolve. In the diesel engines under discussion, modern bimetal or trimetal, split sleeve, precision type main bearings are used exclusively. Bimetal bearings consist of a thin inner layer of soft low-friction metal encased in a shell of harder metal fitted to the bearing support or bearing cap. Trimetal bearings have an intermediate layer of bronze between the shell and soft metal layers. Both types are split sleeve, divided horizontally through the center, for installation. Precision type manufacture requires that the bearing housing be precision bored to a close tolerance and that the bearing halves, when tightly drawn together, align perfectly and fit the bearing journals with a predetermined clearance. The purpose of this clearance is to provide for a thin film of lubricating oil which is lubricant to prevent a metal-to-metal contact between the journal and bearing surfaces. Excessive clearance permits the free flow of the fluid oil to the edges of the bearing. This reduces the pressure developed and consequently may overload the bearing. The stress of overload will cause the bearing to wipe and eventually burn out. Both bearing clearances and the amount of wear may be checked by measuring the thickness of the soft metal lining of the bearing shell either with a ball point micrometer or by the use of appropriate feeler gages.

Proper seating of the bearing shells and proper clearances of precision type bearing shells require that the bearing caps be drawn to the proper tightness. This is done with a torque wrench by means of which the proper torque limits in foot-pounds are obtained. As this torque varies with engine models, the current instructions should be consulted.

d. Pistons. The function of a piston is to form a freely movable, gastight closure in

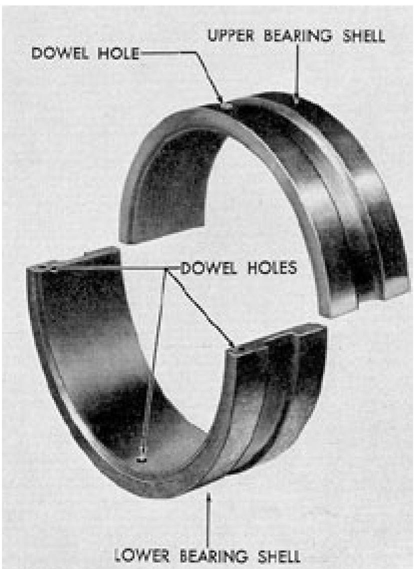


Figure 3-3. Main bearing shells.

forced under pressure between the journals and bearing surfaces. Under proper operating conditions this oil film entirely surrounds the journals at all engine load pressures.

All main bearings contain oil inlet holes and oil grooves which permit the oil to enter and be evenly distributed throughout the inside of the bearing. These oil inlets and grooves are invariably in the low oil pressure area of the bearing.

Proper bearing lubrication depends upon accurate bearing clearances as well as the type of lubrication. Too little clearance will cause the bearing to run hot and wipe out under continued operation. At high operating speeds with too little clearance, the load pressure on the bearing does not leave sufficient room for the

the cylinder for the combustion chamber. When combustion occurs, the piston transmits the reciprocal motion or power created to the connecting rod. Pistons for all the modern submarine 2-stroke cycle diesel engines are of the trunk type. Pistons of the trunk type have sufficient length to give adequate bearing surface against the side thrust of the connecting rod. Trunk type pistons have a slight amount of taper at the crown end of the piston to provide for the greater expansion of the metal at the combustion end where temperatures as high as 3000

and the integral hub of the connecting rod. The piston pin must be strong enough to transmit power developed by the piston to the crankshaft through the connecting rod. Piston pins are usually hollow and are made of special alloy steels, case hardened and ground to size. The connection between the piston and the piston pin is either by means of needle type roller bearings or by plain bushings. The ends of the pins must not protrude beyond the surface of the piston, and their edges must be rounded to facilitate entry of the piston into the cylinder. This is

degrees F may be encountered. This taper is sufficient so that at normal operating temperatures the piston assumes the same diameter throughout its entire length.

The piston crowns on both the GM and F-M engines are concave. The purpose of this shape is to assist in air turbulence which mixes fuel with air during the last phase of the compression stroke.

Pistons are usually constructed of either a cast iron or aluminum alloy. They must be designed to withstand the gas pressure developed in the combustion chamber during the compression and expansion strokes. They must also be light enough to keep the inertia loads on the piston pins and main cranks to a minimum.

e. Piston rings. Piston rings have the following three primary functions:

1. To seal compression in the combustion chamber.

2. To transfer heat from the piston to the cylinder wall.

3. To distribute and control lubricating oil on the cylinder wall.

In general, piston rings are of two types. One, the compression type ring, serves primarily to seal the cylinder against compression loss; the other, the oil type ring, distributes oil on the cylinder walls and controls cylinder wall lubrication by collecting and draining excess oil.

usually accomplished by means of piston pin caps.

g. Connecting rods. Just as its name implies, the connecting rod connects the piston with the crankshaft. It performs the work of converting the reciprocating, or back-and-forth, motion of the piston into the rotary, or circular, motion of the crankshaft. The usual type of connecting rod is an I-beam alloy steel forging, one end of which has a closed hub and the other end an integral bolted cap. The cap is accurately located by means of dowel pins. Through the closed hub, the connection is made between the piston and the connecting rod by means of the piston pin. At the other end, the connecting rod bearing connection is made between the connecting rod and the crankshaft. The shaft of the connecting rod is drilled from the connecting rod bearing seat to the piston pin bushing seat. Through this passage, lubricating oil is forced from the connecting rod bearing to the piston pin bearing for lubrication and piston cooling.

h. Connecting rod bearings. The purpose of these bearings is to form a low-friction, well-lubricated surface between the connecting rod and the crankshaft in which the crankpin journals can revolve freely. The bearings used are generally of the same material and type as the main bearings. Connecting rod bearings consist of two halves or bearing shells. The backs of these shells are bronze or steel, accurately machined to fit into a precision machined bearing seat in the connecting rod. The shells are lined with a layer of soft metal of uniform thickness. When

Piston rings are generally constructed of cast iron. On the average diesel piston there are four to five compression rings and two or three oil control rings.

f. Piston pins. Each piston is connected to the connecting rod by a piston pin or wrist pin. This connection is through bored holes in the piston pin hubs at the center of the piston

the bearing caps are drawn tight on the connecting rod, the contact faces of the bearing shells form an oiltight joint. Also, because of the precision manufacture of all parts,

the bearing shells give the proper clearance between the bearing shells and the crankpin journals. The connecting rod bearings are pressure lubricated by oil forced through oil passages from the main bearings to the crankpin journals. The oil is evenly distributed over the bearing surfaces by oil grooves in the shells.

Figure 3-4. Connecting rod bearing shells.

3B3. Valves and valve actuating gear.a. General. Control of the flow of fuel, inlet air, starting air, and exhaust gases in a diesel cylinder is accomplished by means of various types of valves. The timing and operation of these valves, for the various processes in relation to piston travel and correct firing with the operation of the crankshaft through the camshaft drive. In addition to actuating valves, camshafts, on some engines, are also used for driving auxiliaries such as governors and tachometers.

Camshafts are usually constructed in one or two parts. The number of cams on a camshaft is determined by the type and cycle of engine. The cams and camshafts are usually forged integral and ground to a master camshaft.

c. Valves. The important valves found on typical diesel cylinders and their functions are:

1. Exhaust valves. Exhaust valves are used to allow the exhaust gases of combustion to escape from the cylinders. They are subject to extremely high temperatures and are therefore made of special heat-resistant alloys. In some large engines, the exhaust valves are water cooled.

2. Inlet valves. Inlet valves are used to govern the entrance of air in the cylinder of a 4-stroke cycle engine. Inlet valves are not used

sequence, are the main functions of the valve actuating gear.

Since certain phases of timing, such as the geometrical angle of the crankshaft cranks and the geometrical angle of the camshaft cams, are fixed, timing adjustments are made through the valve actuating gear. Hence, timing adjustments must be made with extreme accuracy and the valve actuating gear must function perfectly for efficient engine operation.

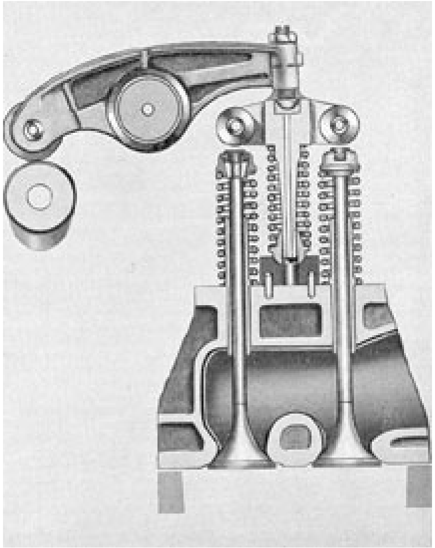


Figure 3-5. Valve actuating gear assembly.

b. Camshafts. The purpose of the camshafts in submarine diesel engines is to actuate exhaust valves, fuel injectors, fuel injection pumps, and air starting valves according to the proper timing sequence of that particular engine.

In order to perform these functions at the various cylinders in relation to their proper firing order, the camshafts are timed or synchronized

in modern submarine diesel engines, having been replaced by inlet ports.

3. Fuel injection valves. Fuel injection valves are used to inject the fuel spray into the cylinder at the proper time with the correct degree of atomization. In addition, some injection valves also measure the amount of fuel injected.

4. Air starting valves. Air starting valves are used to control the flow of starting air during air starting of an engine. These pressure readings of the cylinder while the engine is in operation.

6. Cylinder relief valves. A cylinder relief, or safety, valve is located on each cylinder of all submarine type engines. The function of this valve is to open and relieve the cylinder when pressure inside the cylinder becomes excessive. These valves are adjustable to be set at varying pressures according to the particular installation. When pressure drops below the setting at which the valve opens, the valve closes automatically.

valves are normally of two types, air starting check valves and air starting distributor valves.

5. Cylinder test valves. Each cylinder is provided with a test valve which is used to vent the cylinder before starting. This valve is also used to relieve the cylinder of compression when turning over the engine by hand. The same valve is used far taking compression and firing

d. Valve actuating gear. Motion of the cams on the camshaft is transmitted to valves, injectors, and injector pumps by means of rocker arms or tappet assemblies. The rocker arms and tappets normally are spring loaded and make contact with the cams by means of cam rollers. Adjustments of the various springs and rods are very important, as they are normally the means by which the engine is correctly timed.

C. GENERAL MOTORS ENGINE COMPONENTS

3C1. General. Descriptions of engine components in this section apply only to the General Motors engine.

3C2. Main stationary parts.a. Cylinder block. The cylinder block of the GM engine (Figure 3-8) is fabricated from forgings and steel plates welded together to form a single unit. The assembly is designed with two cylinder banks, the axes of which are 40 degrees apart, forming the V-type design of the engine. The unit is fabricated from main structural pieces called transverse frame members, upper and lower deckplates for each bank, and cross braces all welded into one rigid compact unit. The upper and lower deckplates are bored to accommodate the cylinder liners. The space between these deckplates, as well as the space between the two banks of cylinders, serves as a scavenging air chamber.

The forged transverse members in the bottom of the cylinder block form the mounting pads openings in the sides of the cylinder block. Access to the injector control shaft is obtained by removing the top row of small handhole covers. The middle row of handhole covers permits access to the scavenging air box for inspection of the cylinder liners and piston rings. The bottom row of handhole covers permits access to the crankshaft, connecting rod, and bearings.

b. Engine oil pan. The engine oil pan is bolted to the bottom of the cylinder block. The bottom of the oil pan is provided with a drain hole at each end. One end of the oil pan is fastened to the camshaft gear train housing and the other end is fastened to the blower bottom housing. The lubricating oil from these units drains into the oil pan. The pan is constructed of welded steel in the 16-278A and of an aluminum alloy casting in the 16-248.

c. Cylinder liner. The cylinder liner (Figure 3-11) is made of cast iron with a cored or hollow space in the wall through which cooling water is circulated. Water enters through

for the lower main bearing seats. The camshaft bearing lower seats are an integral part of the cylinder block. These bearing seats and their caps are match-marked and must be kept together.

Removable handhole covers close the

a synthetic rubber gasket sealed connection near the bottom of the cylinder and circulates out through similarly sealed steel ferrules into the cylinder head. The cylinder liner is held in the engine block by the lower deckplate and a

Figure 3-6. LONGITUDINAL CUTAWAY OF GM 16-278A ENGINE.

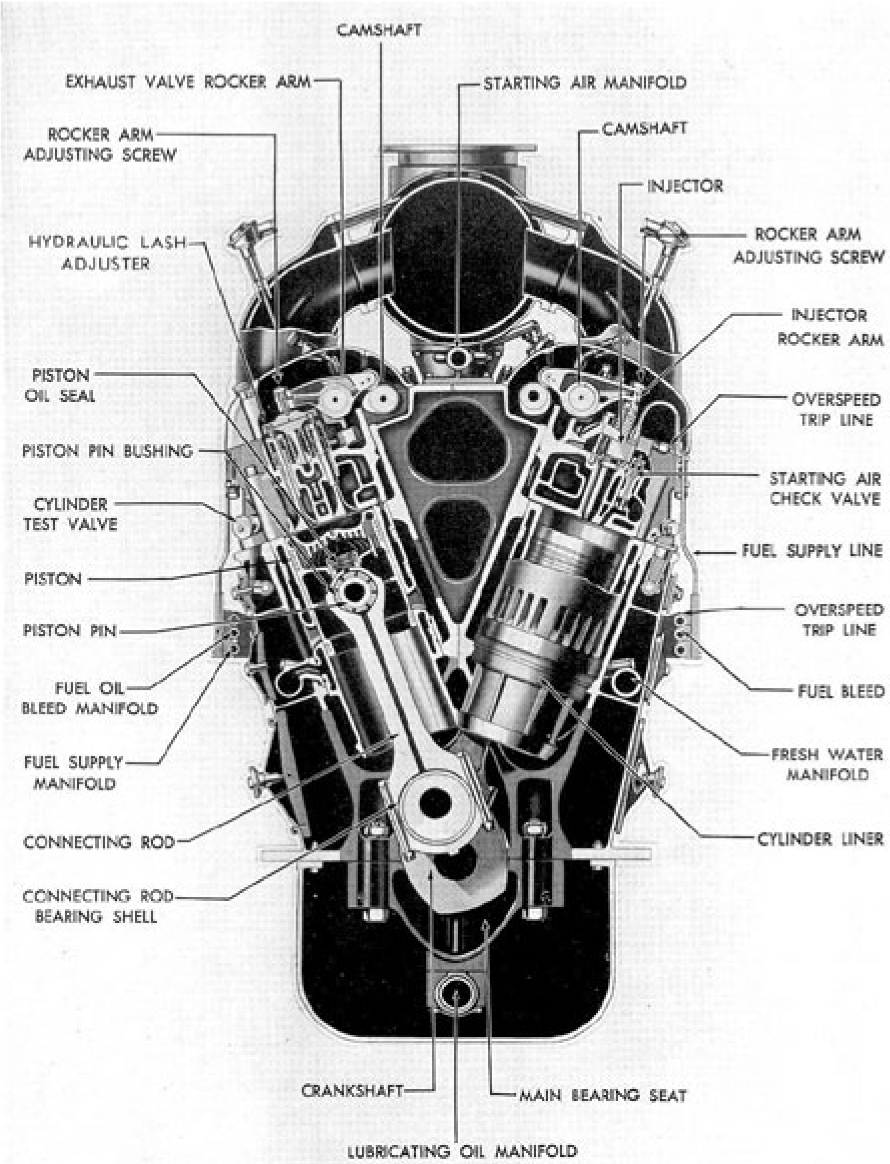


Figure 3-7. Cross section of GM 16-278A engine.

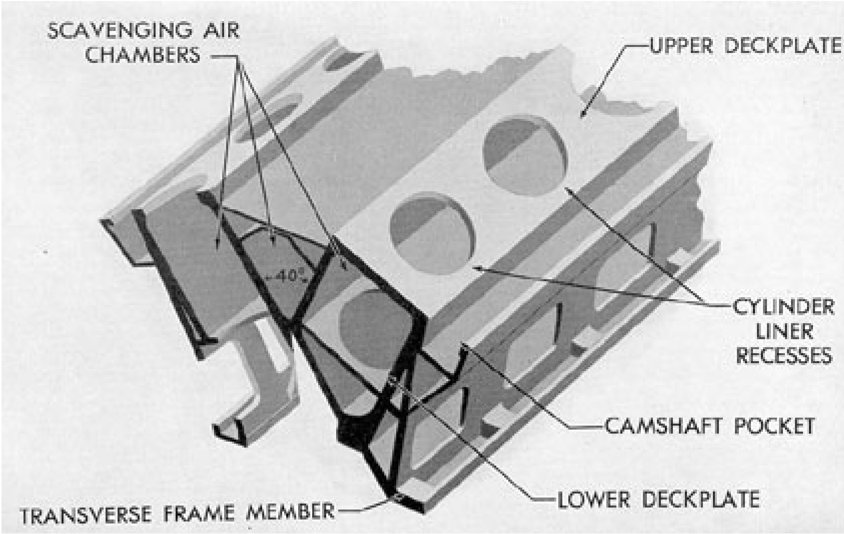


Figure 3-8. Section of cylinder block, GM.

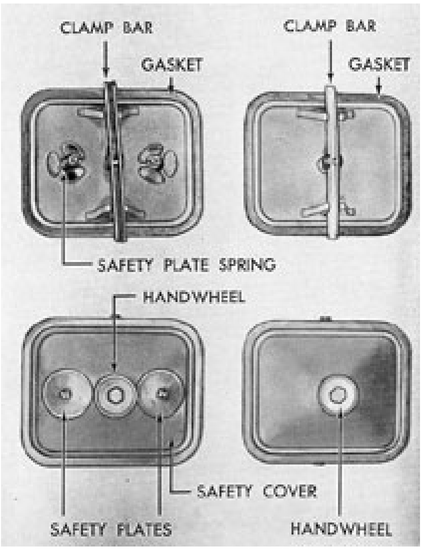


Figure 3-9. Crankcase handhole covers, GM.

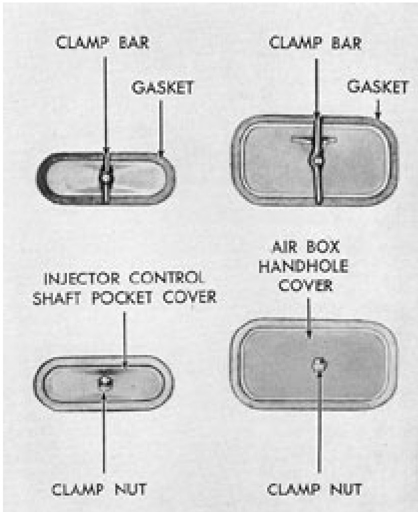


Figure 3-10. Injector control shaft and air box handhole covers, GM.

recess in the upper deckplate, and is held securely to the cylinder head by six steel studs and nuts The joint between the liner and the lower deck plate is made up with an oil-resistant seal ring made of neoprene which is compressed in a groove in the deckplate bore. This makes a tight joint and prevents the leakage of scavenging air from the air chamber and the leakage of oil from the crankcase into the air chamber. A solid copper gasket, slightly recessed

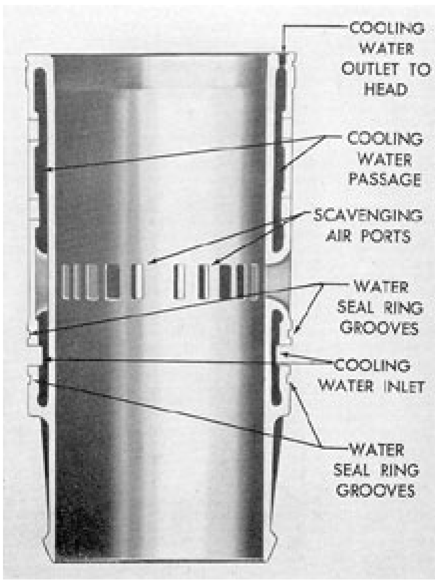


Figure 3-11. Cross section of cylinder liner, GM.

in a groove of the cylinder liner, seats against the cylinder head to form a pressure seal. Scavenging air intake ports are located near the center of the liner. They also serve as piston and ring inspection ports.

The distance from the upper ends of the scavenging air ports to the finished top of the cylinder liner must be closely held to the required dimension, so that the opening and closing of these ports by the travel of the piston are accurately timed in relation to the respective opening and closing of the exhaust valves.

In recent years it has been found that the wearing qualities of the liner can be greatly increased by chrome plating the inside of the liner. These chrome-plated liners are used in all late installations.

d. Cylinder head. The cylinder head attaches to the cylinder liner to form the top closure of the combustion chamber. It forms the support and houses the four exhaust valves, the unit injector, and the rocker lever assemblies. It also contains the overspeed injector lock, air starter check valve, cylinder relief valve, and cylinder test valve (Figure 3-12).

The cylinder head is an individual unit for each cylinder. It consists of an alloy iron casting, cored with water cooling passages. Cooling water flows from the cylinder liner through synthetic rubber sealed steel ferrules, and circulates through the cylinder head. It then passes through a watertight connection into the cylinder head is sealed against compression loss by a solid copper gasket which is slightly recessed in a groove of the cylinder liner. All other joints or openings of the cylinder head are made watertight or oiltight by gaskets.

3C3. Main moving parts.a. Crankshaft. The GM crankshaft (Figure 3-15) is an integral type, alloy steel forging, heat treated for stress and wear resistance, and dynamically and statically balanced. Shaft and crankpins are hollow bored to reduce weight and bearing load. The entire crankshaft is machine finished, and the main bearing and crankpin journals are precision ground. Crankshafts for right-hand and left-hand engines are interchangeable. There are eight cranks spaced 45 degrees apart and nine main bearing journals on each crankshaft. In both right-hand and left-hand engines, the cylinders are numbered from 1 to 8 inclusive in the right bank, and from 9 to 16 inclusive in the left bank. Cylinders 1 and 9 are at the blower end of each engine. Two pistons that are

water jacket of the exhaust elbow. All cylinder heads are equipped with a pressed steel or aluminum alloy cover secured by a handwheel nut. This cover has breather openings which serve as ventilating ports for the crankcase breather system. Each cylinder head is fastened to the cylinder block by four hold-down studs and nuts. The joint between the cylinder liner and

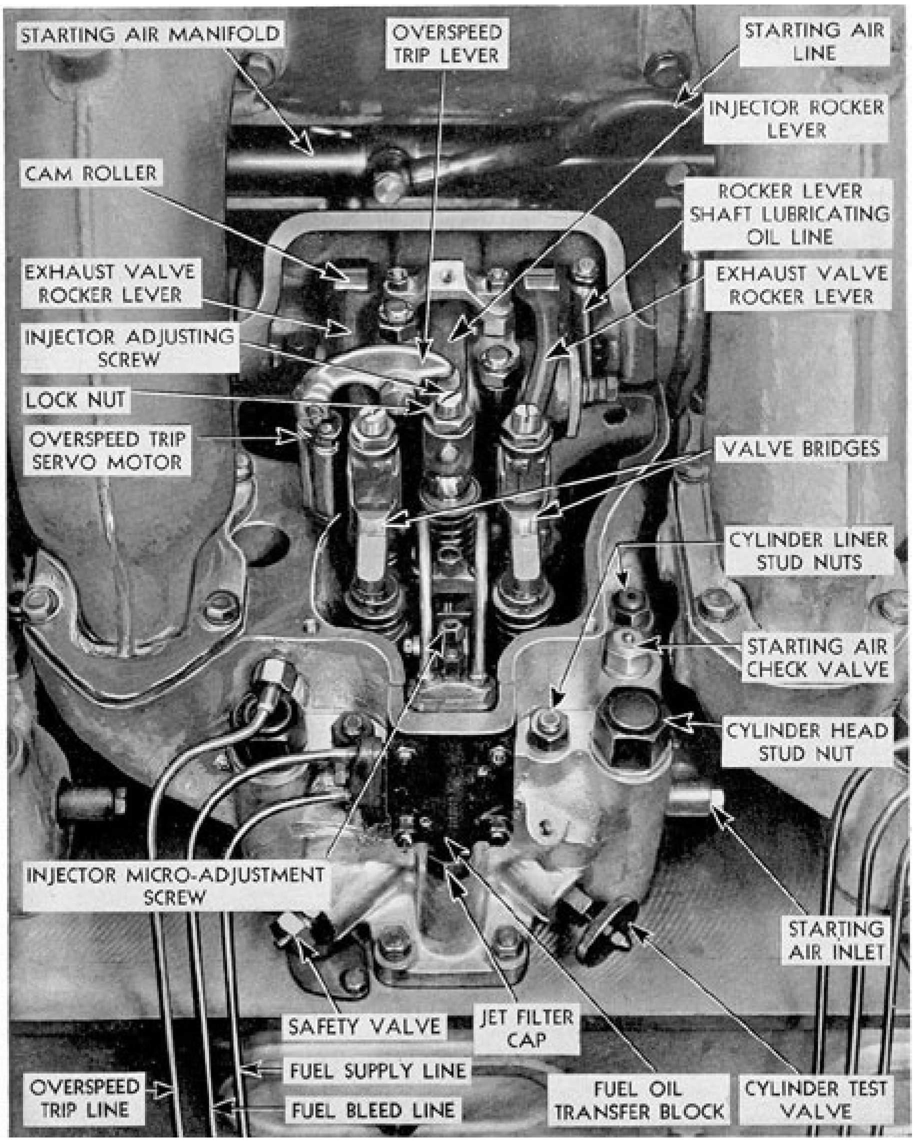


Figure 3-12. Cylinder head, GM.

opposite each other in the two banks are connected to each crank by connecting rods. Each

crank or crankpin is referred to by the numbers of the two cylinders to which it is related.

The firing interval is alternately 5 degrees and 40 degrees and these intervals are determined by the angle between the cylinder banks, which is 40 degrees, and by the relation of the crankpin positions of successively fired cylinders, which is 45 degrees. Two successively fired cylinders are connected either to two separate crankpins that are 45 degrees apart, or to one crankpin. When two successively fired cylinders have crankpins that are 45 degrees apart, which is 5 degrees greater than the bank angle of 40 degrees, the firing interval is 5 degrees. When two successively fired cylinders are connected to one crankpin, the firing interval is the same as the bank angle, which is 40 degrees.

Oil passages are drilled through each crankpin, crank webs, and main bearing journals, for lubricating oil to flow under pressure from the main bearings to the connecting rod bearings. The connection between the crankshaft and the main generator is by means of an elastic coupling.

b. Main bearings. The crankcase contains nine bearings (Figures 3-16 and 3-17) for the support of the crankshaft. Each main bearing consists of an upper and lower double flanged precision bearing shell. Two types of main bearing shells are used. One type is bronze backed with a centrifugally cast lining of high lead bearing metal known as

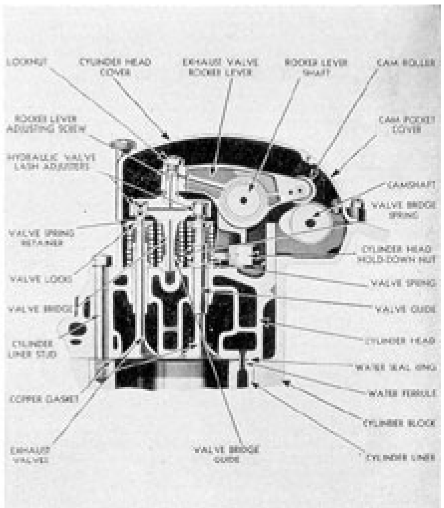


Figure 3-13. Cylinder head cross section through exhaust valves, GM.

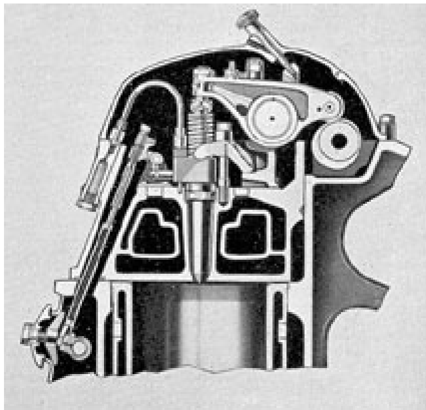


Figure 3-14. Cylinder head cross section through injector, GM.

Satco metal. The other type is steel backed with an intermediate lining of bronze and lined with Satco metal.

The bearings are carried in a steel bearing support and held by a steel bearing cap. Both bearing supports and bearing caps are made of drop-forged, heat-treated steel. Each of the bearing supports is secured to the main frame of the crankcase. Two large dowel pins locate the supports for perfect alignment.

The upper bearing shell is mounted in the bearing cap, the lower shell in the main bearing seat. The joint faces of the upper and lower bearing shells project slightly from the seat and cap. This is to insure that the backs of the shells will be forced into full contact when the cap is fully tightened. A drilled hole in the upper shell

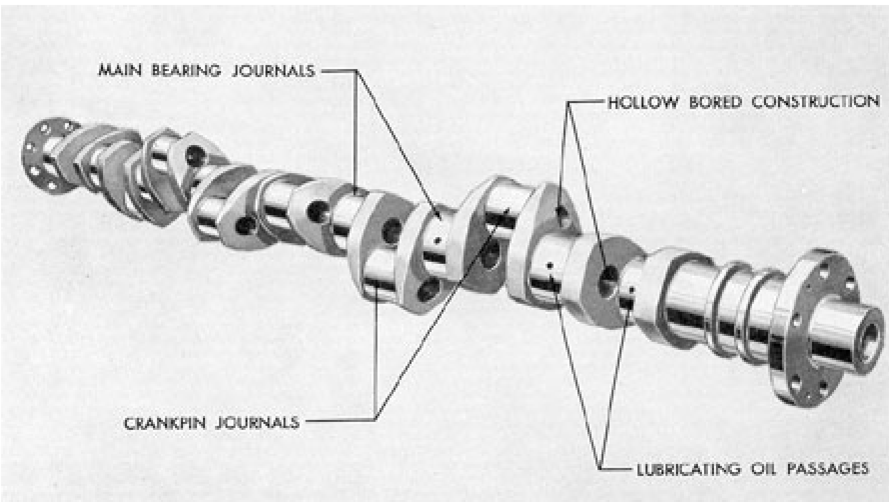


Figure 3-15. Crankshaft for GM engine.

fits on a dowel pin in the cap. The dowel pin locates the upper shell in the bearing cap and prevents both the upper and lower shells from rotating.

(No. 9) is the thrust bearing. Thrust bearing shells are the same as the other main bearing shells except that the bearing metal is extended to cover the flanges. With the exception of the thrust bearing, all

Bearing caps are held down on the bearings by jack screws locked with cotter pins. The jack screw fits into a recess in the arch of the crankcase frame and takes the upward thrust on the bearing cap. Close fit between shoulders on the crankcase frame prevents side play in the bearing cap. End play is controlled by two dowel pins. When the bearing supports and caps are assembled on the crankcase frames, the seats for the bearing shells are accurately bored in dine, and the ends of its faces are finished for a close fit between the bearing shell flanges.

Each bearing shell is marked on the edge of one flange. For example, the designation 2-L-B.E. indicates that the shell is for the No. 2 main bearing, that it is the lower shell, and that the flange of the shell thus marked should be placed toward the blower end of the engine. The main bearing nearest the blower end of the engine is the No. 1 main bearing. The rear main bearing upper bearing shells are alike and interchangeable before they are assembled and marked This is also true of the lower bearing shells. Upper and lower shells, however, are not interchangeable with each other.

Each lower bearing shell has an oil groove starting at the joint face and extending only partially toward the center of the bearing surface. The upper bearing shells are similarly grooved except that the groove is complete from joint face to joint face.

The main bearings are lubricated by oil under pressure received from the oil manifold under the bearing supports. The oil is forced up through a passage in the bearing support and through holes drilled in the lower bearing shell. From these holes, oil flows the entire length of the oil groove formed by the combined upper and lower shells. The oil lubricates the entire bearing surface and is carried off through the

drilled passages in the crankshaft to the connecting rod bearings.

c. Pistons and piston rings. The pistons for GM engines are made of cast iron alloy which is tin plated. Each piston is fitted with five compression rings at the upper, or crown, end and two oil control rings at the bottom, or skirt, end. In latest installations, the oil control rings are of the split type backed by expanders. of small oil grooves cut lengthwise in the bore and these receive lubricating oil that splashes from the sprayed head and side wall surfaces.

A cooling oil chamber is formed by an integral baffle under the piston crown. Lubricating oil under pressure flows from the top of the connecting rod, through a sealing member, and into the cooling chamber. The oil seal is a spring loaded shoe which rides on the

All piston rings are made of cast iron.

The bored holes in the piston pin hubs are fitted with hard bronze bushings which are cold shrunk in the piston bores. The outer ends of the bore for the piston pin are sealed with cast iron caps to prevent injury to the walls of the cylinder from floating piston pins.

The bores in the piston pin bushing are accurately ground in line for the close, but floating, piston pin fit. Each bushing has a number

cylindrical top of the connecting rod. The heated oil overflows through two drain passages.

d. Piston pins. The piston pin used on the GM engine is full floating, hollow bored, and case hardened on the bearing surface. The connection between connecting rod and the piston is by means of the connecting rod piston pin bushing. This bushing rotates freely inside the integral end of the connecting rod, and the connection is completed by pushing the piston

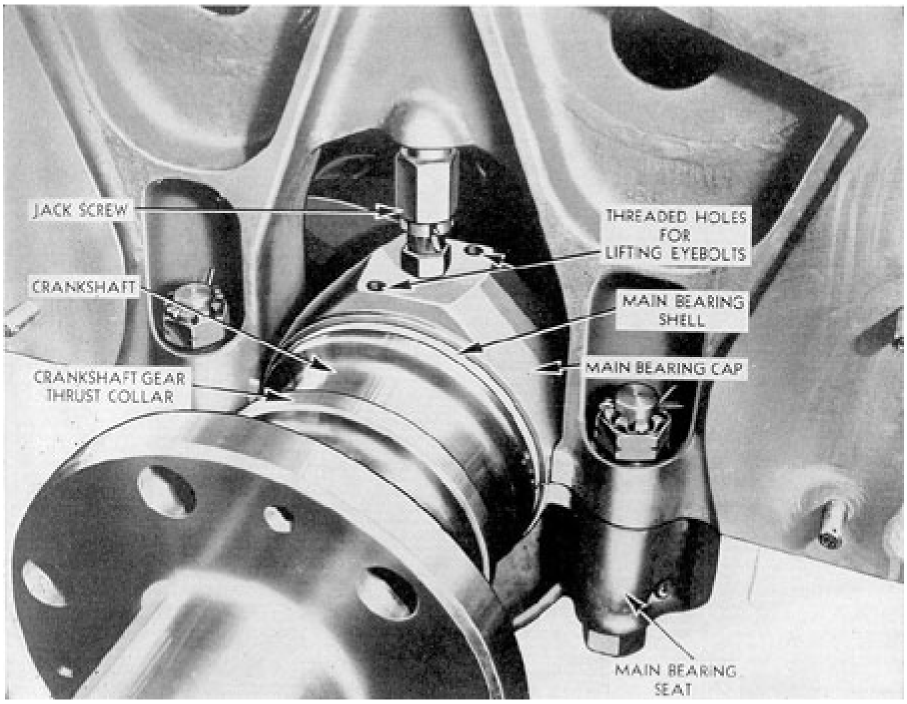


Figure 3-16. Main bearing cap installed, GM.

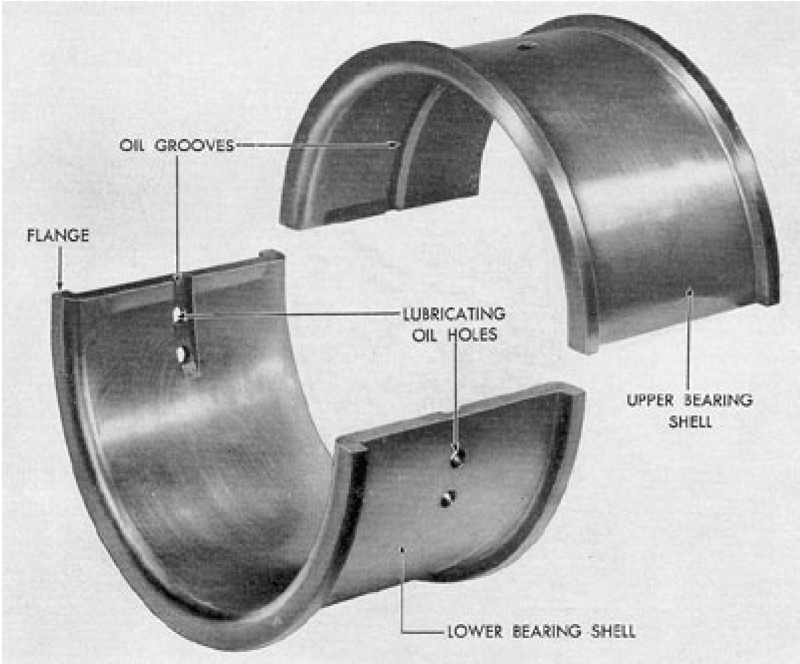


Figure 3-17. Main bearing shells. GM.

pin through the connecting rod piston pin bushing and the piston pin hub bushings.

In some older installations a needle type bearing containing three rows of 53 small roller bearings each was used instead of the connecting rod piston pin bushing. These have now been replaced by the bushing type of bearing.

The connecting rod piston pin bushing is constructed of steel-backed bronze. The entire length of the inner surface of the bushing is grooved to provide for lubrication of the piston pin assembly.

e. Connecting rods and connecting rod bearings. GM connecting rods are made of alloy steel forgings. The rod is forged in an I-section with a closed hub at the piston pin end and with an integral cap at the lower end. The cap is saw-cut from the rod in the machining operation. The cap is accurately located on the rod by two dowel pins. On the 16-248 the cap is fastened to the rod by four studs and castle nuts. For greater security, the studs are pinned in the rod. On the 16-278A the cap is fastened to the rod by four bolts with castle nuts. The crankpin bearing hub of the rod is turned to a lateral diameter which is smaller than the cylinder bore, so that the connecting rod will pass through the cylinder bore.

The connecting rod bearing is made up of upper and lower bearing shells. There are two types of connecting rod bearing shells used in the Series 16-278A engines. One type is bronze backed with a centrifugally cast lining of Satco metal of the same composition as that used in the main bearings. The other type is steel backed with an intermediate lining of bronze and an inner lining of the same bearing material. Connecting rod bearing shells are marked similarly

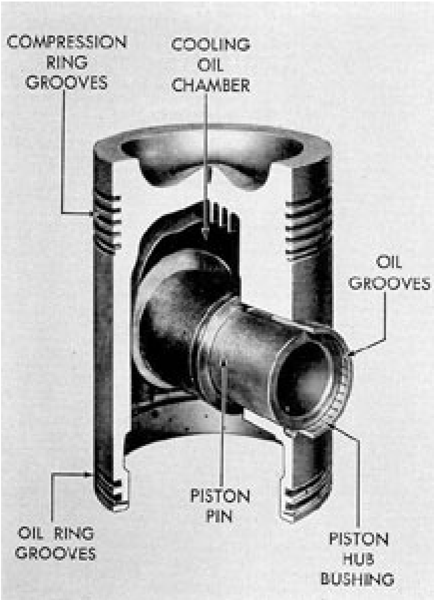


Figure 3-18. Cutaway of piston, GM.

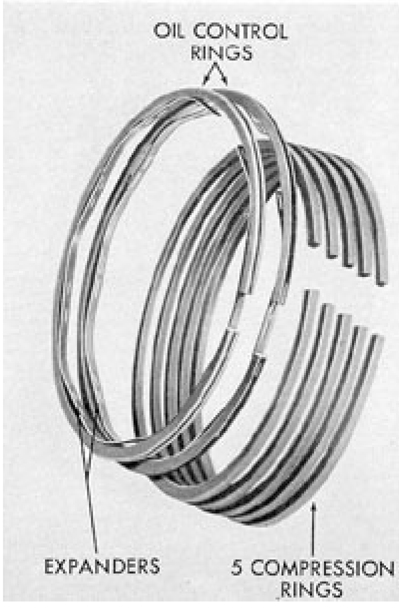


Figure 3-19. Piston rings, GM.

to main bearing shells to indicate their position in the engine.

In both types of bearings the lower bearing shell is located in the connecting rod bearing cap by means of a dowel pin. This pin prevents the lower shell from rotating. The joint faces between the upper and lower shells are compressed when the cap is fully tightened to make the joints oiltight and to force the backs of the shells into full bearing in their seats.

Each connecting rod bearing is lubricated with oil received from the adjacent main bearings through oil passages drilled in the crankshaft. The oil passage in the crankpin has two outlet holes in the connecting rod bearing that are 90 degrees apart, and from one or the other of these outlets, oil flows continuously into two grooves in the connecting rod bearing surface. These oil grooves are on opposite sides of the connecting

Two oil holes, drilled through the bearing shell, connect the upper end of each groove in the bearing surface with an oil groove in the upper part of the bearing shell seat in the connecting rod. An oil hole, which is rifle drilled through the center of the connecting rod, conveys the oil from the groove in the bearing shell seat to the piston pin end of the rod.

The upper and lower connecting rod shells now being manufactured are interchangeable. Any shell of present design may be installed either as an upper or lower. However, shells previously furnished were not interchangeable, and if not machined for interchangeability, must be installed in the correct position. Upper and lower shells of the old design must not be interchanged unless the shells have previously been machined to make them interchangeable.

3C4. Valves and valve actuating gear.a. Camshafts. There are two camshafts on the GM engine, one

rod bearing surface to insure a constant flow of oil regardless of the position and rotation of the crankshaft.

for each bank of cylinders. Each camshaft is made up of two sections which are

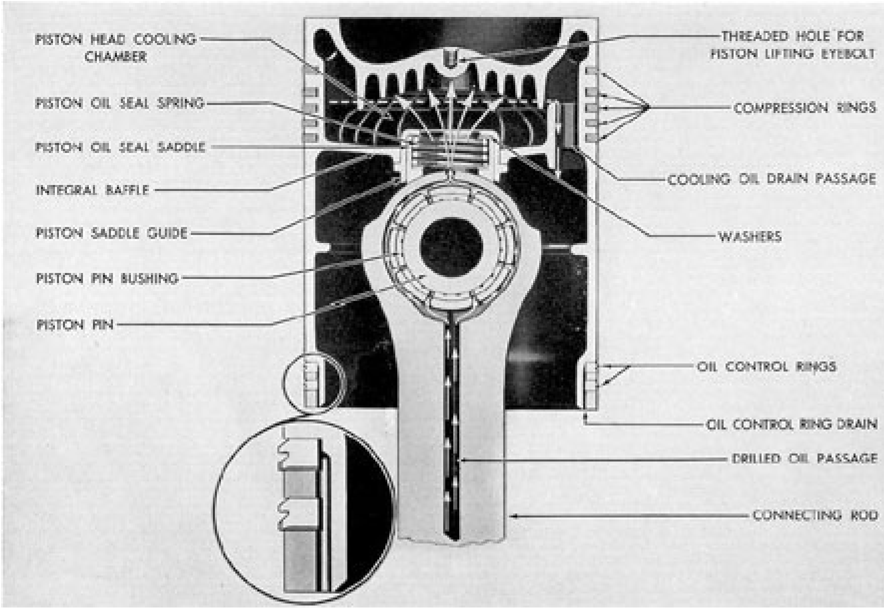


Figure 3-20. Cross section of piston showing cooling and lubrication, GM.

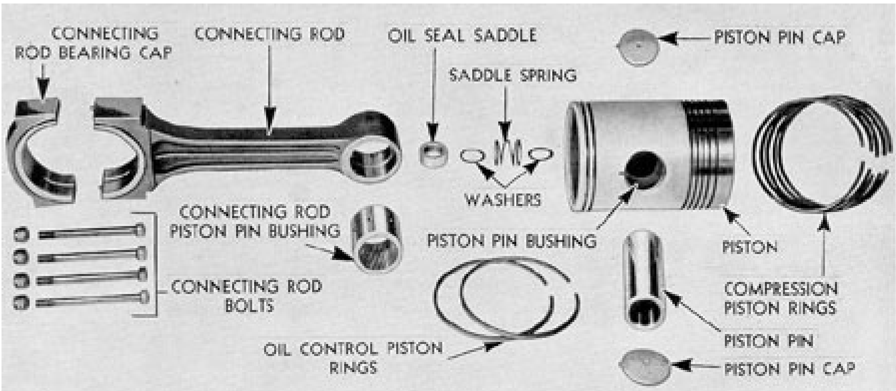


Figure 3-21. Piston and connecting rod disassembled, GM.

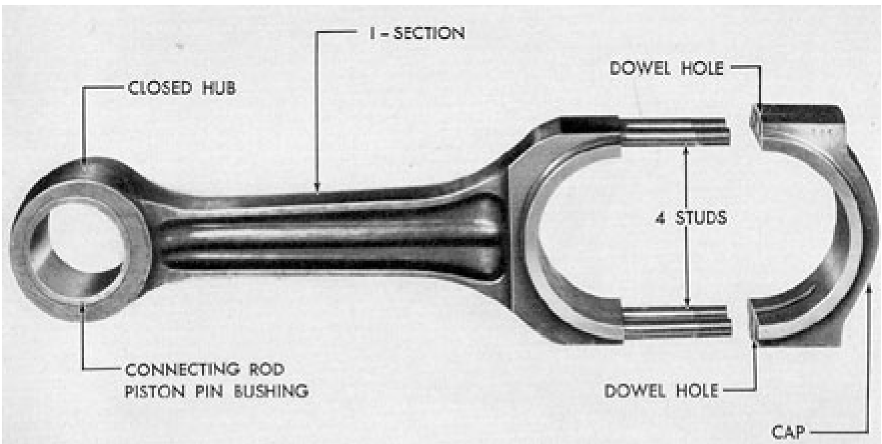


Figure 3-22. Connecting rod, GM 16-248.

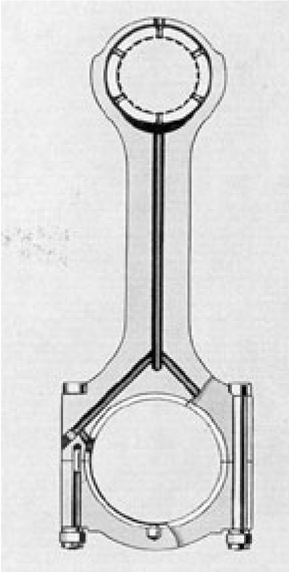


Figure 3-23. Connecting rod oil passages, GM 16-278A.

Figure 3-24. Connecting rod bearing shells, GM.

flanged and bolted together. The sections are accurately centered in relation to each other by means of a key in one section, which fits in a recess in the other section. Each flange coupling is made up with eight bolts, even of which serve as driving dowel pins, and one of which is smaller than the others to insure the correct angular matching of the shaft sections. The cams are case hardened and are an integral part of each shaft section. There are three large cams on the shaft for each cylinder. Of these, the two outer cams operate the exhaust valves, and the center cam operates the unit injector. The narrow cams located between the cylinder cam groups operate the air starting distributor valves.

Each camshaft is supported in 16 bearings in the cam pocket on the cylinder block. The bearing bases are integral with the cam pocket and have forged steel caps. The bearings consist of

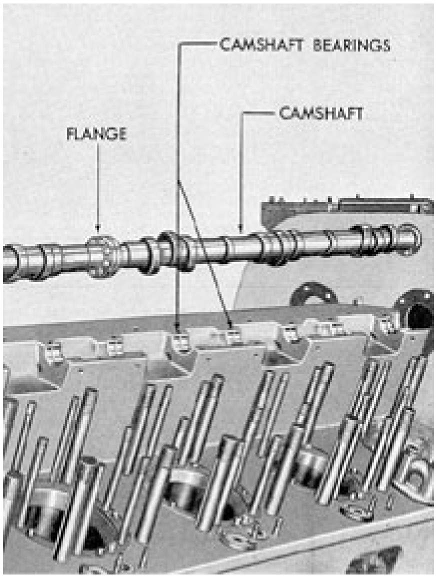


Figure 3-25. Camshaft, GM.

Bushings are pressed into the lever hubs and are reamed for the bearing fit on the rocker lever shaft.

The roller follows or rolls with the cam on the camshaft. The high point on the cam forces the roller end of the rocker lever up and the opposite end down. It is this motion that actuates the valves and injector. Each of the exhaust valve rocker levers is fitted at the outer end with a nut-locked, adjusting screw that has a ball

upper and lower shells with flanged steel backs and babbitt linings. The upper shell of each bearing is held from turning by a dowel pin in the bearing cap.

Each of the two camshafts is bolted and doweled to a camshaft driving sleeve at the drive end of the engine. The sleeve in turn is driven by the camshaft gear of the camshaft drive gear train. The camshaft thrust is taken at the camshaft gear.

The camshaft bearings in each bank are lubricated by oil piped from the main lubricating oil manifold to the camshaft gears. The oil flows under pressure through a passage in each driving sleeve to the hollow bore in the camshaft and then through radial drilled holes to each bearing on the camshaft. Tubes from the camshaft bearing caps carry the oil to the cam pockets. The cam pockets provide a reservoir into which the cams dip, insuring lubrication as soon as the engine is started.

b. Rocker lever assembly. Each cylinder head is equipped with three rocker levers; two of them operate the two pairs of exhaust valves, the third operates the unit injector. All three are made of alloy steel forgings. The rocker levers rock up and down in a fixed shaft which is clamped in a bearing support. They are fitted with cam follower rollers which operate in contact with the exhaust and injector cams.

point. The ball point fits into a ball socket on the exhaust valve bridge. Thus, the downward pressure on the rocker lever end is transmitted to the valve bridge which actuates a pair of exhaust valves.

The injector rocker lever is fitted at the outer end with a nut-locked adjusting screw having a ball socket at the end. A hardened steel shoe fits around the ball socket to give flexibility of movement. Downward pressure of the rocker lever end causes the shoe to bear down on the plunger follower in the injector. The rocker lever assemblies are lubricated

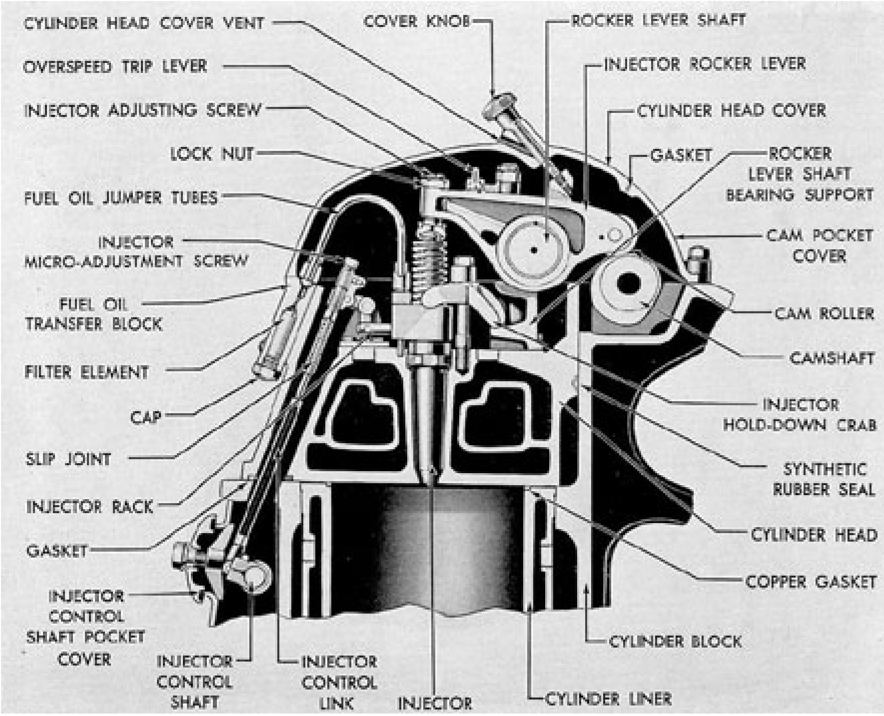


Figure 3-26. Cross section of cylinder head through injector, GM.

through oil pressure tubes leading from the camshaft bearings, through the endplate, and to the hollow bore in the rocker lever shaft. The oil is forced through holes in the rocker lever shaft to the rocker lever hub bearings. From the hub bearings, it is conducted through drilled passages and holes to the bearings of the cam rollers and the tappet mechanism on the injector rocker lever.

c. Exhaust valves and valve bridges. Each cylinder contains four exhaust valves. The valves are operated in pairs by the rocker levers through the valve bridges.

The exhaust valves are made of special analysis, heat-resisting, alloy steel. They are held in operating position by cast iron valve stem guides. Valve springs secured to the ends of the valve stems by locks draw the valve heads tight on the valve seats of the cylinder head.

The valve bridges are made of forged steel and have a hardened ball socket into which fits the ball end of the adjusting screw on the rocker lever. The valve bridge has two arms, each of which extends over an exhaust valve stem.

Each arm is fitted with an adjusting screw at the valve stem to equalize valve clearance. The lower part of the valve bridge is ground for a sliding fit in the valve bridge guide. This guide has a ball and socket bearing in the top of the cylinder head. The valve bridge spring keeps valve bridge tension off the valve stems until the bridge is actuated by the rocker lever. When the valve end of the rocker lever is pressed down by the cam action, the valve

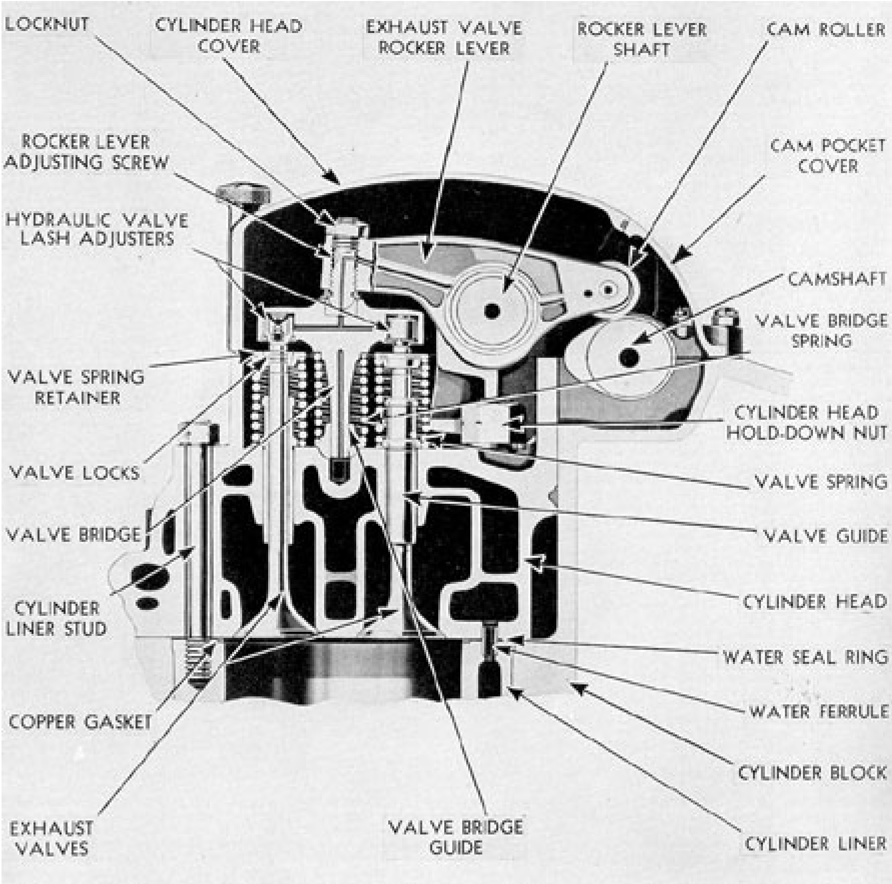


Figure 3-27. Cross section of cylinder head through exhaust valves, GM.

bridge and valve springs are compressed and the valves open. As the cam action passes, the springs force the valves closed.

The ball and socket bearings in the valve bridges and the valve stems are lubricated by the oil spray that is thrown off by the rocker lever.

Clearances between the valve bridge adjusting screws and the valve stem caps are adjusted by loosening the lock bolts and turning the adjusting screws. A lock wire in the counterbore of the spring seat at the upper end of the valve stem prevents accidental separation of the spring seat from the cap and the split spring lock from the valve stem. If a valve spring breaks, these assembled parts are held together so that the valve does

head is machined to fit the valve seat and opens or closes a passage leading from the combustion chamber to the outside of the cylinder. The valve face is held against the valve seat by a pressure spring. Tension on the spring is varied with an adjusting nut and locked when the desired setting is attained. This setting varies with the type of engine and may be found by referring to manufacturers' instruction books. If the pressure in the cylinder exceeds that set on the valve spring, the valve will open and remain open until the pressure in the cylinder is less than the spring pressure, at which point the valve will close.

not drop into the cylinder. The lock-wire also guards against accidental removal of the cap when the rocker lever is not in place.

d. Cylinder test valve. The cylinder test valve is located in the cylinder head and is made up of a valve body which is screwed into the cylinder head, and a valve stem which has a threaded fit in the body and a handwheel at the outer end. The valve itself has two faces, an inner face and an outer, or secondary, face. From the valve seat two passages are bored in the cylinder head casting, one leading to the inside of the cylinder and the other leading to the outside. This outside connection is fitted with an indicator adapter which is used when a pressure indicator reading is taken of hot or cold compression pressure. When the handwheel is in the closed position, the inner valve face seats against the main valve seat, closing the passage to the combustion chamber, and preventing the pressure in the cylinder from escaping to the outside. If the handwheel and valve stem are open, the passage to the outside is connected to the passage to the inside of the cylinder. When the valve stem is at its full open position, the outer or secondary valve face bears against the valve body, thus preventing the passage of exhaust gases through the valve body.

e. Cylinder relief, or safety valve. Each cylinder head is equipped with a safety valve (Figure 3-29) which opens if the cylinder

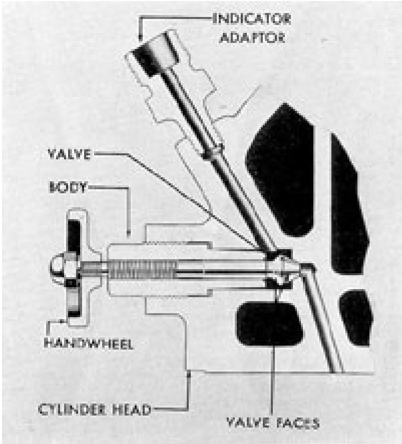


Figure 3-28. Cylinder test valve, GM.

f. Camshaft drive. The camshafts are driven from the control end of the crankshaft through a train of helical spur gears, with a crankshaft idler gear and a camshaft idler gear between the two camshaft gears and the crankshaft gear. The camshafts run at the same speed as the crankshaft but in the opposite direction of rotation. The drive gear for the lubricating oil pump is driven from the left bank camshaft gear in a left-hand rotation engine and from the right bank camshaft gear in a right-hand rotation engine. All of the other gears are in the same location regardless of rotation. These gears are made of steel forgings.

pressures exceed a safe operating limit. This valve

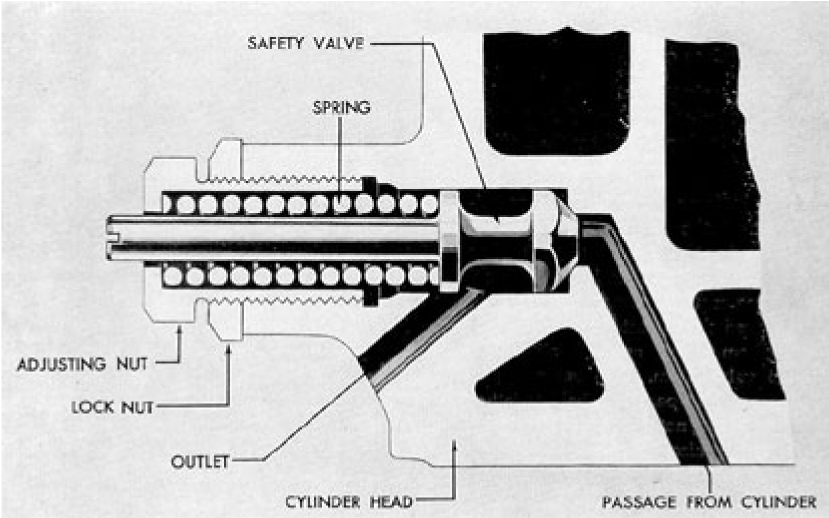


Figure 3-29. Cylinder relief or safety valve, GM.

The split crankshaft gear is mounted loose on the crankshaft and held together with clamping bolts. The bore of the crankshaft gear is babbitted and a circumferential groove in the bearing forms the thrust surfaces which bear against a collar on the crankshaft. The crankshaft gear is driven through a spline ring on the elastic coupling.

Each of the two idler gears and the lubricating oil pump drive gear are mounted on a heat-treated steel shaft, which is pressed into the gear hub. The two idler gear shafts are supported in inner and outer bearing supports fitted with single-flanged steel bushings, which are lined with babbitt. The bearing supports are accurately aligned with dowel pins and fastened together with studs. The pump drive gear is supported in the bearing supports of the mating camshaft gear.

and bearing support assemblies are located accurately in the camshaft drive housing with dowels and fastened with studs.

The outer flange of each camshaft driving sleeve is fastened to the outer face of the camshaft gear hub by capscrews. The inner end of the driving sleeve is flanged and doweled to the flanged end of the camshaft. The camshaft is driven through the dowel pins in the connection, and a bolt, smaller than the dowel pins, prevents incorrect assembling of this drive connection. The holes in the outer flange of the driving sleeve are slotted, so that the camshaft may be accurately adjusted to the correct timing position. When this adjustment has been made, the timing position is permanently fixed by dowel pins, through which the driving sleeve and the camshaft are driven.

Oil for lubricating the gear teeth and the gear bearings is received from two oil-distributing blocks in

The hub projections on the outside of the camshaft gears are finished to form journals, and are supported in babbitt-lined steel bushings which are pressed in the inner and outer bearing supports. The inner and outer bearing supports are accurately aligned with dowel pins and are fastened together with studs. The gear

the camshaft drive housing. The two distributing blocks are supplied with oil from the main manifold in the oil pan. The engaging gear teeth are lubricated with jets of oil delivered through tubes and nozzles. The outer bearings of all the gears, except the

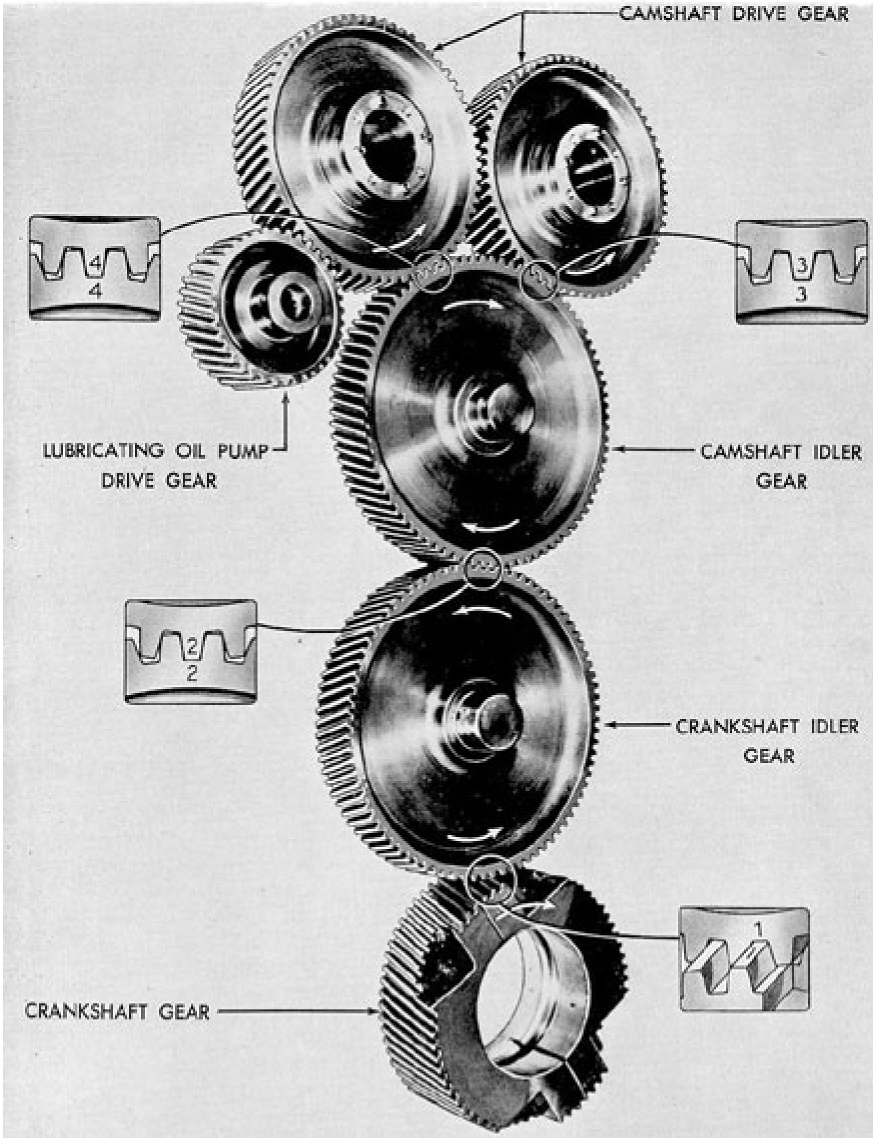


Figure 3-30. Camshaft drive gears, GM.

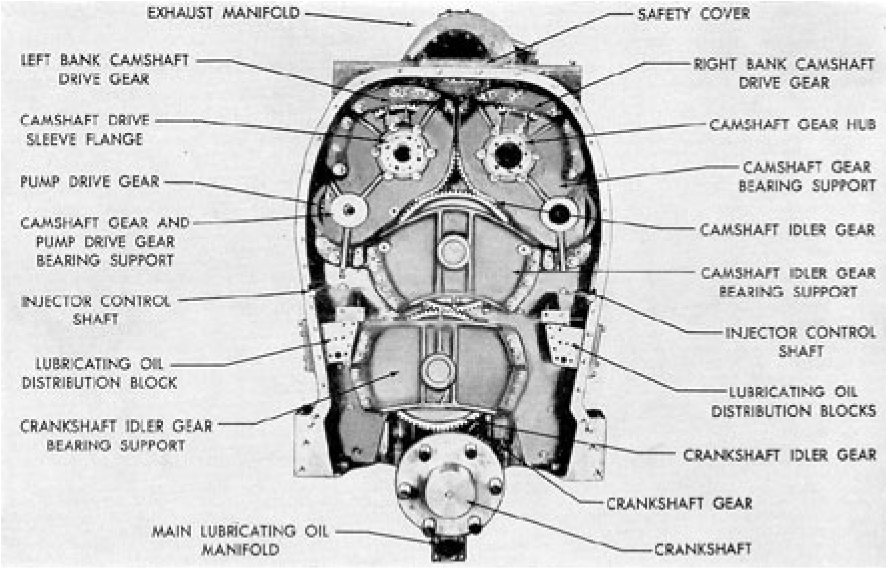


Figure 3-31. Camshaft drive assembly, GM.

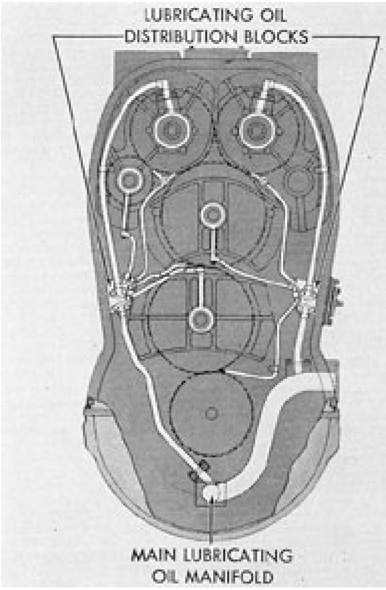


Figure 3-32. Camshaft drive lubrication, GM.

crankshaft gear, receive lubricating oil through tubes and drilled holes in the outer bearing supports. The inner bearings are lubricated with oil that is received from the outer bearings, through holes drilled in the gear hubs.

The gear train is enclosed in an oiltight housing. The housing is accurately located on the end of the crankcase with dowel pins and is held in place with studs, some of which secure both the housing and the gear assemblies. A pressure relief opening in the top of the housing is fitted with a spring-loaded plate.

g. Accessory drive. The accessory drive is located on the blower end of the engine and consists of a train of helical gears transmitting the rotation of the crankshaft to the blower and water pumps. The gears are enclosed in a case bolted to the blower housing.

The blower and accessory drive gear, which drives the water pump idler gears and the blower drive gear, is driven from the crankshaft through a splined shaft, one end of which fits into a hub that is bolted to the crankshaft, while the

other end fits into the blower drive gear hub. The water pump drive gears are driven by

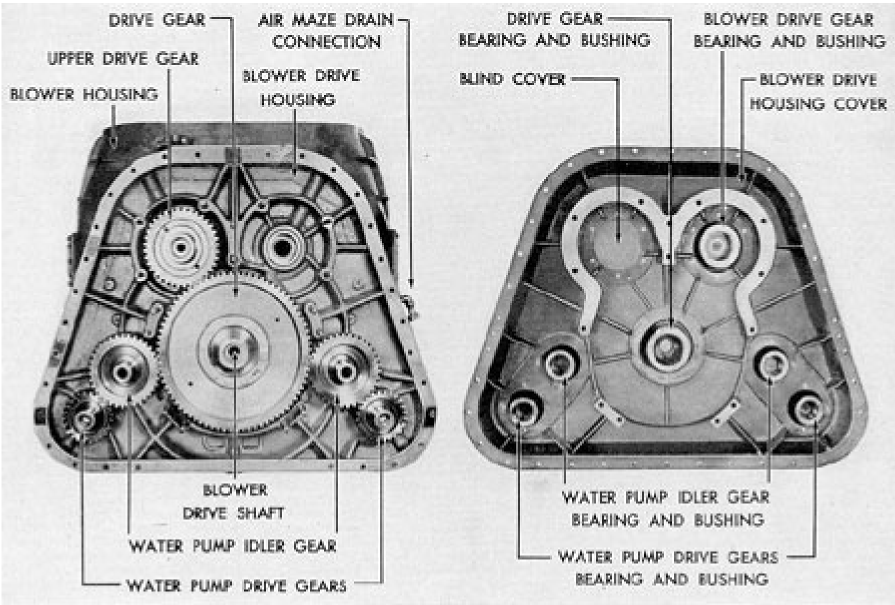


Figure 3-33. Accessory drive assembly with cover, GM.

the idler gears. All gears are steel forgings and have integral shafts. The inner gear bearings are babbitt lined, with integral thrust faces and are pressed into the housing. The outer bearings are pressed into the housing cover plates, which are bolted to the accessory drive cover.

Oil for lubricating the gear bearings is received from a manifold bolted to the main lubricating oil manifold, which carries oil to passages formed by steel tubing cast into the ribs of the drive housing. The outer bearings are lubricated by oil flowing through passages in the gear hubs.

D. FAIRBANKS-MORSE ENGINE COMPONENTS

3D1. General. Descriptions of engine components in this section apply only to the Fairbanks-Morse 9- and 10-cylinder engines.

3D2. Main stationary parts. a. Cylinder block. The cylinder block is the main structural part of the engine and is designed to give the engine the necessary strength and rigidity. It is constructed by welding various structural members and bracings horizontal decks are bored to receive the cylinder liners along the axis of the engine.

The cylinder block consists of the following compartments:

1. Control end compartment, forming an enclosure for the timing chain, controls, and flexible gear drive for the attached pumps and governor.

2. Vertical drive compartment, forming the enclosure for the

into one unit. The transverse vertical members together with four horizontal decks farm the enclosures and housings for the various operating or functional parts. The four bearing assembly housings of the vertical drive shaft connecting the upper and lower crankshafts.

3. Upper crankshaft compartment,

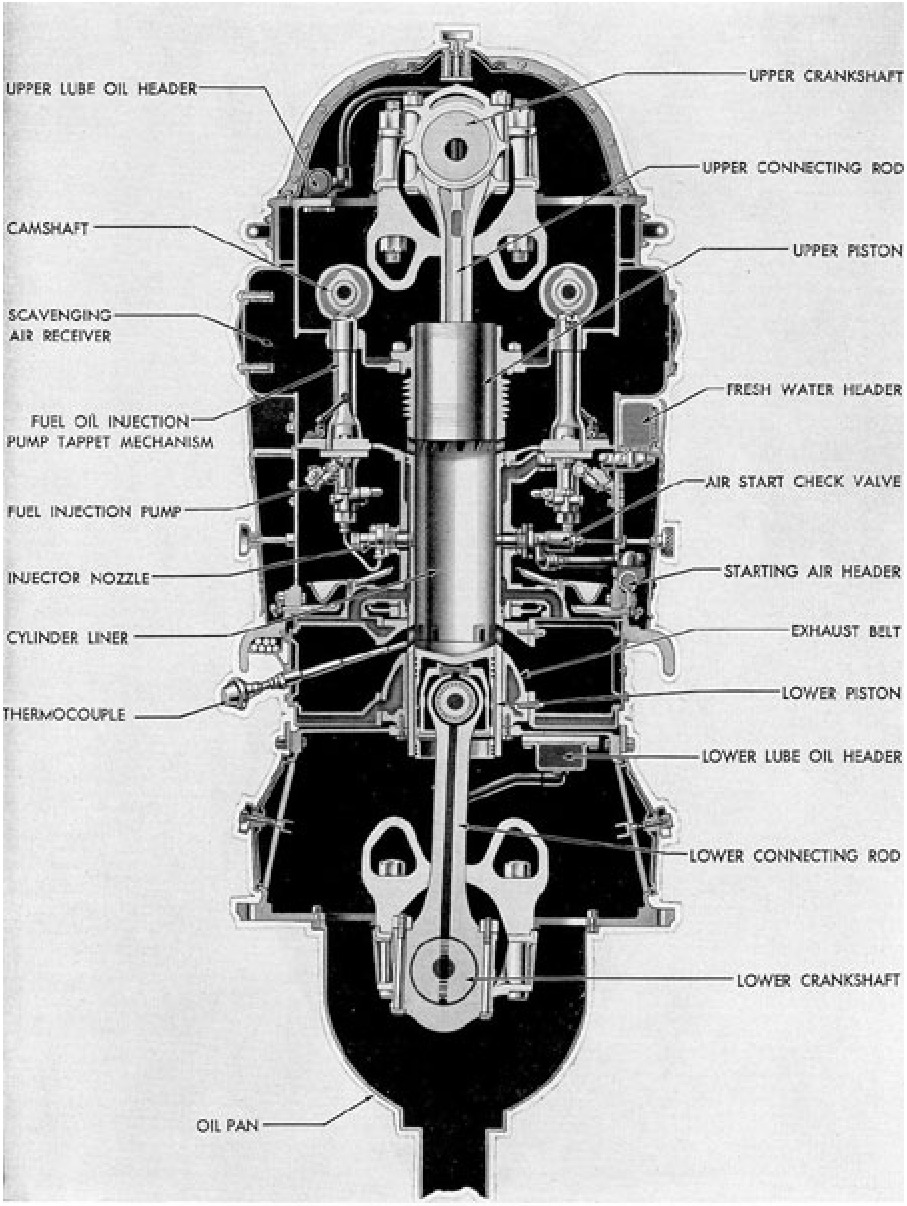


Figure 3-34. Cross section of F-M 38D 8 1/8 engine.

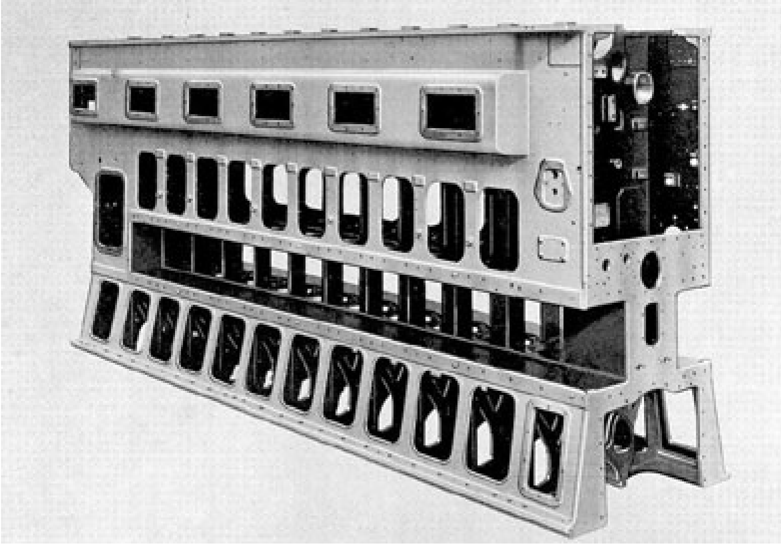


Figure 3-35. Cylinder block, F-M.

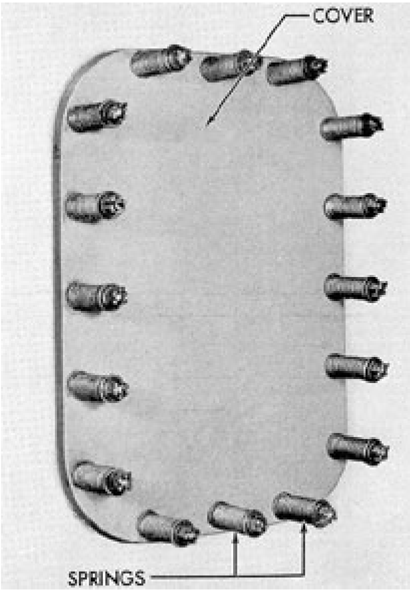


Figure 3-36. Vertical drive compartment spring-loaded access plate, F-M.

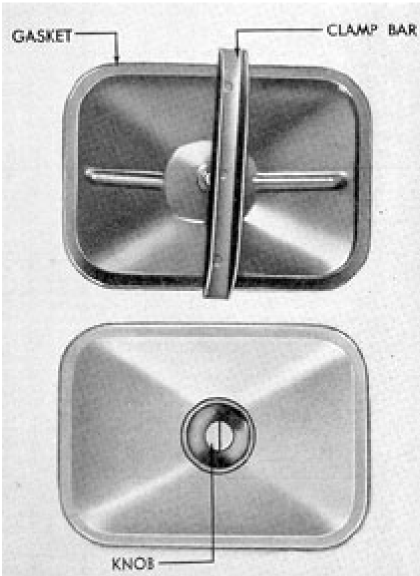


Figure 3-37. Inspection covers, F-M.

forming the bearing saddles for the upper crankshaft bearings and hubs for the bearings of the two camshafts.

4. Scavenging air compartments and air receivers running lengthwise on each side of the cylinder block, forming a passage for scavenging air to the inlet ports of the cylinders.

5. Valve compartments, forming enclosures for the injection

electric furnace to remove most of the internal strains introduced by welding. Lastly, it is magnafluxed to check the welding at all welded joints.

The air receiver, vertical drive, and control end compartments are provided with covers. The upper crankcase compartment is closed with a sheet metal top cover having several small inspection covers over the cylinders. These inspection covers are spring

nozzles, injection pumps, air start check valves, cylinder relief valves, and governor control shafts.

6. Exhaust manifold and belt compartment, extending lengthwise on each side of the cylinder block. With the installation of the exhaust belt and two exhaust manifolds in this compartment, a passage is formed for the exhaust gases from the cylinders to the external exhaust system.

7. Lower crankcase compartment, forming the bearing saddles for the lower crankshaft bearings.

The block is sand blasted after welding. It is then stress relieved by seasoning in an loaded so that in an emergency undue pressure in the crankcase compartment will be relieved. One of the vertical drive compartment access plates is spring loaded for the same purpose.

b. Cylinder liner. The cylinders are bolted into the cylinder block in a row along the centerline of the engine. They are spaced so that the lower end will enter the bored hole in the exhaust belts. The spacing must be horizontally correct so that the pistons and connecting rods coincide with the throws of the crankshafts. No. 1 cylinder is always at the control end of the engine.

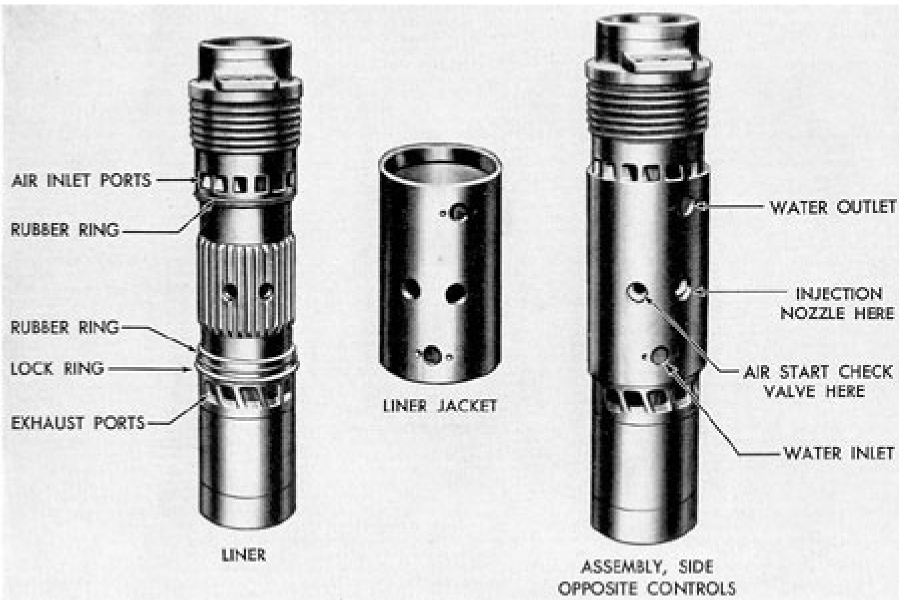


Figure 3-38. Cylinder liner, F-M.

The complete cylinder consists of an inner cast iron liner fitted into a steel jacket. The jacket extends over the high-pressure, high temperature part of the liner and helps to reinforce the area of greatest stress. Between the iron, dynamically balanced. The lower crankshaft is connected to the generator by means of the crankshaft flexible coupling. The upper crankshaft is connected to the lower crankshaft by a vertical drive shaft assembly and bevel

inner liner and the jacket is a space for cooling water. Cooling water enters through an elbow connection near the bottom on each side of the steel jacket and leaves through a pipe connection near the top of the steel jacket. The upper and lower circumferences of the water cooling passage between the liner and the jacket and the pipe connections at the inlet elbows of the liner are made watertight with synthetic rubber seal rings. A lock ring is also installed to position the steel jacket over the liner and prevent any movement between the liner and jacket due to expansion from the heat of engine operation.

The tangentially shaped scavenging air inlet ports are located near the top of the liner and are opened and closed by the upper piston. The exhaust ports are located near the bottom of the liner and are opened and closed by the lower piston. Each cylinder liner has four valve ports bored near its center for two injection nozzles, an air start check valve, and a cylinder relief valve with indicator cock which are adapted together.

Circular ribs or radiating fins are provided near the top of the liner to allow the scavenging air to carry away some of the heat of combustion. Vertical ribs in the liner between the inlet and exhaust ports direct the water travel upward, absorbing heat from this part of the cylinder. The liner is bolted to the top deck of the cylinder block by means of lugs. The liner is held rigid at this point and any gears. As the lower crankshaft leads the upper crankshaft by about 12 degrees, it is found that the lower cylinders develop about 72 percent of the power at rated load and the upper cylinders about 28 percent of the power. As the upper crankshaft also drives the scavenging air blower and other auxiliaries, a relatively small percentage of the total power is transmitted from the upper crankshaft through the vertical drive shaft to the lower crankshaft.

Both crankshafts on the 10-cylinder engine have ten cranks. The 9-cylinder engine has two crankshafts, each having nine cranks. Main bearing and connecting rod journals are stone ground to a smooth finish. Weight and bearing loads are reduced by hollow casting the shaft and crankpins. Oil passages are drilled so as to permit lubricating oil to be forced from each main bearing journal to the adjacent crankpin journals.

The crankshaft sprocket for the timing chain drive is keyed to the upper crankshaft at its control end. The air start distributor camshaft is also fastened to the upper crankshaft at the control end. At the opposite or blower end, the blower flexible drive gear is keyed and held with a retainer plate to the crankshaft.

The torsional damper is keyed to the control end of the lower crankshaft and secured by means of a key and a damper hub nut. The flexible pump drive gear for driving the governor and attached pumps is keyed to the torsional damper spider. The flexible crankshaft coupling driving gear is

expansion of the liner due to the heat of combustion is downward through the counterbores of the engine framing and exhaust belts. Tapped holes for lifting eyebolts are also provided in the lugs that bolt the liner to the cylinder block.

3D3. Main moving parts, Fairbanks-Morse. a. Crankshafts. Each Fairbanks-Morse engine has an upper and a lower crankshaft. The upper pistons are connected to the upper crankshaft and the lower pistons are connected to the lower crankshaft. Both crankshafts are of the integral type, constructed of machined, fine grain cast bolted to a flange on the blower end of the lower crankshaft.

b. Main bearings. Main bearings in the upper and lower crankcase support the upper and lower crankshafts. Each main bearing consists of an upper and lower precision made bearing shell that is lined with Satco metal. The upper and lower shells fit into the enclosures formed by the saddles or bearing seats in the cylinder block and the bearing caps. The bearing caps are made of forged steel. They are assembled with the bearing saddles in the cylinder

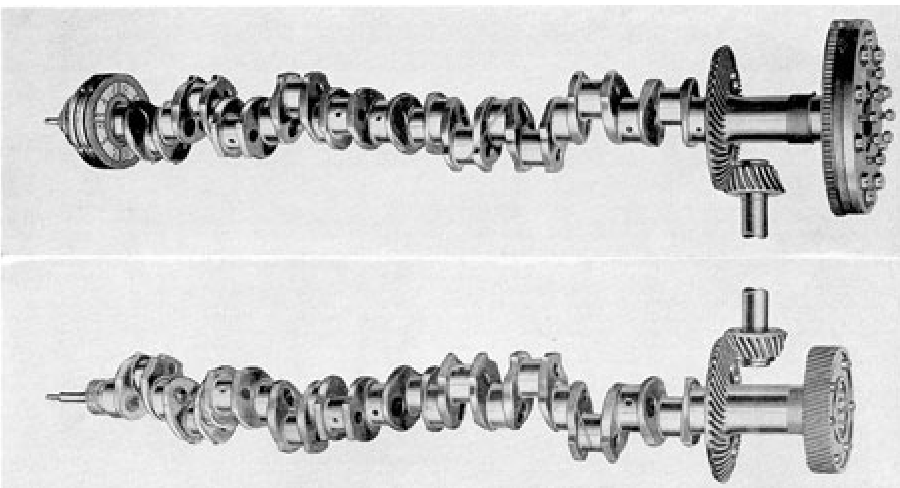


Figure 3-39. Upper and lower crankshafts, F-M.

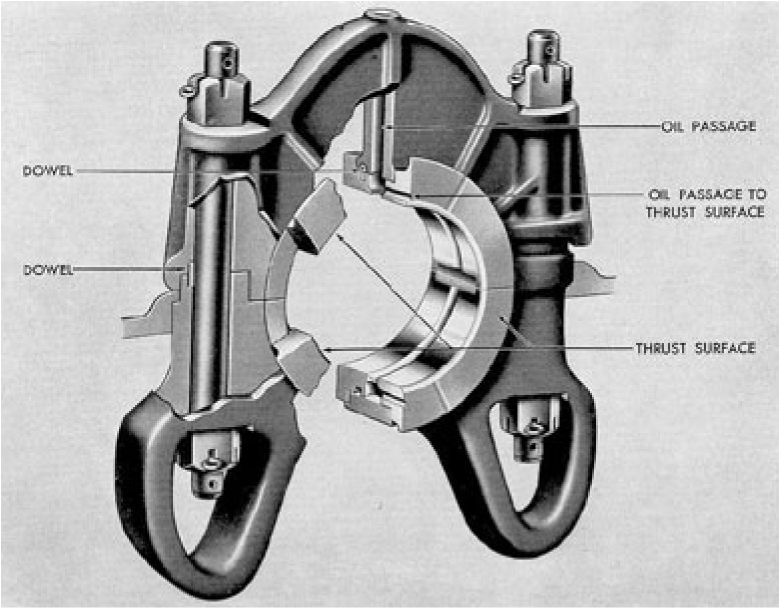


Figure 3-40. Upper crankshaft thrust bearing, F-M.

block and bored in line to give precision alignment and close fit to the bearing shells. Both bearing caps and saddles are finished for a close fit and form the bearing seats of the bearing shells. The bearing caps are located in the cylinder block by dowels and held by two bolts with castle nuts and cotter pins on each end. The locating dowels also prevent side and end play in the bearing cap. The upper and lower bearing shells are doweled together and marked for proper location on the edge toward the control end of the engine. The bearing shells housed in the bearing caps have dowel pins that prevent the bearing shells from rotating.

Both upper and lower bearing shells have oil grooves around the center of the inside surface. The bearings are lubricated by oil under pressure from the engine pressure system. The oil is piped to the bearing caps through lines from the main oil header and fed through holes thrust bearing located at the blower end of each crankshaft. The bearing shells of the thrust bearings are similar to the regular main bearing shells except that they have enlarged flanges with bearing metal extending over the flanges to take the thrust. Slots and a drilled passage conduct oil to the thrust surfaces.

c. Torsional damper. Every crankshaft with attached rotating parts has a natural period of torsional vibration, the frequency of which depends upon the mass and elasticity of the shaft and of the parts attached to it. If turning impulses are applied to the shaft at regular intervals, and if their frequency of application is a multiple of the natural frequency of the shaft, a condition of synchronous torsional vibration is produced. This condition is not usually found in the F-M 9-cylinder engine but is definitely present in the F-M 10-cylinder engine. The points at which it occurs are known as the critical speeds.

into the grooves where it lubricates the bearing surface. Oil is conducted through oil passages in the crankshaft to the connecting rod journals and bearings. Crankshaft thrust is taken by a

If the engine is to be permitted to run at one or more of these critical speeds, a means of damping the torsional vibration is advisable.

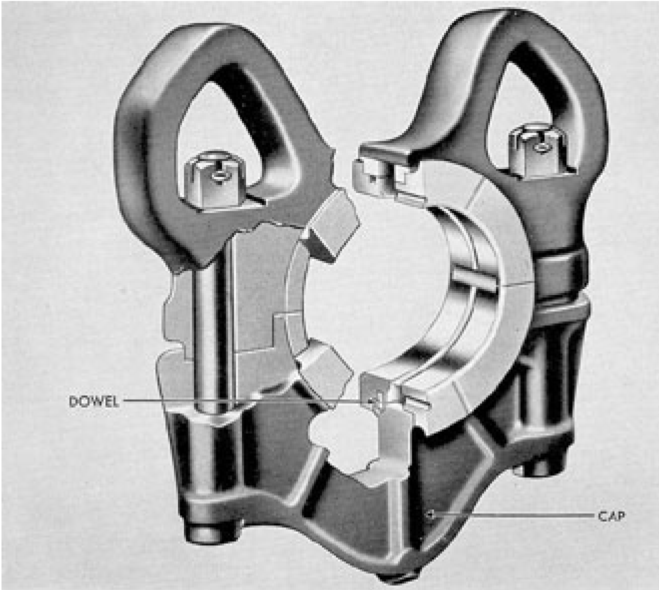


Figure 3-41. Lower crankshaft thrust bearing, F-M.

Otherwise the amplitude of the vibration may become great enough to cause breakage of the crankshaft.

The torsional damper is mounted on the lower crankshaft at the control end of the F-M 10-cylinder engine. This unit consists of a spider fitted with eight damper weights. These are installed in two rows in slots in the spider. Each weight is located and free to move in or out on the two weight pins, according to the speed of rotation of the crankshaft. Lubrication is furnished to the moving parts of the damper from the engine pressure system by means of grooves and holes in the spider hub.

similar, but are not interchangeable because of the position of the injector nozzle grooves in the piston crown. The pistons are made of closely grained cast iron and are tin plated. Each piston has four compression piston rings near the crown end. One oil control ring and two oil drain rings are located near the piston skirt. The oil control ring controls lubrication of the cylinder wall, and the oil drain rings prevent excessive lubrication of the cylinder wall. The amount of oil on the cylinder walls is also controlled by a row of small, drilled holes at the skirt end of the piston. These holes allow the lubricating oil to escape and drain to the crankcase through the piston wall after the piston rings have scraped it off the liner. They

In addition to the torsional damper, it is necessary to devise a method of preventing torsional vibrations between the crankshafts and the various auxiliary drives. This is usually done by means of flexible drive gears in each auxiliary gear train and by a flexible spring coupling in the vertical drive shaft.

d. Pistons and piston rings. The upper and lower pistons of the F-M diesel engines are also prevent excessive pressure being built up behind the oil rings, thereby cutting down the amount of ring wear. The piston pin fits into a cast steel piston pin bracket which is in turn bolted to the main piston. The pistons are cooled by oil under pressure from the engine lubricating system. The oil is forced into the oil cooling chambers under

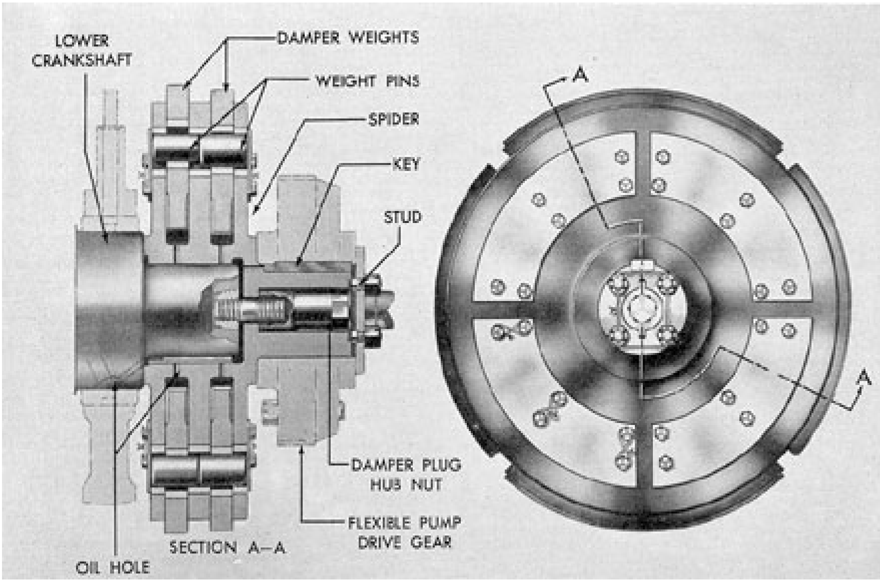


Figure 3-42. Torsional damper, F-M.

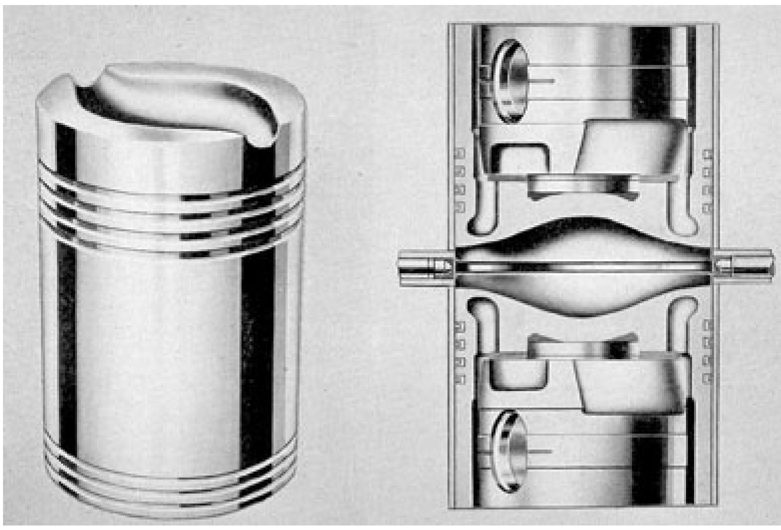


Figure 3-43. Pistons, F-M.

the piston crown. This oil drains out of the piston by means of a cooling oil outlet pipe which is The piston pin fits into the bores in the piston pin bracket which is bolted to the inside of the main

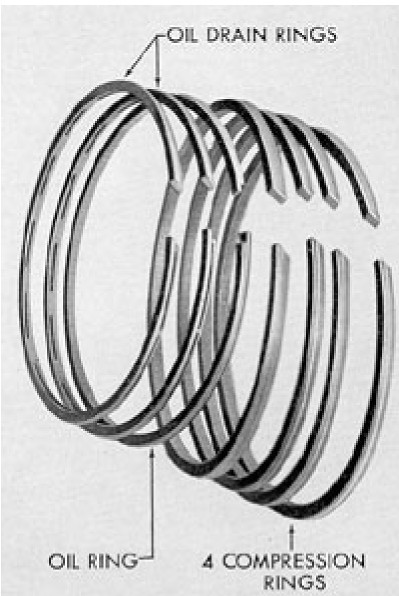
set at an angle and jets the outlet stream of oil so that it follows the throw of the crank, thereby keeping the hot oil from splashing on the connecting rod bearing and cylinder liner and preventing excessive churning and frothing of the oil. The compression rings are gold seal. These are made of cast iron and have a small bronze insert in a slot around the face of the ring. The bronze insert protrudes slightly beyond the surface of the face part of the ring. They are used because the bronze, being a softer metal, makes the ring conform more rapidly to the already worn-in cylinder wall than would an all cast iron ring. This shortens the wearing in time of a new ring.

e. Piston pin assembly. Connection between the piston and connecting rod is made by the piston pin and the piston pin bracket. The latter is bolted inside the piston, clamping the piston pin tightly and forming an enclosure for the piston cooling oil.

piston and holds the piston pin in place. Thus there is no possibility of loose piston pins damaging the cylinder walls. The bearing receives oil through holes in the connecting rod bushing which are aligned with the oil groove in the upper connecting rod eye.

There are two types of piston pin bearings. The sleeve bearing, or bushing type of piston pin bearing, consists of a cast bronze lining pressed into the steel bushing in the connecting rod eye. Lubrication is supplied by oil holes in the steel bushing which line up with the drilled oil hole in the connecting rod. Grooves on the surface of the lining distribute oil over the bearing surface.

The F-M 9-cylinder diesel engines were originally equipped with a needle roller type piston pin bearing instead of the bushing type of bearing. However, replacements for this assembly are of the plain bushing type. In the roller bearing type the inner race is formed by



the case-hardened steel piston pin. There are two rows of hardened steel needle rollers with 43 rollers in each row. The rows are separated by three retainer rings. The outer race of the bearing is formed by the case-hardened steel bushing that fits into the eye of the connecting rod.

f. Connecting rods and connecting rod bearings. The construction of the upper and lower connecting rods is basically the same except

Figure 3-44. Piston rings, F-M.

that the lower connecting rod is longer than the upper connecting rod. The connecting rods are made of alloy steel forgings. The rods are forged in an I-section with a closed eye at the piston end to receive the piston pin bearing and a removable cap at the crank end which encloses the connecting rod bearing shells. The cap is secured to the rod by two bolts, castle nuts, and cotter pins.

The connecting rod bearing is made up of upper and lower bearing shells. These are bimetal precision type bearings with machined bronze or steel backs and with a centrifugally cast soft lining of Satco metal, a high lead bearing metal. Both bearing shells are kept from rotating by dowel pins in the cap and bearing

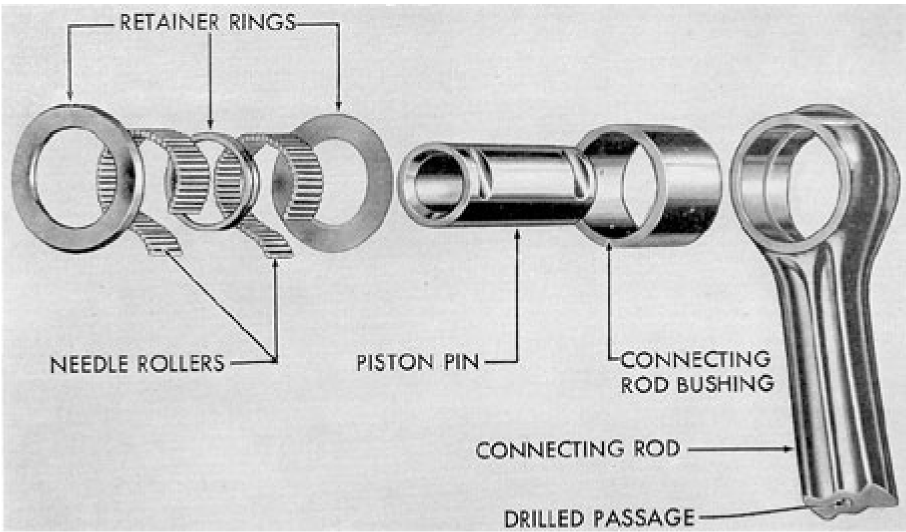


Figure 3-45. Needle roller type piston pin assembly, F-M.

seats. Oil is forced under pressure through holes in the bearing shell to the grooves on the inner surface of the bearing shell and to the oil passage in the connecting rod shaft. Bearing shells are marked on the outside of one flange with the number of the connecting rod. Shells should

opposed piston engines the upper and lower crankshafts are connected at the blower end of the engine by a flexible, vertical drive shaft (Figure 3-48). A portion of the power of the upper crankshaft is expended in driving accessories and in driving the blower. The remaining power of

be installed with the markings placed toward the control end of the engine.

g. Vertical drive. On Fairbanks-Morse the upper crankshaft is delivered through the vertical drive shaft to the lower crankshaft of the engine. Gears and bearings on

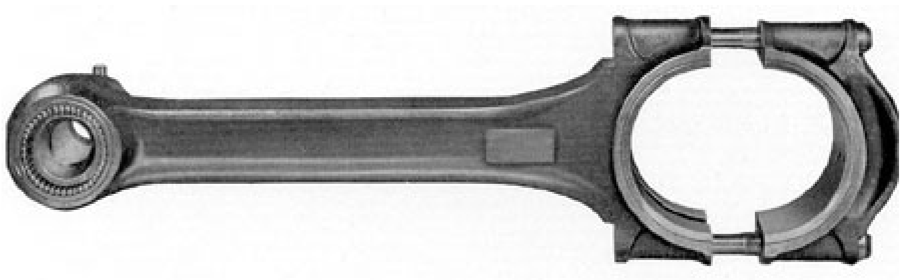


Figure 3-46. Connecting rod with needle roller type piston pin bearing, F-M.

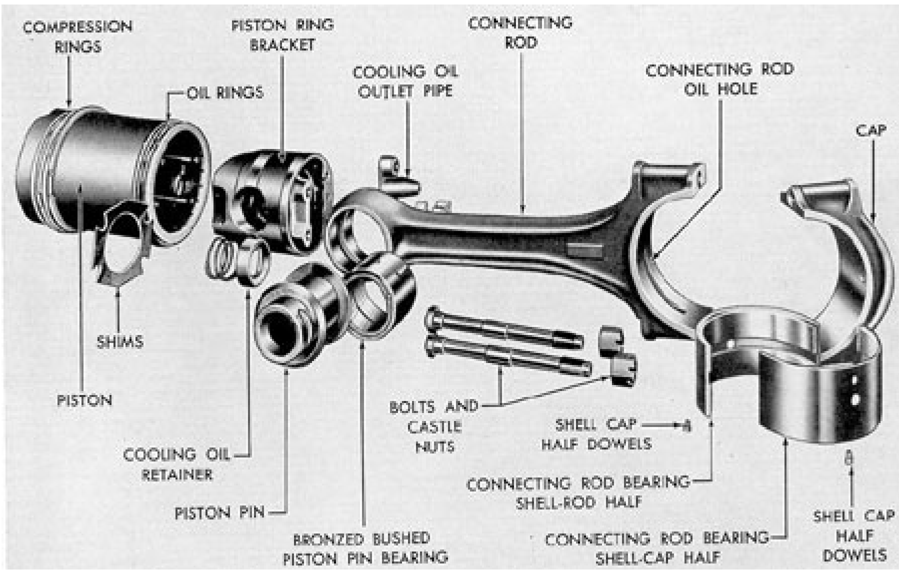


Figure 3-47. Connecting rod and piston assembly, F-M.

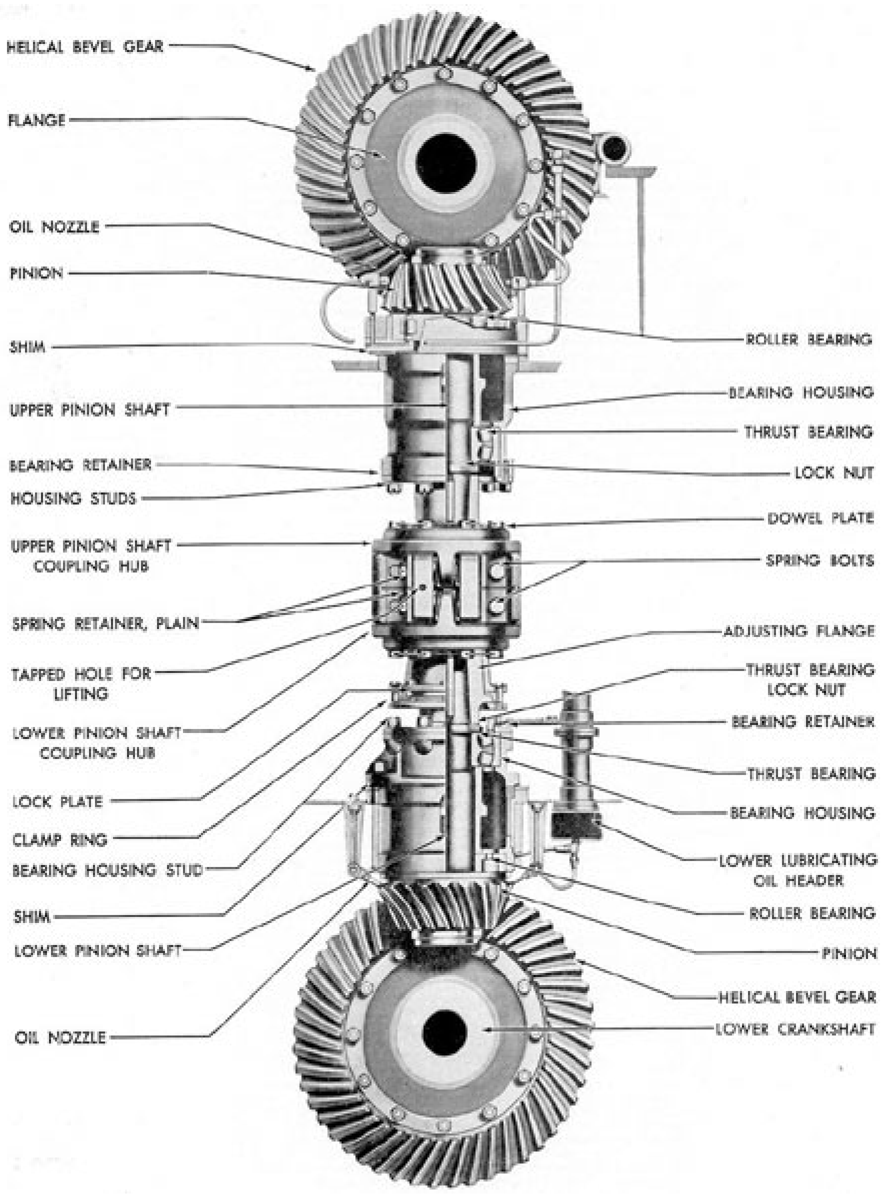


Figure 3-48. Assembled view of crankshaft vertical drive on 10-cylinder F-M engine.

the vertical drive shaft are lubricated directly from the main engine lubricating systems through tubes leading from the riser duct connecting the upper oil header to the lower oil header.

The 9- and 10-cylinder engine vertical drives differ in construction. In the 10-cylinder engines, helical bevel gears or ring gears are bolted to flanges on both the upper and the lower crankshafts. Each of the ring gears meshes with a helical bevel shafts by means of intermediate lock plates and friction between the two cone, or tapered, surfaces. It permits an unlimited adjustment of timing to achieve an exact 12-degree crank lead of the lower crankshaft. Timing of the crankshafts for the 12-degree lead of the lower crankshaft is achieved by setting the crankshafts before clamping, then locking the clamp ring and installing the lock plates.

The 9-cylinder Fairbanks-Morse vertical drive and the 10-cylinder vertical drive use a flexible

pinion. Each of the pinions is fitted and keyed to a vertical pinion drive shaft. The two pinion shafts rotate in roller and thrust bearings located in the upper and lower drive housings. These housings are bolted to the horizontal decks of the cylinder block.

The inner ends of the pinion drive shafts are keyed to coupling hubs. The upper and lower pinion shaft coupling hubs are connected together by means of a flexible coil spring coupling unit having an upper and lower coupling hub and an adjusting flange, or cone coupling as it is sometimes called. The upper pinion shaft coupling hub bolts directly to the flexible spring coupling upper hub. The lower pinion shaft coupling hub is connected to the lower flexible spring coupling hub through the adjusting flange. Thus the upper and lower pinion shafts are connected by a spring-loaded flexible coupling which consists of upper and lower members between which are housed 16 coil springs held by retainers. Torque on the upper hub of the flexible coupling is passed to the coil springs which in turn apply torque to the lower hub of the flexible coupling. Thus the coupling has torsional flexibility which permits it to absorb crankshaft torsional vibrations.

The flexible spring coupling also has a certain amount of vertical flexibility to allow for expansion due to operating temperatures. It has sufficient flexibility to account for a small amount of coupling, which consists of a coupling shaft with upper and lower collar, and which has a set of laminated saw steel rings at each end. Each laminated group consists of 30 to 40 rings, each .019 inch thick. The set, when installed and compressed, is about 5/8 inch thick. The upper pinion shaft coupling hub and the upper collar of the coupling shaft are bolted to the laminated rings at different points. The lower collar of the coupling shaft and the upper collar of the adjusting flange are bolted to the lower set of laminated rings at different points. The vertical flexibility of the coupling through the laminated rings allows variations due to expansion of the engine. In addition some of the Fairbanks-Morse 10-cylinder engines use a coil spring type of flexible coupling.

h. Flexible drive far auxiliaries. The fresh water and sea water circulating pumps, the attached fuel oil and lube oil pumps, and the Woodward governor are all driven from the lower crankshaft through a flexible gear drive at the control end.

In this drive, power is transmitted through springs which absorb shocks inherent in the engine and transmitted by the lower crankshaft. The two circulating water pumps are driven directly from the flexible gear through their driven gears. The fuel oil pump drive gear and idler gear in turn rotate the fuel oil pump driven gear. The lubricating oil pump driven, gear and drive shaft drive the pump by means of a coupling meshing with slots in the end of the shaft.

misalignment between the upper and lower pinion shafts.

The adjusting flange serves as a means of disconnecting the vertical drive so that the crankshafts may be turned separately for servicing. It is clamped to a tight fit over the tapered lower pinion shaft coupling hub by means of a clamp ring and retains a fixed relation between

A bevel gear, mounted on the lubricating oil pump drive shaft, meshing with a mating gear on the governor drive shaft, drives the governor through the governor coupling shaft The ball bearings and gears of the governor drive are lubricated with oil thrown off from the

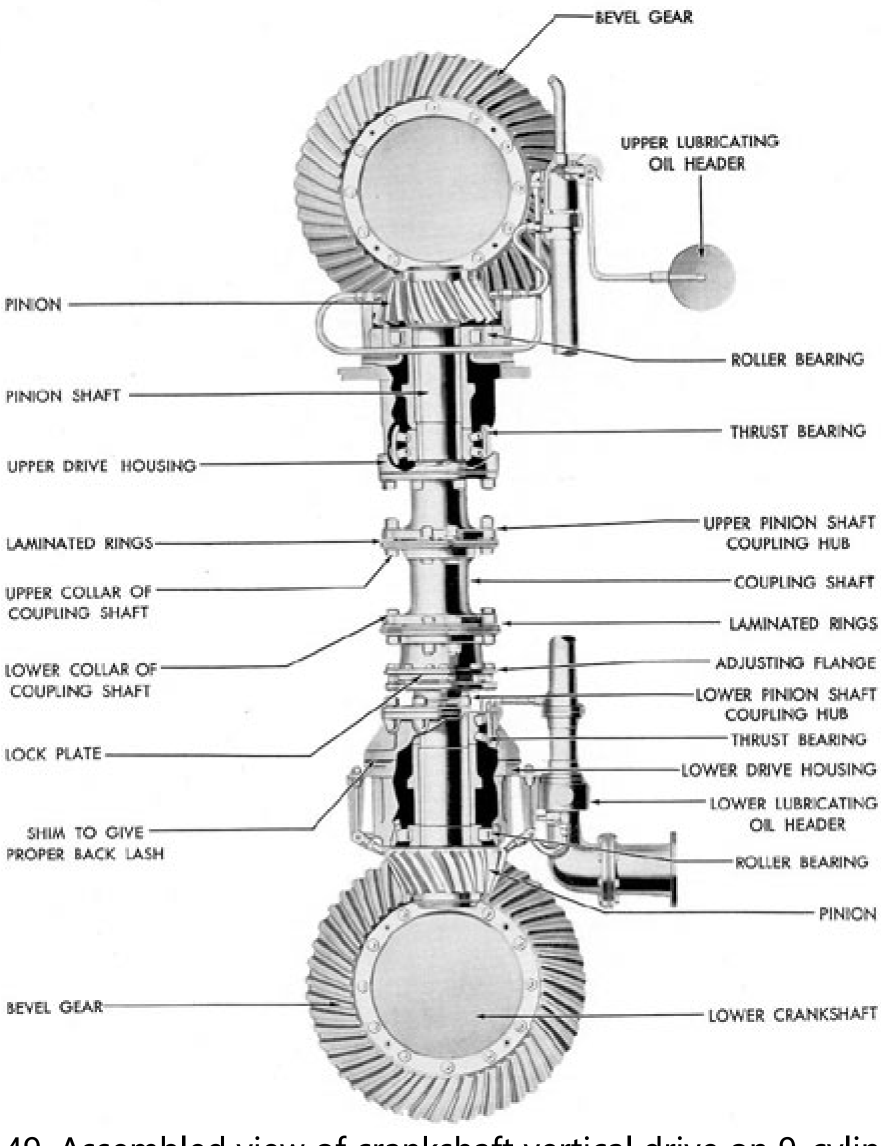


Figure 3-49. Assembled view of crankshaft vertical drive on 9-cylinder F-M engine.

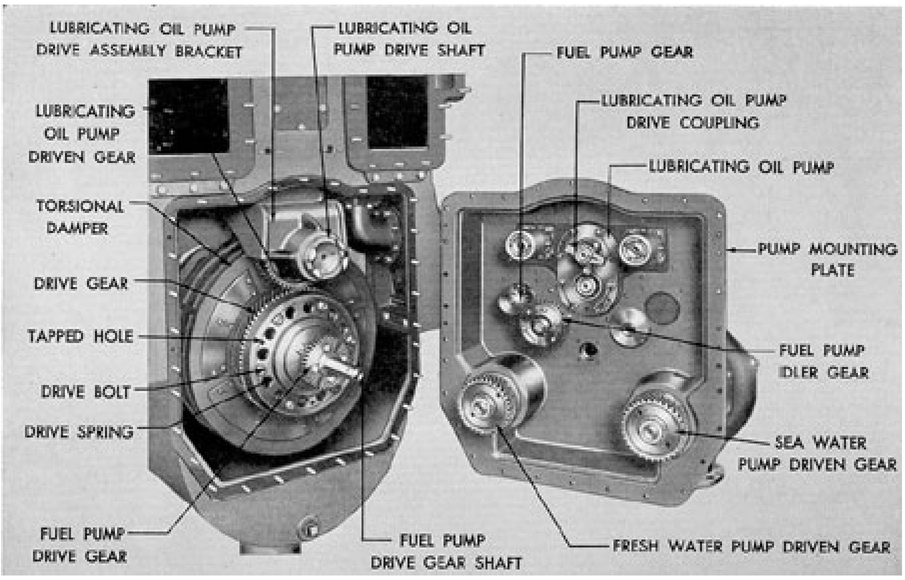


Figure 3-50. Flexible drive with housing cover removed, F-M.

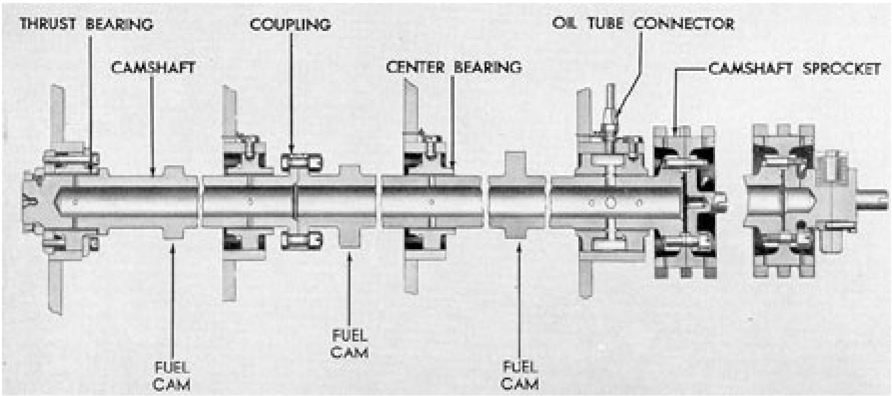


Figure 3-51. Camshaft cross section showing control end of both camshafts, F-M.

timing chain in the control end compartment. The flexible drive is lubricated by oil forced through the crankshaft to the flexible drive gear, oil dropping from the Woodward governor drive

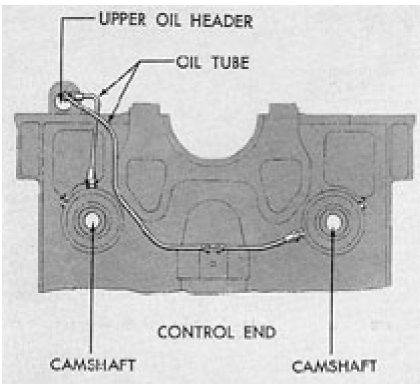


Figure 3-52. Camshaft lubrication, F-M.

shafting, and returning oil from the upper crankcase.

3D4. Valves and valve actuating gear. a. Camshaft. The Fairbanks-Morse engine has two camshafts which are located in the upper crankshaft compartment. The function of the camshafts is to actuate the two fuel injection pumps at each cylinder in exact unison and at the proper time.

Both camshafts are made of steel and each consists of three or four sections flanged and bolted together. The cams are forged integral with the camshaft and then ground to a master camshaft.

The sections are match-marked so that they may be connected in proper relationship to each other. The camshafts may be removed a section at a time or in one unit. There is one cam for each cylinder on each camshaft. Each camshaft is operated by a sprocket bolted to a flange on the control end of the camshaft. Both camshaft sprockets are driven by one timing chain so that proper timing between the two camshafts is maintained.

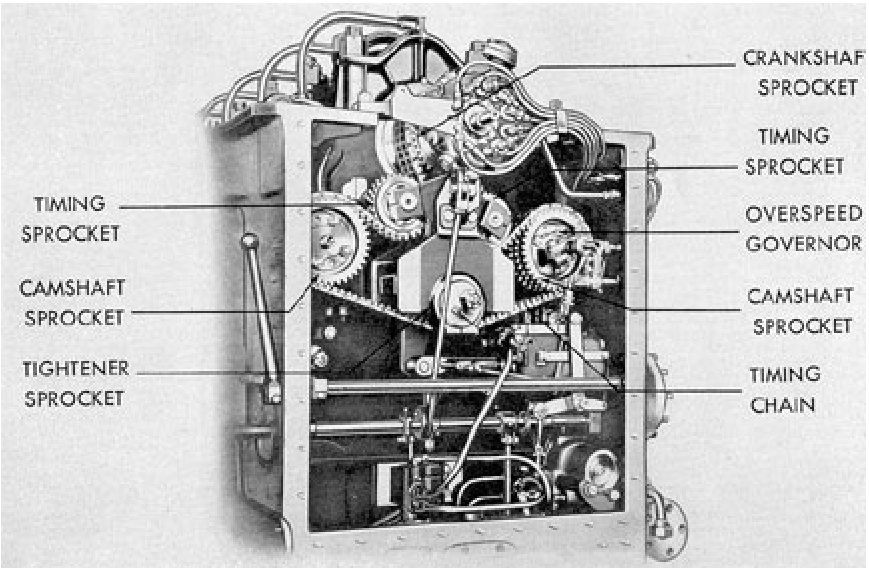


Figure 3-53. Timing chain, F-M.

Each camshaft operates in bearings located at each vertical member or crosswebbing of the cylinder block. Camshaft thrust is taken by a thrust bearing located at the blower end of the camshaft. The camshaft bearings are of the split sleeve type, the upper and lower shells consisting of steel backs with soft metal or babbitt linings. The bearings are located and held in place by setscrews. The two halves are held together by snap rings.

The camshaft bearings are lubricated by oil from the upper

b. Camshaft drive. Both camshafts are driven by one endless type timing chain connecting the crankshaft sprocket with the camshaft drive sprockets. The crankshaft sprocket is attached to the upper crankshaft at the control end of the engine. The camshaft sprocket is attached to the end of each camshaft.

The endless timing chain passes over the crankshaft sprocket, under two opposed timing sprockets, over the two camshaft drive sprockets, and under a chain tightener sprocket. The two timing

lubricating oil header. The oil is led through oil tubes to the control end bearing of each camshaft. Oil enters the bearing cap, is forced through a hole in the bearing shell and camshaft journal to the hollow bore of the camshaft. It is then forced through radial drilled holes to each of the bearings along the entire camshaft. Oil holes in the hubs at the driving ends of the camshafts connected with the holes in the camshaft sprockets provide oil for the timing chain. The overspeed governor is mounted at the control end of the left-hand camshaft in the left-hand rotation engines and on the right-hand camshaft in right-hand rotation engines.

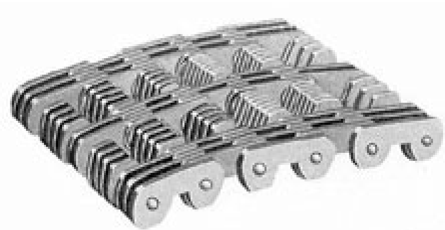


Figure 3-54. Timing chain details, F-M.

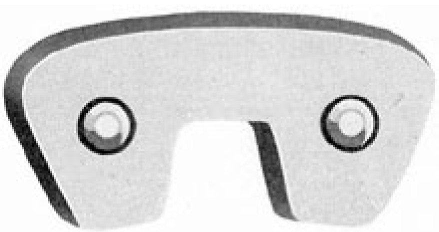


Figure 3-55. Timing chain link, F-M.

sprockets are mounted on a timing bracket. The timing bracket has an 8-degree pitch adjustment. Moving the timing bracket arm moves both timing sprockets on the chain at once, changing the camshaft relation to the crankshaft. Thus the chain is tightened in operation between the crankshaft sprocket and one camshaft drive sprocket, and given more slack between the crankshaft sprocket and the other camshaft drive sprocket. This provides a precision adjustment for securing the exact phase relation desired between the crankshaft and two camshafts. The tightener sprocket is adjustable to secure the proper slack in the chain.

The timing chain is assembled as shown in Figure 3-53. The guide links for guiding the chain on the timing sprockets and tightener sprocket ride in slots in the crankshaft and camshaft drive sprockets.

c. Cylinder relief and indicator valves. Each cylinder in the engine is fitted with a water-cooled, automatic relief valve for the out let of excessive pressures in the combustion space. The relief valves are normally set to open at about 2,000 psi, and to close as soon as the pressure has dropped below this setting.

The relief valve is threaded into the adapter valve which also has a tapped opening for an indicator valve. The complete assembly is attached to a cylinder liner valve adapter sleeve by means of a threaded collar and stud nuts.

The relief and indicator valve adapter sleeve is cooled by water

which is admitted to the adapter water jacket through a groove machined on the outside of the cylinder liner.

The indicator valves are threaded and screwed permanently into the cylinder relief valve adapters. When a pressure reading is to be

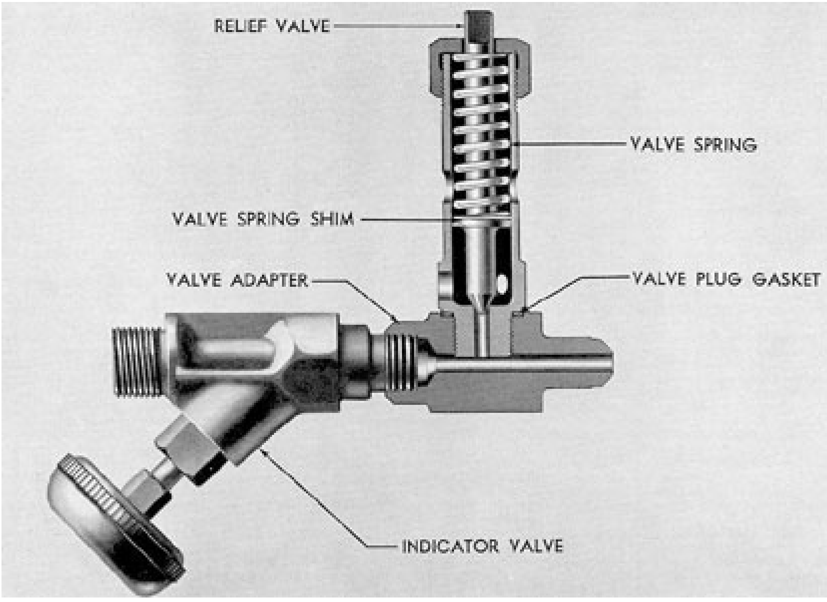


Figure 3-56. Cylinder relief valve, F-M.

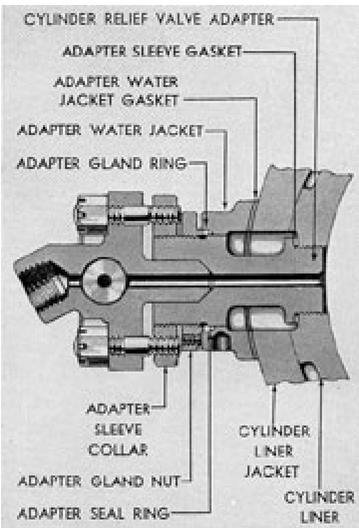


Figure 3-57. Cylinder relief valve and adapter, F-M.

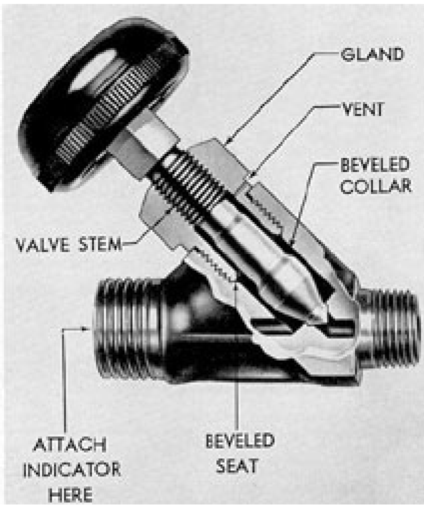


Figure 3-58. Indicator valve, F-M.

taken, the pressure indicator is screwed on the end of the indicator valve. Opening of the

leakage and burning of the valve stem which is not packed.

valve backs a beveled collar seat on the valve stem against a beveled seat on the valve gland and closes small vent holes in the valve body. Pressure is then free to pass to the indicator without

As the indicator valve is closed, the beveled collar is forced against the other seat in the inlet port, and the pressure in the indicator is free to escape through the vent holes in the valve body.

GENERAL MOTORS

ENGINE DATA, RATINGS, AND CLEARANCES

	MAIN ENGINES		AUXILIARY ENGINE
Model number	16-278A	16-248	8-268
Number of cylinders	16	16	8
Bore and stroke	8 3/4 x 10 1/2	8 1/2 x 10 1/2	6 3/8 x 7
Brake horsepower (continuous)	1600	1600	437
Generator output	1134 kw	1134 kw	300 kw
Rated engine speed	750 rpm	756 rpm	1200 rpm
BMEP (continuous)	83.6 psi	88.6 psi	80.7 psi
Type of engine	2 cycle	2 cycle	2 cycle
Cylinder arrangement	40 degrees "V"	40 degrees "V"	In line
Cylinder firing order for left-hand rotation	1-9-8-16-2-10-6-14-4-12-5-13-3-11-7-15	1-9-8-16-2-10-6-14-4-12-5-13-3-11-7-15	1-6-5-2-8-3-4-7
Cylinder firing order or right-hand rotation	1-15-7-11-3-13-5-12-4-14-6-10-2-16-8-9	1-15-7-11-3-13-5-12-4-14-6-10-2-16-8-9	
Starting system	Air	Air	Air
Injection system	Mechanical	Mechanical	Mechanical
Scavenging	Uniflow	Uniflow	Uniflow

Operating Pressures of Full Load and Speed

Lub. oil-cooler to engine	40-50 psi	50 psi	35-45 psi
Fuel oil to injectors	40-50 psi	35 psi	50-60 psi
Sea water-pump to coolers	15-35 psi	34 psi	10-40 psi
Fresh water-pump to engine	18-35 psi	31 psi	15-28 psi
Starting air pressure to engine	500 psi	500 psi	500 psi

Operating Temperatures at Full Load and Speed

Max. exhaust at 3" Hg. back pressure	550-650 degrees F	670 degrees F	650 degrees-750 degrees F
Lubricating oil-engine to cooler			
Range	140-180 degrees F	138 degrees-147 degrees F	140 degrees-180 degrees F
Preferred	170 degrees	F 145 degrees F	170 degrees F
Lubricating oil-from cooler	130-160 degrees F	114 degrees-119 degrees F	120 degrees-155 degrees F
Fresh water-engine to cooler			
Range	140 degrees-170 degrees F	150 degrees-154 degrees F	140 degrees-180 degrees F
Preferred	160 degrees F	160 degrees F	160 degrees F
Fresh water-from cooler	105 degrees-130 degrees F	130 degrees-134 degrees F	115 degrees-135 degrees F
Sea water temperature rise through fresh water cooler	10-20 degrees F	17-19 degrees F	10-25 degrees F

Pressure Relief Valve Settings

Lubricating oil regulating			
At full load and speed	40-50 psi		
At 375 rpm - no load	8-12 psi		
Lubricating oil at engine	50 psi	50 psi	35-45 psi
Fuel oil to injectors	40-50 psi	35 psi	50-60 psi

Miscellaneous

Overspeed trip setting	802 rpm	805 rpm	1350 rpm
Torque limits for bolts, studs-ft lb			
Cylinder head studs	650		325
Connecting rod bolts	100		70
Main bearing bolts	550		300
Cylinder liner studs	175		

Clearances of General Motors Principal Parts (in Inches)

Main Bearing	Model 278A	Model 248	Model 268A	Model 268
Shell to Shaft Clearance	.006-.009	.006-.009	.0035-.0065	.0035-.0065

(New)				
Shell to Shaft Clearance-Max. Allowable (Worn)	.030	.030	.020	.020
Min. Allowable Shell Thickness (Worn)	.360[1]	.360	.240	.240
Thrust Bearing End Clearance-New	.030-.035 .118-.125	.030-.035	.010-.013	.010-.013
Thrust Bearing End Clearance-Max. Allowable (Worn)	[31]	.050	.025	.025
Connecting Rod				
Shell to Shaft Clearance-New	.0065-.0085	.0065-.0085	.0025-.0065	.0025-.0065
Shell to Shaft Clearance-Max. Allowable (Worn)	.025	.025	.010	.010
Min. Allowable Shell Thickness (Worn)	.238[1]	.238[1]	.180	.180
Min. Allowable Piston Pin Dia. (Worn)	2.992	2.992[66]	2.620	2.620
Min. Allowable Piston Pin Bushing-Outer Dia. (Worn)	3.864	3.864[66]	Shrink fit	Shrink fit
Max. Allowable Piston Pin Bushing-Inner Dia. (Worn)	3.012	3.012[66]	2.637	2.637
Clearance-Piston Pin to Conn. Rod Bushing New)	.0025-.0045	.0025-.0045[66]	.004-.005[7] .0022-.0037[8]	.004-.005[7] .0022-.0037[8]
Max. Allowable Piston Pin to Conn. Rod Bushing Clearance (Worn)	.015	.015[66]	.010	.010

Piston and Liner				
Piston to Liner Clearance-New	.0525-.0555[2] .0275-.0305[3] .0155-.0185[4]	.0485-.0515[2] .0235-.0265[3] .0115-.0145[4]	.0365-.0395[2] .0095-.0125[9] .0075-.0105[4]	.0365-.0395[2]}[66] .0185-.0215[9]}[66] .0065-.0095[4]}[66]
Max. Allowable Piston to Liner Clearance (Worn)	.080[2] .055[3]	.075[2] .050[3] .040[4]	.055[2] .030[9] .030[4]	.055[2] .030[9] .030[4]
Min. Allowable Piston Dia. (Worn)	8.720[4]	8.475[4]	6.480[4]	6.355[4]
Max. Allowable Liner Dia. (Worn)	8.775	8.525	6.520	6.395
Max. Allowable Liner Out of Roundness (Worn)	.007	.007	.005	.005
Max. Piston Pin Bushing Dia. (Worn)	3.008	3.008	2.632	2.632
Piston Pin to Piston Bushing Clearance-New	.0005-.0025	.0005-.0025	.0007-.0022	0007-.0022
Piston Pin to Piston Bushing Clearance-Max. Allowable (Worn)	.016	.016	.012	.012
Piston Rings[19]				
Compression-Gap Clearance (New)	.030-.050	.030-.050	.025-.040	.025-.040
Oil Control-Gap Clearance (New)	.030-.050	.030-.050	.026-.041	.026-.041
Compression-Gap Clearance-Max. Worn	.100	.100	.100	.100
Oil Control-Gap Clearance-Max. Worn	.100	.100	.100	.100
Compression-Side Clearance (New)	.008-.0105[6] .004-.0065[6]	.008-.0105[5] .004-.0065[6]	.0075-.0100[5] .0045-.0070[12]	.0030-.0055[66]
Oil Control-Side Clearance	.002-.0045	.002-.0045	.0025-.0055	.0020-.0045

(New)				
Compression-Side Clearance-Max. (Worn)	.020	.020	.012	.012
Oil Control-Side Clearance-Max. (Worn)	.020	.020	.012	.012
Camshaft				
Min. Shell Thickness (Worn)	.117	.180	.092	.092
Clearance Shell to Shaft-(New)	.0035-.0061	.0025-.0045	.0025-.0055	.0025-.0055
Max. Allowable Shell to Shaft Clearance (Worn)	.010	.010	.010	.010
Valves				
Exhaust Valve Tappet Clearance (Cold)	.015	.015[65]	.018	.018
Exhaust Valve Guide to Valve Stem Clearance-New	.002-.004	.002-.004	.003-.005	.003-.005
Max. Allowable Exhaust Valve Guide to Valve Stem Clearance (Worn)	.012	.012	.010	.010
Miscellaneous				
Injection Nozzle Opening Pressure	1000[67] 3200[68]	3200	1000[67] 3200[68]	1000[67] 3200[68]
Injector Tappet Clearance				
Injector Timing (BTC)	7 degrees[32]	7 degrees[32]	12 degrees[32]	12 degrees[32]
Lube Oil Recommended-Navy Symbol (First Choice Substitute in Paren.)	9370-(9250)	9370-(9250)	9370-(9250)	9370-(9250)

[1] For Satco shells only-replace Tri-metal bearings when intermediate bronze lining starts to show through

[2] At top of piston

[3] Between fourth and fifth ring grooves

[4] At skirt

[5] First and second rings from top

[6] Third, fourth, and fifth rings from top

[7] Bushing with six radial holes

[8] Bushing with two radial holes

[9] Bottom of taper below fourth ring groove

[31] Replace when backing begins to show through bearing metal

[32] Position of flywheel for checking injector with injector timing tool

[66] Engines with C. I. pistons

[67] Spherical check valve type

[68] Needle valve type

FAIRBANKS-MORSE
ENGINE DATA, RATINGS, AND CLEARANCES

	MAIN ENGINES		AUXILIARY ENGINE
Model number	38D 8 1/8	38D 8 1/8	38E 5 1/4
Number of cylinders	10	9	7
Bore and stroke	8 1/8 x 10	8 1/8 x 10	5 1/4 x 7 1/4
Brake horsepower			
Emergency	1600 hp @ 720 rpm	1600 hp @ 720 rpm	
Continuous	1280 hp @ 650 rpm	1280 hp @ 650 rpm	
Generator output -			
Emergency	1120 kw @ 720 rpm	1120 kw @ 720 rpm	300 kw
Continuous	900 kw @ 650 rpm	900 kw @ 650 rpm	
Rated engine speed	-	-	1200 rpm
BMEP (continuous)	84 psi	94.3 psi	64.3 psi
Type of engine	2 cycle	2 cycle	2 cycle
Cylinder arrangement	Opposed	Opposed	Opposed
Cylinder firing order for left-hand rotation	1-8-7-3-5-9-4-2-10-6	1-9-2-7-4-5-6-3-8	1-7-2-5-4-3-6
Cylinder firing order for right-hand rotation	1-6-10-2-4-9-5-3-7-8	1-8-3-6-5-4-7-2-9	
Starting system	Air	Air	Air
Injection system	Mechanical	Mechanical	Mechanical
Scavenging	Uniflow	Uniflow	Uniflow

Operating Pressures at Full Load and Speed

Lub. oil-cooler to engine	50 psi	50 psi	40 psi
Fuel oil to injectors	25 psi	25 psi	25 psi
Starting air pressure to engine	250 psi	250 psi	250 psi

Operating Temperatures at Full Load and Speed

Maximum exhaust, individual cylinder	770 degrees F	770 degrees F	590 degrees F
Lubricating oil-engine to cooler			
Range	140 degrees-180 degrees F	125 degrees-190 degrees F	140 degrees-180 degrees F
Preferred	165 degrees F	150 degrees-170 degrees	F 165 degrees F
Fresh water-engine to cooler			
Range	140 degrees-170 degrees F	120 degrees-190 degrees F	140 degrees-170 degrees F
Preferred	160 degrees F	150 degrees-170 degrees F	160 degrees F

Oil Pressures

Lubricating oil-engine upper header	17-32 psi	17-32 psi	21-47 psi -
Fuel oil to fuel injection pump	5-20 psi	5-20 psi	7-21 psi

Miscellaneous

Overspeed trip setting	778-835 rpm	800 rpm	1290-1370 rpm

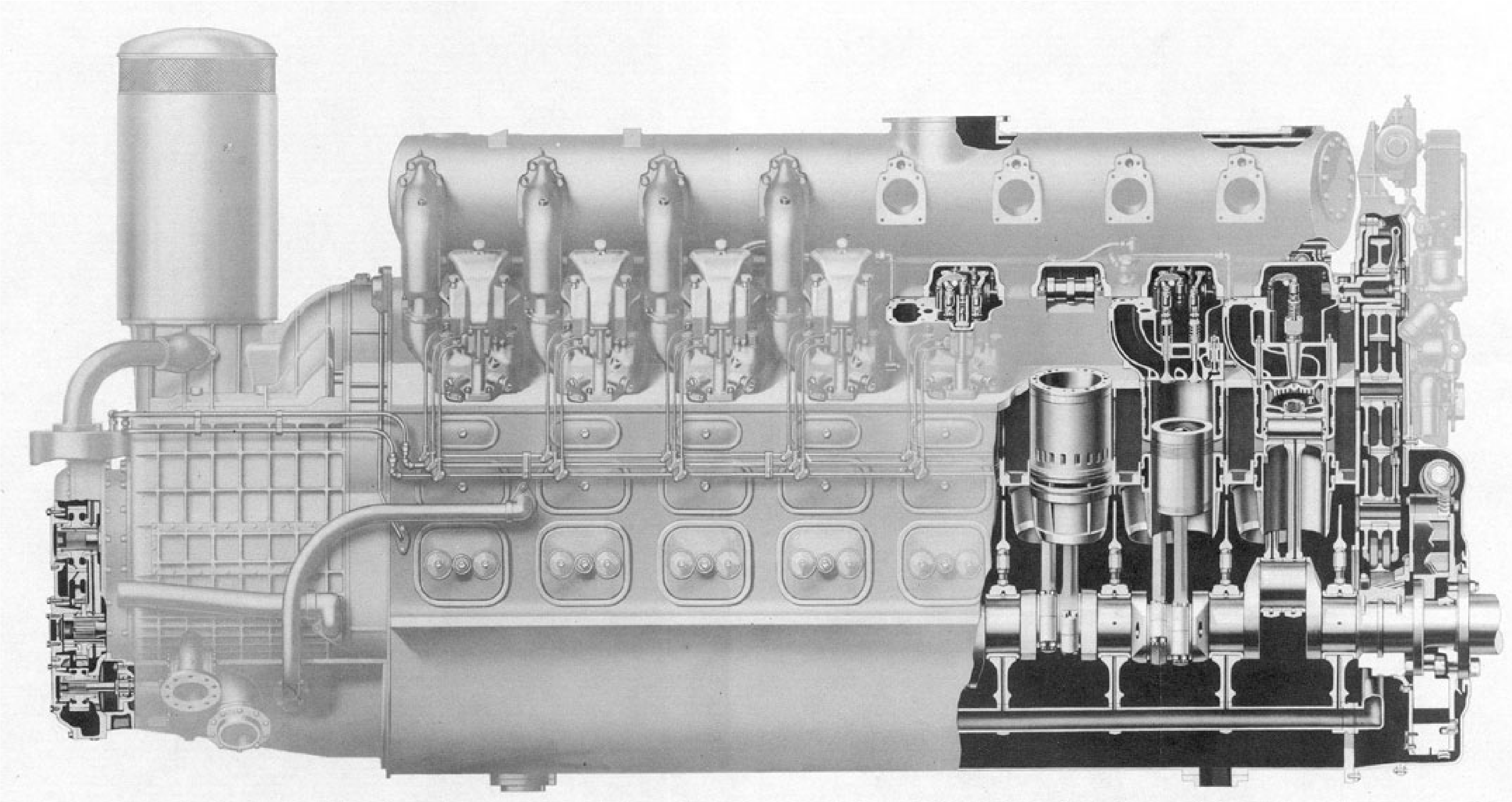
Clearances of Principal Parts (in Inches)

Camshaft bearings	0.0035-0.0060	0.0035-0.0060	0.0015-0.0040
Connecting rod bearings	0.0050-0.0075	0.0050-0.0075	0.0030-0.0055
Main bearings	0.0090-0.0115	0.0090-0.0115	0.0070-0.0095
Main thrust bearings-width	0.0050-0.0100	0.0050-0.0100	0.0050-0.0110
Piston diameter	0.0085-0.0135	0.0095-0.0125	0.0080-0.0100
Piston rings:			
Compression rings (top 2)			
side clearance	0.0080-0.0100	0.0080-0.0100	0.0090-0.0110
gap clearance	0.0300-0.0400	0.0300-0.0400	0.0200-0.0300
Compression rings (3 and 4)			
side clearance	0.0050-0.0080	0.0050-0.0075	0.0070-0.0090
gap clearance	0.0300-0.0400	0.0300-0.0400	0.0200-0.0300
Oil rings			

oil cutter ring, side			0.0005-0.0020
oil cutter ring, gap			0.0250-0.0300
other oil rings, side	0.0015-0.0045	0.0015-0.0040	0.0030-0.0036
other oil rings, gap	0.0150-0.0250	0.0150-0.0250	0.0150-0.0250
Piston pin bearing	0.0055-0.0065	0.0020-0.0034	0.0015-0.0029
Vertical drive gears Backlash	0.0120-0.0160	0.0120-0.0160	0.0030-0.0100

Figure 3-6. LONGITUDINAL CUTAWAY OF GM 16-278A ENGINE.

Sub Diesel Home Page



Version 1.10, 22 Oct 04

4
ENGINE AIR STARTING SYSTEMS

A. GENERAL

4A1. Description. Modern submarine diesel engines are started by admitting compressed air into the engine cylinders at a pressure capable of turning over the engine. This process is continued until the pistons have built up sufficient compression heat to cause combustion. The pressure used in air starting systems is approximately from 250 to 500 psi.

4A2. Source of starting air. Starting air comes directly from the ship's high-pressure air service line in which pressures up to 3,000 psi are normally maintained, or from starting air flasks which are included in some systems for the purpose of storing starting air. In either instance, the air on the way to the engine, must pass through a pressure reducing valve which reduces the higher pressure to the operating pressure required to start a particular engine. A relief valve is installed in the line between the reducing valve and the engine. This relief valve is normally set to open at 25 to 50 pounds in excess of the air starting pressure. Thus, if the air pressure leaving the reducing valve is too high, the relief valve will protect the engine by releasing air in excess of the value for which it is set and permit only air at approximately the proper pressure to reach the engine cylinders.

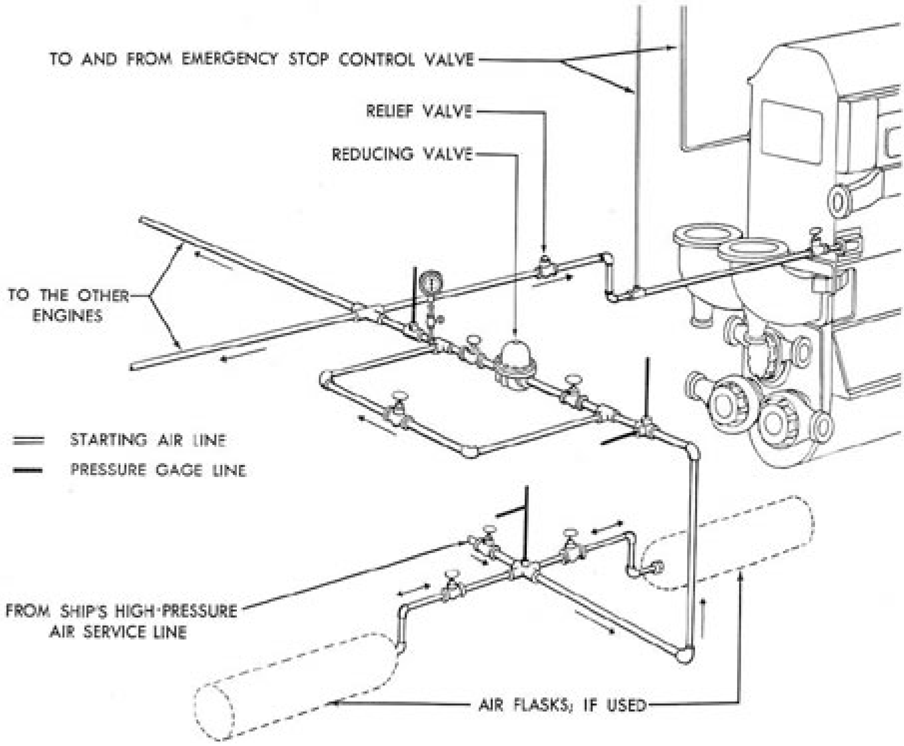


Figure 4-1. Typical starting air piping system.

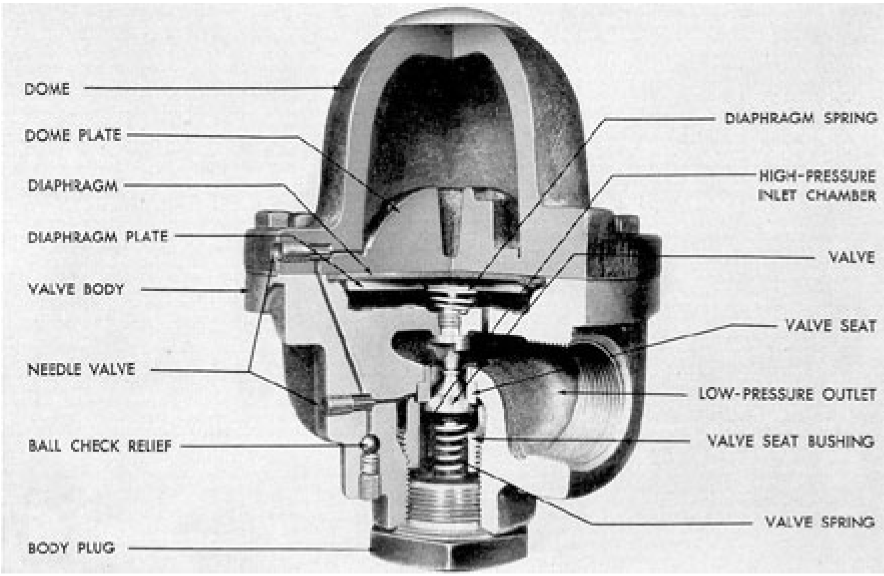


Figure 4-2. Grove regulator valve.

4A3. Pressure regulating valve. The pressure reducing valve is a Grove regulator (Figure 4-2) in which compressed air, sealed in a dome, furnishes the regulating pressure that actuates the valve. Thus the compressed air in the dome performs the same function as a spring used in a conventional type of valve.

The dome is tightly secured to the valve body which is position and the valve is forced shut by the high-pressure air acting on the valve head. When air is being used from the low-pressure side of the regulator, this action is continuous and very rapid in order to maintain the correct pressure on the discharge side.

High-pressure air entering the valve body is filtered through a screen to prevent the entrance of any particles of dirt which would

separated into an upper (low pressure outlet) and a lower (high-pressure inlet) chamber by the main valve. At the top of the valve stem is another chamber which contains a rubber diaphragm and a metal diaphragm plate. This chamber has an opening leading to the low-pressure outlet chamber. When the outlet pressure drops below the pressure in the dome, air in the dome forces the diaphragm and the diaphragm plate down on the valve stein. This opens the valve and permits high-pressure air to pass the valve seat into the low-pressure outlet and into the space under the diaphragm. As soon as the pressure under the diaphragm is equal to that in the dome, the diaphragm returns to its normal prevent the valve from seating properly. The screen is held in position around the space under the valve head by the threaded valve seat bushing. The screen should be removed and cleaned periodically to insure an unrestricted flow of air, If particles of dirt are permitted to remain and accumulate in the screen, the high air pressure may tear the screen from its position and force it into the working parts, causing damage to the valve seat.

Air for the original charging of the dome is obtained from the high-pressure chamber of the valve body by opening two needle valves, As soon as the desired pressure, as indicated by the gage on the discharge side of the regulator,

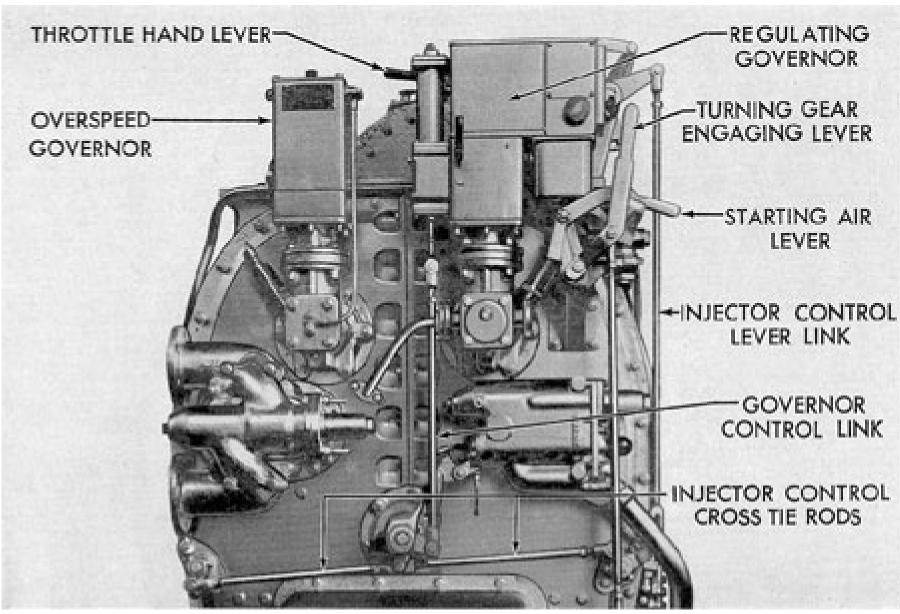


Figure 4-3. Engine starting control levers, GM.

is reached, the needle valves must be closed. The dome will then regulate and maintain the discharge of air at that pressure.

4A4. Starting the GM engine. The GM engine is started by The cylinder test valves are then closed. The engine is started by holding the throttle hand lever in the START position and opening the air starter hand valve. The engine should start after a few revolutions if the fuel supply has

means of two control levers, the throttle hand lever and the air starter hand valve lever. The throttle hand lever has three positions, STOP, START, and RUN. In the STOP, or central, position, the fuel supply to the cylinders is cut off. Moving the lever toward the START position rotates the fuel pump plunger toward the full pump position. The RUN position gives the Woodward regulating governor unrestricted control of the engine. The air starter hand valve lever has only two positions, OPEN and CLOSED.

Prior to starting the engine, and with the throttle hand lever on the STOP position, the engine is turned over several times by opening the air starter hand valve with the cylinder test valves open. This insures that there are no obstructions to prevent the starting of the engine.

been primed and is not airbound. As soon as the engine is firing, the air starter hand valve is closed and the speed of the engine adjusted by means of the throttle hand lever. As soon as the governor oil pump has built up a working pressure, the throttle lever is shifted to the RUN position. This shifts the engine to governor control.

4A5. Starting the F-M engine. The F-M engine is started by means of a control shaft lever. This lever has three positions, START, STOP, and RUN. In the STOP position, the fuel cutout cam on the control shaft moves the fuel injection pump control rod to the no fuel position. When the lever is in the START position, the air start control valve is opened, allowing air starting of the engine. In the RUN position, the engine is under full governor control.

To start the engine, the governor is set at idling speed and the control shaft lever moved from the STOP position to the RUN position and then toward the START position. When the lever passes the RUN position, the fuel injection pump control rod is unlocked. When the lever reaches the START position, air starting air begins to enter the cylinders. As soon as the engine is firing, the control shaft lever should be shifted to RUN. This allows full governor control and closes the air start control valve.

B. GENERAL MOTORS ENGINE AIR STARTING SYSTEM

4B1. Description. The engine air starting system used on GM engines is known as the separate distributor type, the starting air distributor valve being a separate unit for each cylinder. Each distributor valve is bracket bolted to the camshaft drive cover near the hand control lever. It is a poppet type valve, opened manually by a lever and closed by a spring. A plug in the valve body holds the spring against the valve head. The valve

individually operated by its cam on the camshaft. Eight of the 16 cylinders, six in one bank and two in the other, are air started, but all of the cylinder heads in both banks are equipped with air starter check valves so as to maintain full interchangeability. On the cylinders that are not air started, the air inlet opening is sealed with a removable plug.

4B2. Operation. Air is supplied to the air starting hand control valve from the air supply line. The air starting control valve is opened by a hand lever, thereby admitting air to the starting air manifold. The starting air manifold is a steel pipe extending the full length of the engine and is located on the top deck of the engine below the exhaust manifold. It is connected by air lines to each of the starting air distributor valves. The distributor valves are opened in engine cylinder firing order by their cams on the camshafts, admitting air into the lines that connect each distributor valve to its air starting check valve. As the distributor valve admits air into the line leading to the air starting check valve, the pressure opens the check valve, thereby admitting air into the combustion chamber;

The air pressure moves the pistons and turns the crankshaft until there is sufficient compression for combustion. Combustion pressure and exhaust gases are kept from backing into the air starting system by the check valves. As soon as the engine is firing, the hand lever is released, and spring pressure closes the air starting stem guide is a bronze bushing pressed into the body. A spring and head placed over the valve stem, where it projects from the body, return the hand lever to the valve's closed position. The hand lever and the operating lever stop are keyed to a shaft in the bracket.

A safety device prevents opening of the air starting control valve while the engine jacking gear is engaged.

4B4. Air starting distributor valve. Each

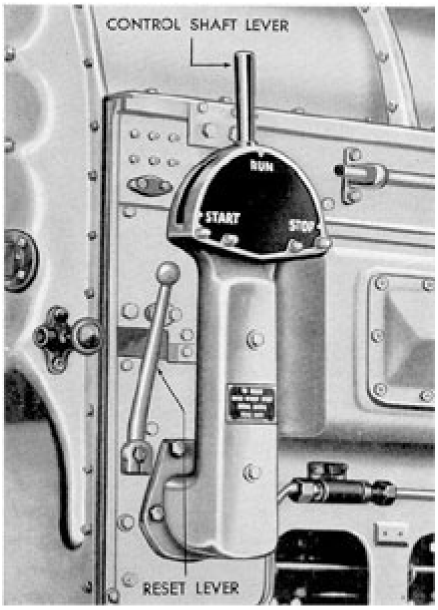


Figure 4-4. Control shaft lever, F-M.

control valve. This shuts off the supply of starting air to the engine.

4B3. Air starting hand control valve. The air starting hand control valve is mounted on a

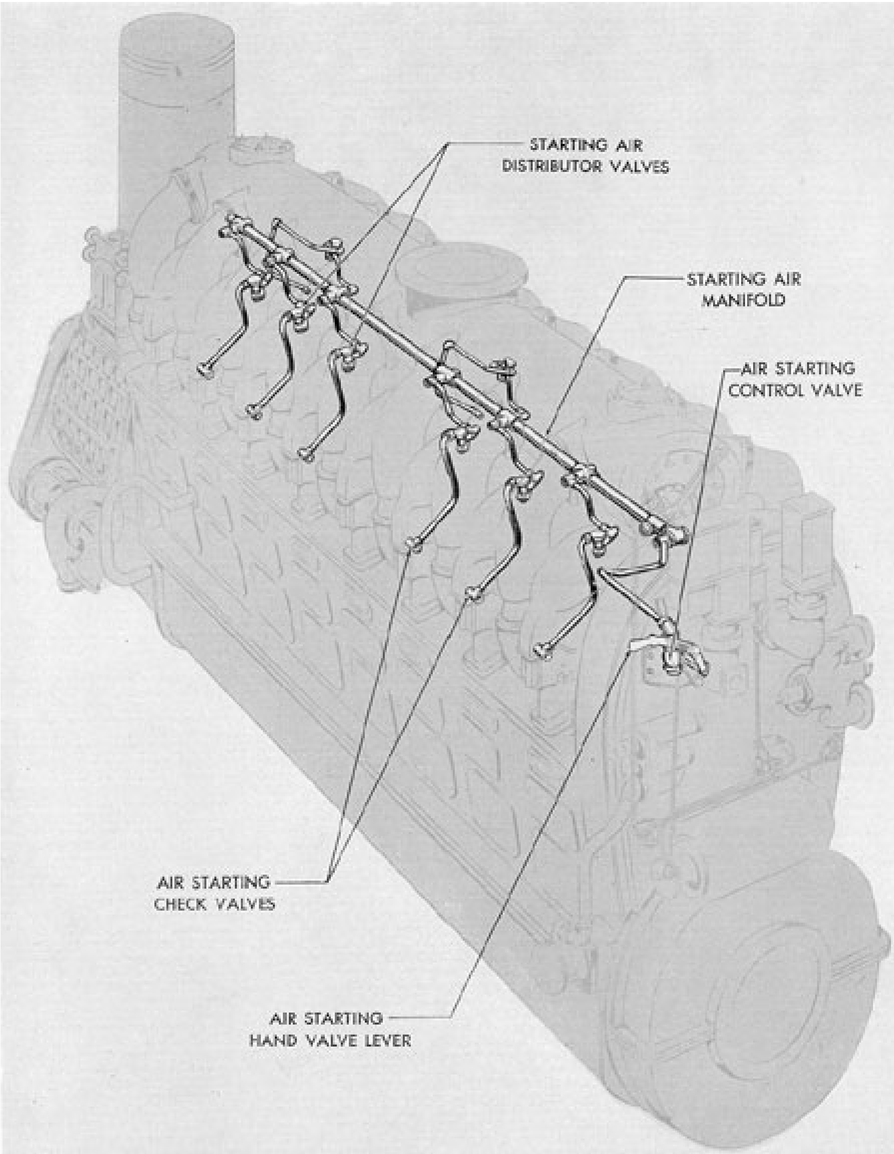


Figure 4-5. GM engine air starting system.

cylinder having air starting is equipped with an air starting distributor valve.

The air starting distributor valves, or timing valves as they are sometimes called, are of the

in the cylinder head. The valve body fits into a recess in the cylinder head and is held in place by a cap nut that screws into the cylinder head and ears on the top of the valve body. The valve body contains the valve seat and serves

poppet type with forged steel bodies that bolt to the camshaft intermediate covers. The valve is held closed by spring pressure bearing against the top of the valve and is guided in the hollow end of a cam follower which rides on the camshaft air starting cam. The cam follower is guided in a bronze bushing pressed into the valve body. A lockpin locates the cam follower in the body.

When cam action opens the valve, starting air passes from the air manifold through a chamber in the valve body above the valve head into a line leading to the air starting check valve in the cylinder head. The cam action opens the valves in the proper firing sequence. The cam follower is lubricated by oil splashed from the cam pocket by the cam.

4B5. Air starting check valve. The air starting check valve is a poppet type valve located as a valve stem guide. Air is prevented from leaking to the outside of the valve body by a synthetic rubber seal ring located above the inlet port. The valve face makes direct contact with the valve seat in the valve body. The valve is held closed by a spring over the valve stem, bearing against the valve body and also against a spring seat locked to the valve stem. The spring seat is locked in position on the valve stem with two half-round seat locks that fit into a groove in the valve stem. The valve opens into a small chamber with a short, open passage to the cylinder.

When the air starting distributor valve admits air into the line leading to the air starting check valve, the air passes into a chamber around the valve seat. The pressure of this air opens the check valve and allows the air to pass into the cylinder, moving the piston. When the

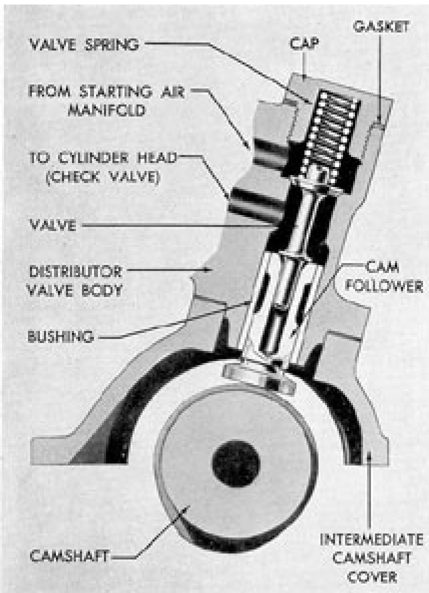


Figure 4-6. Air starting distributor valve, GM.

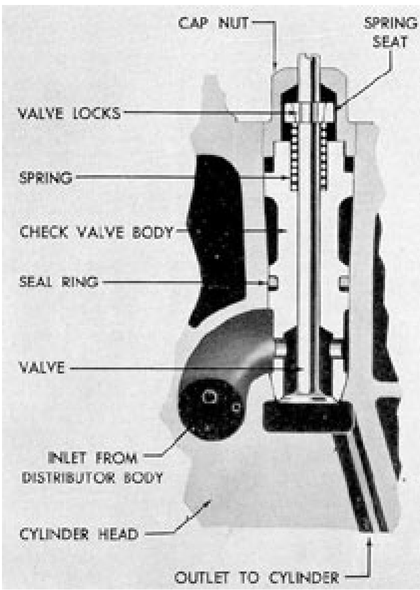


Figure 4-7. Air starting check valve, GM.

air starting distributor valve closes, the pressure drops and spring tension closes the air starting check valve.

When combustion begins, the air starting check valve remains closed, as the pressure in the combustion chamber is greater than the pressure of the starting air that actuates the check valve. This prevents exhaust gases and combustion pressures from backing up into the air starting system.

4B6. Maintenance. Line connections and valves of the air starting system should be maintained in a closely fitting, airtight, operating condition. Leakage at the air starting distributor valve is likely to result in starting failure. Leakage at the air starting check valve will start scoring of the valve seat, a condition that will become progressively worse and impair the operation of the valve.

Valve seats should be inspected at least at every major overhaul period, and the valves ground and reseated if necessary. The air starting distributor valve on the GM engines should have a clearance of between 0.010 and 0.020 inch measured between the cam and the cam follower. If the cam follower cannot be ground off sufficiently to bring the clearance within these limits, a new assembly should be installed.

C. FAIRBANKS-MORSE ENGINE AIR STARTING SYSTEM

4C1. Description. The F-M engine air starting system consists of the starting air piping and the engine starting mechanism. The engine starting mechanism includes the air start control valve, air start distributor, the starting air header, the pilot air tubing, and the air start check valves at the individual cylinders. This type of air starting system has a distributor block consisting of several pilot valves which provide actuating or pilot air to regulate the opening of the air start check valves at the proper moment, allowing the starting air itself to enter the cylinders. All cylinders of the submarine type F-M engines are air started.

4C2. Operation. The air starting control valve is manually operated from the engine forces each pilot valve plunger down into contact with the cam. Regardless of where the camshaft has stopped, one pilot air valve will be on the low point of the cam and hence will be open. Two other valves, one on each side of the open valve, will be partly open. Each of these three valves admits pilot air through a connecting tube to its individual air start check valve. This pilot air under pressure in the pilot air tubes opens the three air start check valves. Then the actual starting air rushes into the engine cylinder from the air header and forces the pistons apart, causing the crankshafts to rotate. The air distributor camshaft is attached to and rotates with the upper crankshaft; therefore the cam begins to open and close other distributor valves in proper sequence. When the engine starts

control lever. When the engine control lever is set at START, a lever linkage opens the air starting control valve, admitting air from the supply line to the air starting main header. This header is connected by branch lines to the air starting check valves at each cylinder. A branch line from the air starting control valve supplies pilot air to the distributor. This distributor includes one pilot air valve for each cylinder in the engine. These pilot valves are arranged radially and in engine firing order around the group distributor camshaft (sometimes referred to as the cam stub shaft). A spring holds each valve out of contact with the cam when the engine is running on its own power. But when air enters the distributor from the air start control valve, the air pressure overcomes the spring tension and firing, the control shaft lever is moved to the RUN position. This actuates linkage on the control shaft which closes the air start control valve, shutting off air pressure to the distributor and the air starting header. Air in the starting mechanism escapes through vents in the pilot valves and in the control valve. As the air pressure drops, the distributor valve springs raise the pilot valves off the cam.

NOTE. The pilot air that opened the check valve is vented by the distributor and does not pass into the cylinder combustion chamber.

4C3. Air starting control valve. The air starting control valve is bolted to the engine frame near the control end on the side opposite the control lever, and consists of a valve cage,

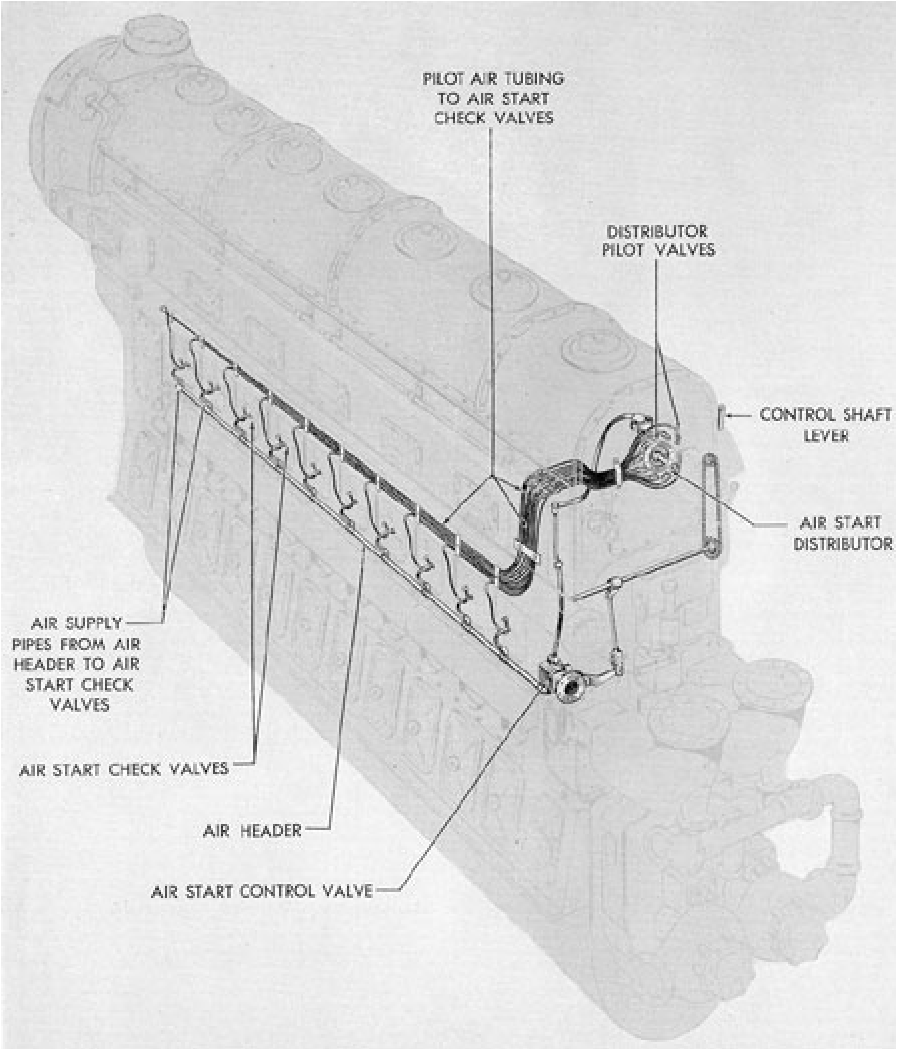


Figure 4-8. F-M engine air starting system.

valve, and valve spring. The valve is of the poppet type and has an integral stem. The valve is held on its seat by the valve spring which is placed between the valve head and the end of the valve cage. The valve stem is grooved to align with a drilled hole in the valve body, in order to vent the valve of air when the valve is closed. The end of the valve stern extends out of the valve body, and the valve is opened against valve spring pressure by a rocker arm. When the rocker arm is withdrawn from the end of the valve stem, the valve closes because of spring pressure and air pressure acting on top of the valve head.

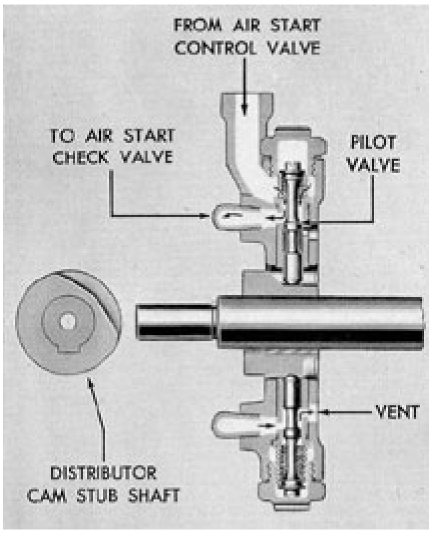


Figure 4-10. Cross section of air starting distributor, F-M.

The distributor body houses one air starting pilot valve for each engine cylinder. These valves are of the piston type with the inner end of each valve stem acting as a

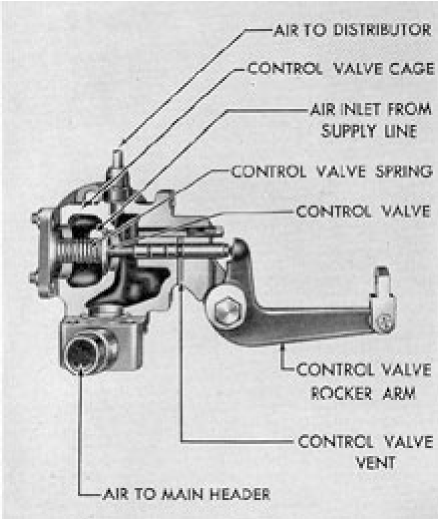


Figure 4-9. Air starting control valve, F-M.

4C4. Air starting distributor. The air starting distributor body is a large circular casting, cored to house the air starting distributor valves. The distributor body mounts on the engine frame at the control end of the upper crankshaft. The distributor camshaft passes through the center bore of the distributor body and is attached to and rotates with the upper crankshaft.

cam follower. During normal engine operation, the valves are held out of contact with the camshaft by spring pressure.

Each of the valve openings connects with an air chamber extending around the outer circumference of the distributor body. During air starting, this chamber is filled with air supplied through the branch line when the air starting control valve is opened. The air in this chamber supplies pressure to each of the air starting pilot valves. The spring tension in the valves is overcome by the air pressure, and each valve is forced into contact with the cam on the camshaft. There is a low sector on the cam, and as each valve approaches this sector of the cam, the air pressure from the outer end moves the pilot valve inward. This inward movement of the valve stem opens a passage connecting the pressure chamber in the distributor body with

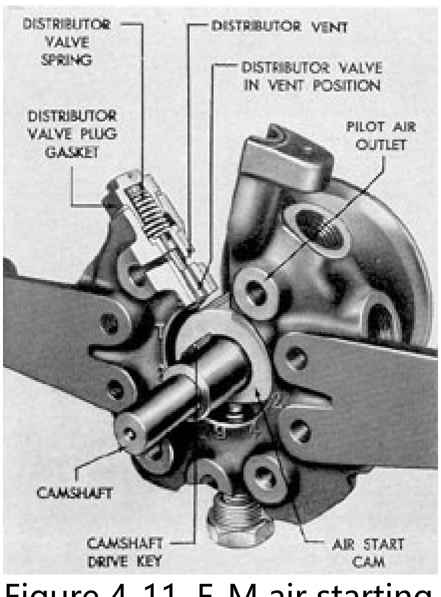


Figure 4-11. F-M air starting distributor, pilot valve in normal position out of contact with distributor cam.

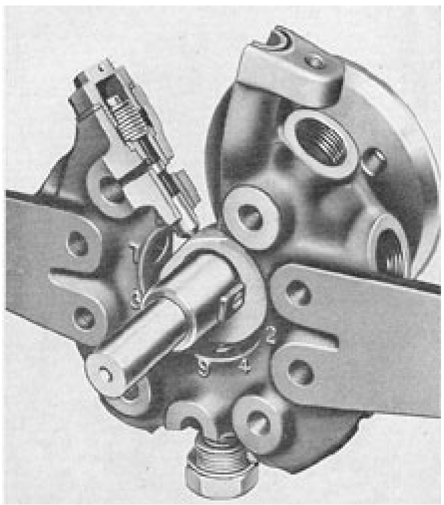


Figure 4-12. F-M air starting distributor, pilot valve on low point of cam.

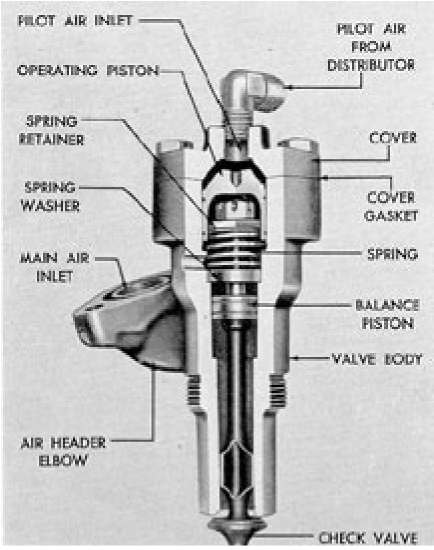


Figure 4-14. Cutaway of air starting check valve, F-M.

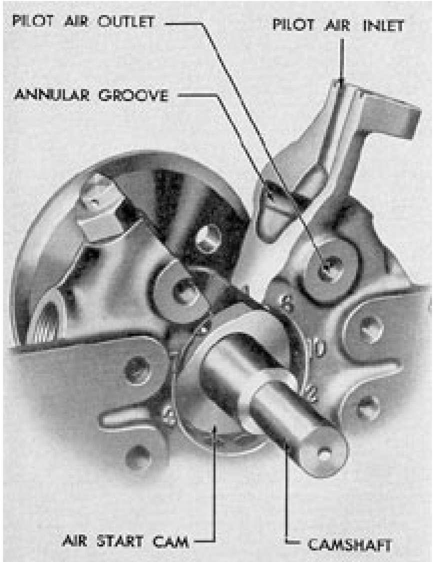


Figure 4-13. Cutaway of air starting distributor, F-M.

an individual pilot air line to the operating piston in the air starting check valve at the cylinder. This action opens the check valve.

As the high sector of the cam approaches, the valve is forced outward, shutting off the actuating air to the check valve and venting the pilot air line. Numbers marked on the distributor body at each branch line connection indicate which cylinder each pilot valve serves.

Timing of the air starting distributor valves is accomplished by positioning the distributor camshaft. The camshaft is placed on the upper crankshaft end and rotated until the proper geometrical angle of relation with the crankshaft is made.

The camshaft is then keyed to the upper crankshaft by means of a dowel pin. This timing is done at the factory. Replacement camshafts have two dowel pin

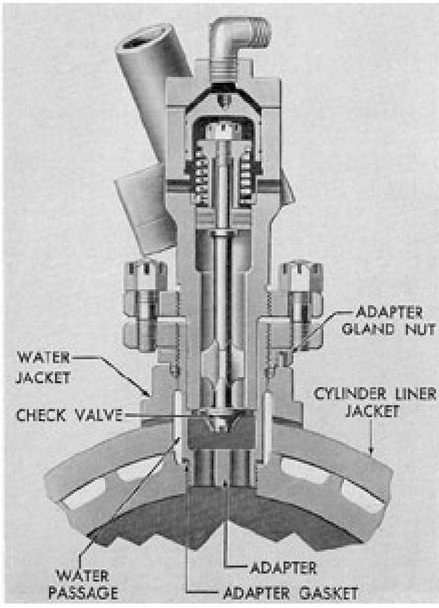


Figure 4-15. Cross section of installed air starting check valve, F-M.

header into the combustion chamber of the cylinder to move the pistons apart and turn the crankshafts. As the individual distributor pilot valve closes, pressure on the operating piston is released, and spring action closes the check valve. When the check valve is closed, the pressure in the pilot lines is vented back through the closed pilot valve and does not

holes for properly locating the camshaft. The pin is placed in one hole for right-hand rotation engines and in the other for left-hand rotation engines.

4C5. Air starting check valve. The air starting check valves are enclosed in bronze bodies and are located at the combustion chamber for each cylinder. Each check valve assembly fits into a water-cooled adapter.

The air starting check valve is held closed principally by spring tension. Near the middle of the valve stem is a balance piston which also serves as a valve stem guide bearing. During air starting there is a constant supply of air from the air starting main header to the air chamber between the valve head and the balance piston. There is a slightly greater pressure area at the balance piston than at the valve head. This prevents the starting air pressure from opening the valve. An operating piston fits over the end of the valve stem opposite the valve head.

When the individual distributor pilot valve opens, actuating air is brought through an individual pilot air line to the air chamber above the operating piston in the check valve body. Pressure of the actuating air forces the operating piston inward, overcomes the spring pressure, and forces the check valve open. This action admits air directly from the starting air main enter the cylinder combustion chamber.

4C6. Maintenance. Frequent inspections should be made of the air starting system to see that line connections and valves are not leaking. Small leaks at the air start check valve will permit gases of combustion to carbonize and burn the valve seat. Unless this condition is remedied by grinding and reseating the valve, larger leaks with consequent serious damage to the air starting system will result.

5
DIESEL ENGINE FUEL SYSTEMS

A. DIESEL FUELS

5A1. General. Normally, diesel fuel oils for use in the Submarine Service are purchased by the Bureau of Supplies and Accounts. At the time of delivery, the diesel fuel oils are inspected to make sure that they meet the specifications set up by the Bureau of Ships. However, emergencies occasionally arise both in the supply and in the handling of diesel fuels that make it imperative for operating engineering personnel to have at least a fundamental knowledge of the requirements for diesel fuel oil.

5A2. Cleanliness. One of the most important properties necessary in a diesel fuel oil is cleanliness. Impurities are the prime sources of fuel pump and injection system trouble. Foreign substances such as sediment and water cause wear, gumming, corrosion, and rust in the fuel system. Diesel fuel oil should be delivered clean from the refinery. However, the transfer and handling of the oil increase the chance of its picking up impurities. The necessity for periodic inspection, cleaning, and care of fuel oil handling and filtering equipment is emphasized under the subject of maintenance for each system.

stopped at any point, leaving a residue of a heavier viscous liquid. This residue may be cracked in cracking stills by the application of heat and pressure in the presence of a catalyst. This cracking process may be controlled so as to get products of almost any given type of hydrocarbon molecular structure. The products mostly desired are those that can be used as gasoline and fuel oil blends.

Fuel oils that meet the specifications for high-speed diesel engine operation are of two types, distillate and blended. The distillate type is obtained by the direct distillation of crude oil only. Blended type is obtained by blending the distillate with the residual products from the cracking stills. As a general rule, distillate fuel oil is superior to blended fuel oil for high-speed diesel operation because it possesses better ignition quality, has a lower carbon content, and contains fewer impurities.

American crude oils are classified into three types: paraffin base, asphalt base, and mixed base. These three classifications depend upon whether paraffin waxes, asphalt, or both remain after all the removable hydrocarbons have been distilled from the petroleum.

5A3. Chemistry of diesel fuel oil. Diesel fuel oils are derived from petroleum, more generally known as crude oil. All crude oils are composed of compounds of carbon and hydrogen known as hydrocarbons. The structure of the oil is made up of tiny particles called molecules. In crude oil, a molecule consists of a certain number of atoms of carbon and a certain number of atoms of hydrogen. The ratio between carbon and hydrogen atoms in a molecule determines the nature of the crude oil.

Crude oil is separated into various products by a process known as fractional distillation. In general, each product is obtained at its particular boiling point in the distillation process. The relative order of products obtained, with their distillation temperature is:

Gasoline-100 degrees to 430 degrees F
Kerosene-300 degrees to 500 degrees F
Fuel oil-400 degrees to 700 degrees F
Lubrication oil-650 degrees F

The fractional distillation process may be

5A4. Differences in internal combustion fuels. The two principal types of internal combustion fuels are gasoline and diesel fuel oil. Both types are hydrocarbons, but the hydrocarbons differ radically in their chemical composition.

Gasoline is a fuel adapted to spark ignition, while diesel fuel oil is adapted to compression ignition. In spark ignition, the fuel is mixed with combustion air before the compression stroke. In compression ignition, the fuel is injected into the combustion air near the end of the compression stroke. Thus a spark-ignition fuel must have a certain amount of resistance to spontaneous ignition from compression heat. The opposite holds true for diesel fuel oils. Entirely different ignition properties are required of the two fuels.

5A5. Properties of diesel fuel oils. The following are the chief properties required of diesel fuel oils. With the definition of each

property is an explanation of its application to engine operation.

a. The ignition quality of a diesel fuel oil is the ease or rapidity with which it will ignite.

A diesel fuel with good ignition quality will auto-ignite (self-ignite) at a relatively low

reference fuel that produced the same standard delay period with the same compression ratio. For example: if the reference fuel required 60 percent cetane and 40 percent alpha-methyl naphthalene to produce the same standard delay period at the same compression ratio as the diesel

temperature. In simple language the fuel will ignite quickly and easily under relatively adverse conditions. Thus, where diesel engines must be started at low temperatures, good ignition quality makes starting easier.

Poor ignition quality will cause an engine to smoke when operating under a light load at a low temperature. It will also often cause the engine to knock and overheat due to the accumulation of fuel in the cylinder between the injection and ignition period. The sudden ignition of accumulated fuel causes the knock.

There are two widely accepted methods of determining the ignition quality of a diesel fuel oil

1. Cetane number test. In this method a standard reference fuel is used in a test cylinder. The most widely used reference fuel is a mixture of cetane and alpha-methyl-naphthalene. Cetane has an extremely high ignition quality (ignites quickly) and is rated for the test at 100. Alpha methyl-naphthalene has a very low ignition quality (is difficult to ignite) and is rated for the test at 0.

The single-cylinder test engine used is like any diesel engine cylinder, except that the compression ratio of the cylinder is adjustable. Other cylinder conditions, including the delay period, that is, the interval between injection and ignition, are held constant. This delay period is measured by electrical equipment. The fuel to be tested is used in the test cylinder and fuel oil tested, then the cetane rating of the diesel fuel oil is 60.

NOTE. The cetane rating for gasoline indicates low ignition quality while cetane rating for diesel fuel oil indicates relatively high ignition quality. Cetane numbers of diesel fuels in use today range from about 30 for engines least critical to fuel to over 60 for the highest ignition quality fuels.

2. Diesel index. This method of determining ignition quality is obtained by a simple laboratory test. This test takes into account the fact that there is a definite relationship between the physical and chemical properties of diesel fuel oils and their ignition quality. The diesel index number method is based on the relation between the specific gravity of the fuel oil and the aniline point, which is the temperature in degrees Fahrenheit at which equal quantities of the fuel oil and aniline (a chemical derived from coal tar) will dissolve in each other. To obtain the diesel index number, the gravity of the fuel oil, in degrees API, is multiplied by the aniline point and divided by 100. The result is the diesel index number of the fuel.

While the diesel index method is accepted as a fairly reliable method of determining the ignition quality, the cetane number test is considered more accurate. Hence it is preferable to use the cetane number test where possible. It must be remembered, however, that the diesel index test possesses the advantage of simplicity and low cost. The normal range of diesel index is

the compression ratio is adjusted until the standard length delay period is reached. Fuel with high ignition quality requires a low compression ratio. Fuel with low ignition quality requires a high compression ratio.

Next the reference fuel is used in the cylinder. Using the same compression ratio, various mixtures or proportions of cetane to alpha-methyl-naphthalene are used until the standard length delay period is attained. The cetane number of the diesel fuel oil tested is then equal to the percentage of cetane in the from below 20 to about 60 for diesel fuels in use.

b. Specific gravity. The specific gravity of a diesel fuel oil is the ratio of its weight to the weight of an equal volume of water, both having the same temperature of 60 degrees F. The specific gravity of the majority of diesel fuel oils ranges from 0.852 to 0.934. As a matter of convenience and to standardize reference, the American Petroleum Institute has established the API gravity scale calibrated in degrees for diesel fuel oil

gravities. Lighter weight fuel oils have high numbers (about 20 degrees to 40 degrees) and heavier weight fuel oils have low numbers (from 10 degrees up to about 20 degrees).

Diesel fuel oils are generally sold by volume. Hence the specific gravity of a fuel oil plays an important part commercially. Knowing the specific gravity, temperature, and quantity of a fuel oil, the volume can easily be computed from standard tables. The specific gravity of a diesel fuel oil is often referred to, but its significance is frequently overestimated. Efforts have been made at various times, but with little success, to establish a definite relationship between gravity and other characteristics such as viscosity, boiling point, and ignition quality.

heat value than a pound of the heavy oils, a gallon of the former is generally lower in heat value than a gallon of the latter. The difference, however, in the normal range of diesel fuels is relatively small. For example, a 24 degrees API diesel fuel has approximately 3 percent greater heating value per gallon than a 34 degrees API fuel. Considering the many factors related to gravity which may affect over-all thermal efficiency, the effect of this difference on fuel economy is usually negligible.

e. Flash point. The flash point of an oil is the lowest temperature at which a flash appears on the oil surface when a test flame is applied under specified test conditions. It is a rough indication of the tendency of the product to vaporize as it is heated. The flash point is important primarily with relation to regulations covering handling and storing of

c. Viscosity. The viscosity of a fluid is the internal resistance of the fluid to flow. The viscosity of a fuel oil is determined by the Saybolt Universal Viscosimeter test. In this test, a measured quantity of the fuel oil is allowed to pour by gravity through an opening of established diameter and with the fuel oil at an established temperature, usually 100 degrees F. The length of time in seconds required for the given quantity of fuel oil to pass through the opening determines its viscosity.

Viscosity is important in diesel fuels because of its effect on the handling and pumping of the fuel, and on the injection of the fuel. Viscosity, together with the rate of fuel consumption, determines the size of fuel lines, filters, and fuel pumps. The efficiency of filtering is greatly increased in a fuel oil of lower viscosity. In the injection system viscosity affects the characteristics of the fuel spray at the injection nozzles. It also affects the amount of leakage past pump plungers and valve stems, and therefore the lubrication of the various types of valves and pumps.

d. Heating value. The heating value of a diesel fuel oil is its ability to produce a specific Btu output of heat per unit of weight or volume. There is a definite relation between the gravity of a diesel fuel oil and the Btu content. The relationship is approximately:

Btu per pound of fuel = 17,680 + 60 x API gravity.

inflammable liquids. It is of little importance to diesel fuel oil performance. Most diesel fuels have a flash point well above 180 degrees F. The minimum flash point required by Navy specifications is 150 degrees F.

f. Pour point. The pour point of a diesel fuel is the temperature at which the fuel congeals and will no longer flow freely. This is usually due to the presence of paraffin wax, which crystallizes out of the fuel at low temperatures. Pour point usually determines the minimum temperature at which the fuel can be handled, although in some cases, where there is considerable agitation preventing the crystallization of wax, the fuel will usually flow at temperatures below the pour point.

g. Carbon residue. The carbon residue of diesel fuels is usually determined by the Conradson test, in which the fuel is burned in a covered dish. The carbon remaining is weighed and expressed as a percentage of the fuel. The test provides a rough indication of the amount of high-boiling heavy materials in the fuel, and is particularly useful where, because of high boiling points, distillation data cannot be obtained. Carbon residue is sometimes taken as an indication of the tendency of the fuel to form carbon in the combustion chamber and on the injection nozzles, although there is a little basis for using the test for this purpose due to the difference in the method of combustion used in the test and that actually encountered in an engine.

It is well to remember that although a pound of the lighter grades of oils has a higher

h. Sulphur content. The sulphur content of a diesel fuel includes both noncorrosive and corrosive forms of sulphur. If the sulphur content is high, the copper strip corrosion test should be made to determine whether or not the sulphur is in corrosive form. If sulphur in corrosive form is present, a sample of the oil should be sent to the nearest laboratory facility for a test to determine the percentage present. Sulphur in excess of Navy maximum specifications is likely to damage the engine. When the fuel is burned, the sulphur is combined with oxygen to form sulphur dioxide which may react with water produced by combustion to form sulphuric acid and cause excessive cylinder wear. It will also act to corrode other internal engine parts.

i. Ash content. The ash content of a diesel fuel oil is the percent by weight of the noncombustible material present. This is determined by burning a quantity of fuel of known weight and weighing the ash residue. Ash is an abrasive material and the presence of ash above the maximum amount allowed by Navy specifications will have an obvious wearing effect on engine parts.

j. Water and sediment. The percent by volume of water and precipitable sediment present in the fuel oil is determined by sediment to separate. The percentage by volume is then determined.

The presence of water and sediment is generally an indication of contamination during transit and while handling. Fuel containing water and sediment causes corrosion and rapid wear in fuel pumps and injectors.

5A6. Engine troubles caused by fuel. As indicated in the discussion of diesel fuel oil properties, any number of engine troubles may be caused by unclean or poor fuel oil. Some of the more common troubles are:

a. Carbon deposits at injection nozzles may be due to excess carbon residue or excessive idling of engine.

b. Excess wear of injection pumps and nozzles may be due to too low a viscosity, excess ash content, or corrosion from water or sulphur content in the fuel oil.

c. Exhaust smoke may result when a fuel with too high an auto-ignition temperature is used. This is particularly true at light loads when engine temperatures are low.

d. Combustion knock in a diesel engine is believed to be due to the rapid burning of a large charge of fuel accumulated in the cylinder. This accumulation is the result of nonignition of fuel when it is first

diluting a quantity of fuel oil with an equal quantity of benzol, which is then centrifuged, causing water and injected into the cylinder, a condition usually caused by fuel oil of poor ignition quality.

B. SHIPS FUEL SYSTEM

5B1. General. The engineering installation on present fleet type submarines consists of four main engines and one auxiliary engine. These are divided between two engine rooms, with two main engines in the forward engine room, and two main engines and the auxiliary engine in the after engine room. The function of the ship's fuel oil system is to supply clean fuel oil to each engine from the ship's storage tanks. The system may be divided into two parts: 1) the tanks and their arrangement, and 2) the different piping systems.

The tanks include normal fuel oil tanks, fuel ballast tanks, clean fuel oil tanks, expansion tank, and collecting tank. All of these tanks are in the spaces between the inner pressure hull and the outer hull of the submarine with the exception of the clean fuel oil tanks which are inside the pressure hull.

The two main piping systems found in the main fuel-oil system are the fuel oil filling and transfer line and the fuel oil compensating water line. These lines connect to the various tanks and give the fuel oil system a flexibility which it otherwise would not have.

5B2. The compensating principle. In order to understand the operation of a submarine fuel system, it is important to know the basic fuel oil compensating principle. In a submarine, to assist in maintaining trim it is necessary to have as little weight change as possible when fuel is being used m a fuel tank. Therefore, a compensating system is used which allows salt water to replace fuel oil as the fuel oil is taken from a tank. Let us assume that the weight of fuel

used is 7.13 pounds per gallon and the weight of salt water is 8.56 pounds per gallon. Therefore, when one gallon of fuel is used from a fuel tank, instead of the submarine-becoming light by 7.13 pounds, it becomes heavy by 8.56 - 7.13 or 1.43 pounds. The submarine, then, becomes heavy as fuel oil is

c. Collecting tank. The collecting tank is one side of a section of tank space between the inner and outer hulls, the other side being the expansion tank. This tank has a connection to the fuel oil filling and transfer line. All of the fuel used by the engines normally passes through the collecting tank. A connection from the top of the

used. This compensating principle is used in the normal fuel oil tanks, fuel ballast tanks, expansion tank, and collecting tank. These tanks must at all times be filled with a liquid, either fuel oil, sea water, or a combination of both. The compensating principle is not used in the clean fuel oil tanks.

5B3. Fuel oil tanks. a. Normal fuel tanks. The normal fuel tanks are used only for the storage of fuel oil. They are usually located toward the extremities of the boat rather than close to amidships. They vary in size, but normally have capacities of from 10,000 to 20,000 gallons each. Most modern submarines have four of these tanks. In a typical installation (Figure 5-1) they are numbered No. 1, No. 2, No. 6, and No. 7.

b. Fuel ballast tanks. Fuel ballast tanks are large tanks, amidships, between the pressure hull and the outer hull, which may be used either as fuel storage tanks or as main ballast tanks. They are connected to the fuel oil system in the same manner as the normal fuel oil tanks, but in addition, they have main vents, main flood valves, and high-pressure air and low-pressure blower connections which are necessary when the tank is in use as a main ballast tank. When rigged as a main ballast tank, all connections to the fuel oil system are secured.

Most fleet type submarines have three fuel ballast tanks varying in capacity from about 19,000 to 25,000 gallons. On a typical installation (Figure 5-1), the fuel collecting tank leads to the fuel oil meters, fuel oil purifiers, clean fuel oil tanks, and eventually to the attached fuel oil pumps on the engines. This tank has a capacity of about 3,000 gallons, and on submarines is located outboard of the forward engine room. The main function of the collecting tank is to insure that no large amount of water gets to the purifiers, clean fuel oil tanks and engine until all the fuel in normal fuel oil tanks, fuel ballast tanks, expansion tank, and collecting tank has been used.

d. Expansion tank. The expansion tank is alongside and on the opposite side of the ship from the collecting tank. It is connected to the fuel oil compensating water line. It serves two important functions: first, as a tank to prevent oil from being blown over the side through the compensating water line in case of small air leaks in either the fuel ballast tanks or the normal fuel oil tanks; and second, as a tank to which oily bilge water may be pumped without danger of leaving a slick. This tank has a capacity of about 3,000 gallons.

e. Clean fuel oil tanks. The clean fuel oil tanks, two in number, are used to store oil prior to its use in the engine and after it has been purified. These tanks are not compensated with compensating water. They have capacities of approximately 600 gallons each.

5B4. Fuel oil piping systems. a. Fuel oil filling and transfer line. The fuel oil filling and transfer line extends the length of the ship and is used for filling the fuel system and transferring the fuel from the

ballast tanks are numbered No. 3, No. 4, and No. 5. Current practice is to depart on war patrol with all fuel ballast tanks filled with fuel oil. Fuel is used first from No. 4 fuel ballast tank, and as soon as that tank is empty of fuel (filled with salt water) it is converted to a main ballast tank. Upon conversion, the tank is flushed out several times to insure that all fuel oil is out of the tank. The conversion of No. 4 FBT to a main ballast tank increases the stability of the submarine and decreases the amount of wetter surface of the hull when on the surface.

various fuel oil tanks to the collecting tank where it can be piped off, purified, and used in the engine. There is a connection from the fuel oil filling and transfer line to the top of each side of each normal fuel oil and fuel oil ballast tank. This may be a direct connection or through a manifold, as shown in Figure 5-1 for normal fuel oil tanks No. 1 and No. 2. There is also a connection from the fuel

---

Figure 5-1. TYPICAL INSTALLATION OF SHIP'S FUEL OIL AND COMPENSATING WATER SYSTEMS.

---

oil transfer line to the bottom of the collecting tank. This is the line through which passes all of the fuel from the main fuel oil tanks. At the forward and after end of the transfer line is a fuel filling line that connects the forward and after fuel filling connections on the main deck with the fuel oil filling and transfer line.

When the fuel system is in use, only one of the normal fuel or fuel ballast tanks is in service at a time. This is made possible by a stop valve in the fuel oil transfer line to the top of each side of each tank. This valve permits all tanks except the one in service to be secured on the fuel transfer line.

b.**Fuel oil compensating water line.** This line runs the length of way of a header box in the conning tower shears, but the amount of water needed to replace the fuel oil used goes down into the compensating water line by way of a four-valve manifold. The header box serves to keep a head of water on the system, insuring that the entire system is completely filled at all times.

The four-valve manifold is really a bypass manifold for the expansion tank. The four valves on the manifold (see Figure 5-2) are used as follows:

Valve A cuts off the four-valve manifold from the header box.

Valve B closes the line from the manifold to the bottom of the expansion tank.

the ship and has a connection to the bottom of each normal fuel oil and fuel oil ballast tank. The salt water that replaces the fuel oil in the fuel tanks comes from the main engine circulating salt water discharge to the compensating water line or, if all engines are secured, from the main motor cooling circulating salt water discharge to the compensating line. Most of this water goes over the side by

Valve C is the bypass valve for expansion. If this valve is open, the compensating water an go directly into the compensating water line without going through the expansion tank. If the valve is closed, the compensating water must go into the compensating water line through the expansion tank. During normal operation this valve is closed.

Valve D closes the line from the manifold to the top of the expansion tank.

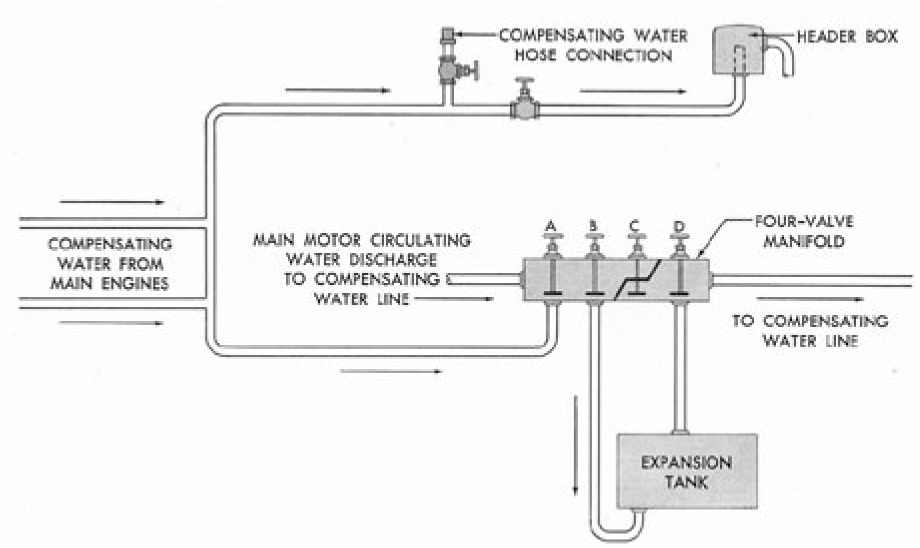


Figure 5-2. Four-valve manifold.

Under ordinary operating conditions, all the valves on the compensating water line to the individual tanks are locked open and valve C is locked closed. This is necessary because sea pressure must be maintained on the inside of the fuel ballast tanks, normal fuel tanks, expansion tank, and collecting tank, when the submarine is submerged. If this were not done, the sea pressure on a deep dive would become so great as to cause a rupture of the relatively weak outer hull. Therefore, it is vital that all the

the header box. It must be emphasized that all the above operations are taking place concurrently and that the entire movement of the liquids is caused by the head of water on the system from the header box.

As soon as the expansion tank is filled with salt water, the salt water comes up to the four-valve manifold through valve D into the compensating water line, and thence into the bottom of No. 4 FBT. As soon as No. 4 FBT is empty of fuel, salt water rises into the fuel oil transfer line and then into the

valves mentioned above be open or closed as indicated. If these valves are properly rigged when the submarine is submerged, sea pressure can enter the system through the header box and then go to the inside of every fuel oil tank except the clean fuel oil tanks, if the valves on the compensating water branch lines to each tank are open. These valves on the individual branch lines are also normally locked open. This maintains the same pressure on each side of the submarine outer hull, insuring that it will not rupture. The valves are always locked to prevent accidental closing or opening.

5B5. Operation of the system. When the submarine is departing on war patrol, all tanks in the fuel oil system are completely filled with fuel. Upon departure, one of the normal fuel oil or fuel ballast tanks will be on service. As soon as fuel is drawn from the top of the collecting tank by means of the fuel oil transfer pump, salt water comes into the bottom of the expansion tank, keeping the system completely filled with liquid.

The path of the water can be traced by referring to Figure 5-1: Assume that No. 4 FBT is in service. As fuel is taken off the top of the collecting tank, fuel comes from the top of No. 4 FBT through the fuel oil filling and transfer line into the bottom of the collecting tank, replacing the fuel taken from the top of that tank. At the same time the fuel taken from the top of No. 4 FBT is replaced by the fuel from the top of the expansion tank by way bottom of the collecting tank. This is a positive indication that the No. 4 FBT has no more fuel in it. In order to tell when the salt water reaches the collecting tank, a liquidometer age which reads directly the amount of fuel in the tank is placed on the collecting tank. As soon as this gage reads less than completely filled, it is evident (in this case) that No. 4 FBT has no more fuel. No. 4 FBT is then secured on the fuel transfer line and another fuel tank is placed on service. The small amount of water may be left in the bottom of the collecting tank, as fuel oil that comes into the tank will rise through the water to the top of the tank. The water normally is left in the bottom of the collecting tank until the ship is refueled. At that time the water is withdrawn by pumping it out with the drain pump through the drain line to the bottom of the collecting tank.

5B6. Blowing and venting of fuel tanks. Each side of each tank is provided with blow connections which connect to the ship's low-pressure 225-pound air line. In an emergency or to effect repairs, it is often necessary to blow a fuel tank completely clear of all liquids. This is done by closing the tank's stop valves to the fuel oil transfer line and blowing the fuel or water over the side or to another tank (through the compensating water line).

The air line from the blow valve to the tank also has a connection to permit venting of the tank if some air has accumulated in its top or if it is desired to fill a completely empty tank with oil or water. All fuel tanks are equipped with either

of the four-valve manifold, the compensating water line, and the compensating water branch line to the bottom of No. 4 FBT. The fuel oil drawn from the top of the expansion tank is replaced by salt water entering the bottom of the expansion tank by way of the four-valve manifold and the line to liquidometer gages or sampling cocks. These sampling cocks are used to take samples of liquid at various fixed levels in the, tank in order to ascertain approximately the

amount of fuel in the tank. The liquidometer gages are adjusted so as to read directly the number of gallons of fuel in the tank.

5B7. Liquidometers. In submarine fuel systems, liquidometers are used to determine:

1) the level of oil in partially filled tanks, such as clean fuel oil tanks, and

2) the level between fuel oil and salt water in completely filled tanks such as normal fuel tanks, fuel ballast tanks, collecting tank, and expansion tank.

The liquidometer is equipped with a float mechanism, the movement of which activates a double-acting opposed hydraulic mechanism which registers upon a properly calibrated dial the volume of oil in a tank in gallons.

The float of a liquidometer used in compensated fuel tanks is usually filled with kerosene to a point where it will float in water but sink in fuel oil. Since the water is below the oil, the float will sink through the oil and stop at the compensating water level.

units, a tank unit located in the tank whose capacity is to be measured, and a dial unit located at some distant point away from the tank (such as in the control room of a submarine). Operation of the instrument is dependent upon the movement of the float in the tank which is mechanically connected to an upper and lower bellows of the tank unit. These two bellows are rigidly supported at one end by a bracket, and both are connected by tubing to two similar bellows in the dial unit. The dial unit bellows are each supported at one end by a bracket which also provides a bearing connection for the indicator pointer. The free ends of the bellows, facing the pointer, are connected to a link which actuates the pointer. When the float moves down, the mechanical linkage between the float arm and the upper and lower tank bellows compresses the lower bellows, forcing a portion of the liquid from it into the interconnected dial unit bellows, causing it to expand. At the same time, the upper bellows in the tank unit is being elongated through the mechanical

The instrument consists essentially of two

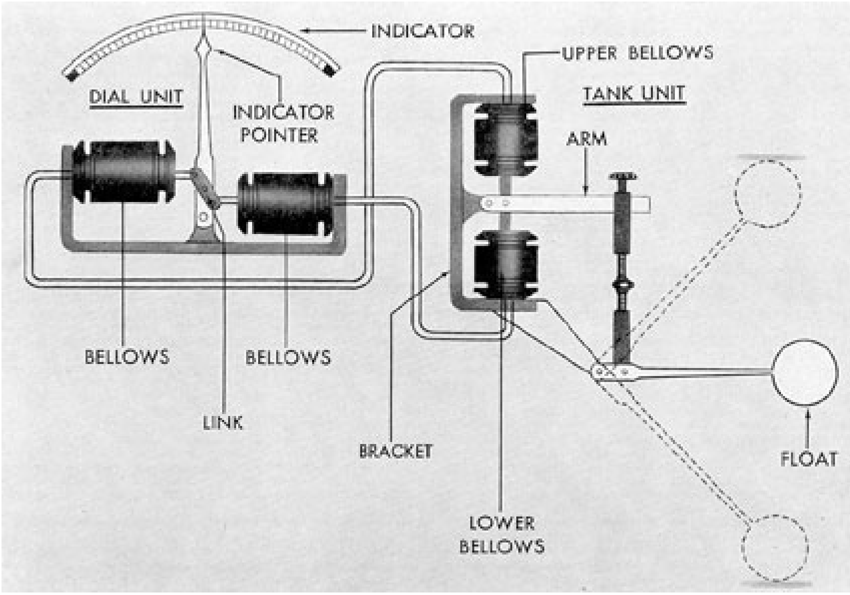


Figure 5-3. Schematic diagram of liquidometer.

connection to the float arm and takes in a portion of the liquid from the other dial unit bellows, which is then caused to contract. Reverse action takes place if the tank float moves upward.

5B8. Maintenance of ship's fuel system. All fuel storage tanks should be periodically inspected and cleaned. This is usually done during submarine overhauls at naval shipyards.

All screen strainers used in connection with the fuel oil system should be periodically removed and cleaned.

The valve seat gaskets used in the fuel ballast tanks are made of special, oil-resisting rubber. These gaskets should be inspected at each filling and replaced if deteriorated or damaged.

In the fuel ballast tanks, all valves are enclosed in galvanized wire

submarines, the connection between the compensating water line and the four-valve manifold is provided with a plug protected sight glass to check the pipe's contents. This glass should be kept in clean and readable condition at all times. In most modern fleet type submarines this sight glass has been blanked off because of possible breakage during depth charge attack.

It is essential that all air be excluded from the fuel system, or the system may become air-bound, thus preventing proper flow of oil to the engines and also disturbing the trim of the submarine. This may be done by venting the system through the vent facilities provided.

In venting fuel tanks in use, the following order should be observed: first, the expansion tank, then the fuel tank on service, then the collecting tank. The remaining fuel tanks may then be vented in

mesh screens. These wire mesh screens should be cleaned whenever inspection indicates that it is necessary. On some any order. The discharge line from the collecting tank to the clean fuel oil tank should be closed during venting operations.

C. SUPPLY FROM SHIP'S FUEL SYSTEM TO ENGINE FUEL SYSTEMS

5C1. General. After leaving the collecting tank, fuel is piped through a system comprised of strainers, fuel meters, fuel oil transfer pumps, purifiers, and clean fuel oil tanks before reaching the engine. This section of the fuel oil system is divided into two parts. One part serves the forward engine room, the other the after engine room. The two are interconnected to provide flexibility of operation.

5C2. Strainers and meters. Fuel oil to be used in the engine is normally taken from the top of the collecting tank. It may, however, in some installations, be drawn directly from the fuel oil filling and transfer line. In either case, the oil should go through a wire mesh type strainer and fuel meter before entering the suction side of the fuel oil transfer pump. Both strainer and meter are fitted with bypass connections by means of which a strainer, or meter, or both may be bypassed.

5C3. Fuel oil transfer and purifier pumps. Located in each engine room is a positive displacement type fuel oil transfer and purifier pump, driven by an electric motor. The primary function of this pump is to transfer fuel oil from the collecting tank to the clean fuel oil tank through the purifier. It may also be used for priming purposes by taking a suction from the clean fuel oil tank and delivering the priming oil to the individual engine fuel system. An engine normally is primed before starting, particularly if it has been secured for some time.

Under normal operating conditions this pump is operated until the clean fuel oil tanks are full. It is then secured until the level of oil in the clean fuel oil tanks becomes such as to indicate need for replenishment.

5C4. Pure oil purifiers. a. General. The fuel oil purifiers are Sharples centrifuge units which operate on the principle of centrifugal force.

Centrifugal force is the force exerted upon a body or substance by rotation that impels that body or substance outward from the axis of rotation. When a mixture of liquids is revolved at high speed in a container, the centrifugal force causes the components of the liquid to separate. The component with the greatest specific gravity will assume the outermost position, and the lightest component, the innermost position. Thus, if a mixture of water and oil is revolved, the water, being the heavier component, will separate from the lighter oil and form

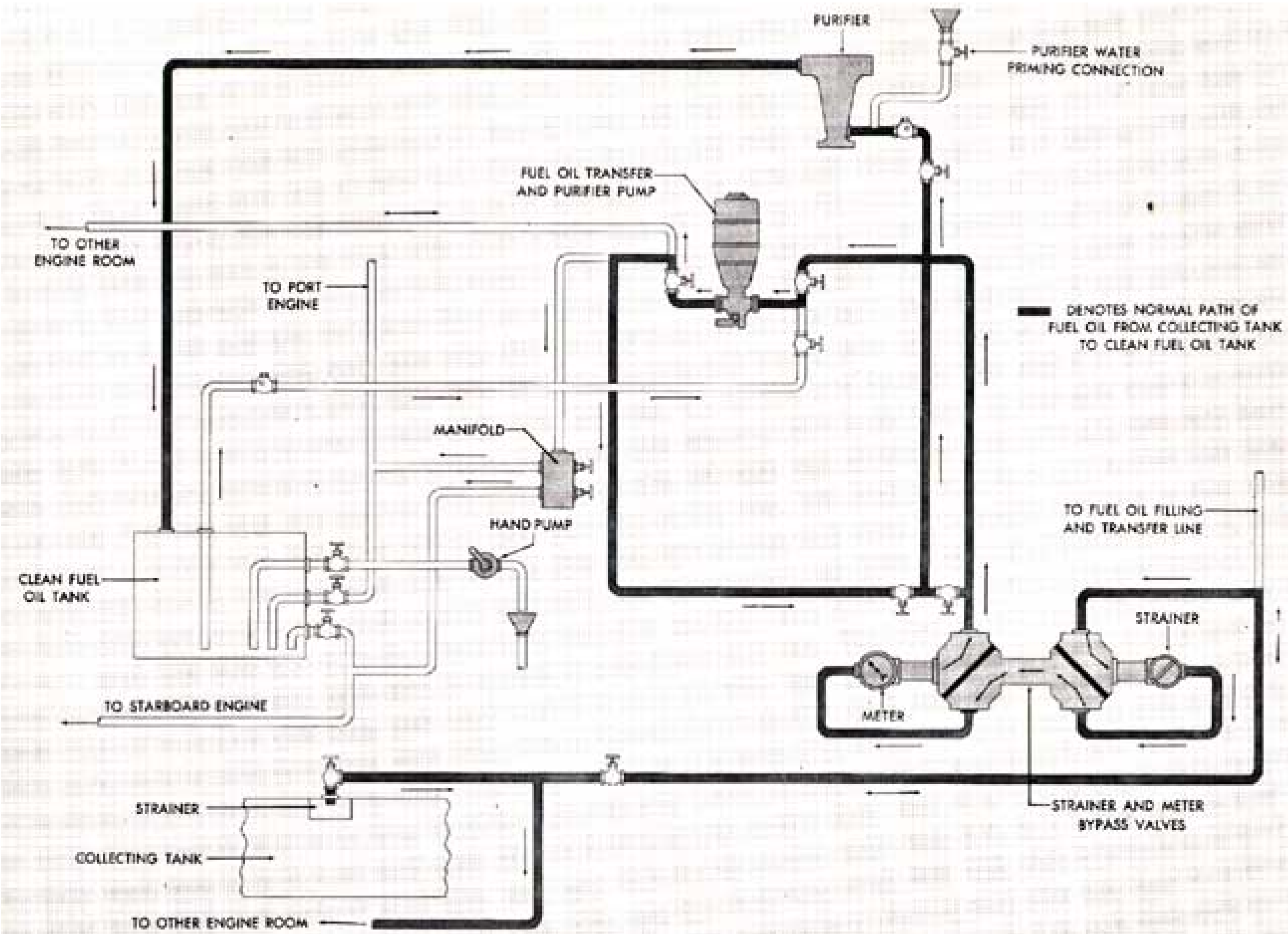


Figure 5-4. Fuel oil supply from ship's fuel system to engine fuel system in one engine room.

a layer around the wall of the container, while the oil remains near the center of the container. The Sharples fuel oil purifier operates on this principle.

The Sharples purifier can be used as a separator or a clarifier. When used as a separator, the purifier separates oil from water and solid sediment. When used as a clarifier, it separates oil from solid sediment only. The unit is usually set up as a separator in fuel oil systems and a clarifier in lube oil systems. (See Section 7B7.)

b. Operation. The fuel oil transfer and purifier pump forces fuel oil through a short connecting line at the bottom of the purifier reduce the effect of the centrifugal force.

When the machine is operated as a separator, the bowl is primed with fresh water until an effective water seal is created at the water discharge outlet. The water priming line is sealed off from the fuel inlet line by means of a check valve which prevents water from finding its way into the fuel system. Then the fuel oil supply is forced into the swiftly revolving bowl. The centrifugal force throws the water, which has a heavier specific gravity than the oil, to the outside wall of the bowl and creates a vertical layer of water at this outer extremity. The fuel oil, which has a lighter specific gravity, forms a layer next to the water.

bowl. The purifier bowl is revolved by an attached electric motor at about 15,000 rpm. A three-wing partition extends the full length of the bowl on the inside. The purpose of this partition is to keep the liquid in the bowl revolving with the bowl. Otherwise there would be slippage of the liquid column which would

Any particles of sediment in the fuel oil have a heavier specific gravity than either the water or oil and are drawn and held against the wall of the bowl by the centrifugal force. Dirt and sediment are cleaned out of the bowl when necessary.

At the top of the purifier bowl is a barrier called a ring dam, which covers the top of the

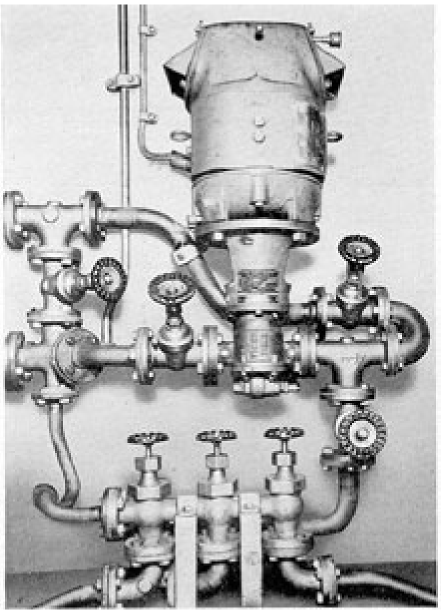


Figure 5-5. Fuel oil transfer and purifier pump.

vertical column of water and fuel oil. There is an opening at the outer diameter of the ring dam through which only excess water is discharged. At the inner diameter of the ring dam is another opening through which only purified fuel oil discharges. Thus, as long as the centrifugal force and the effective water seal are maintained, it is impossible for fuel oil to displace the water and get out through the water discharge opening. It is just as impossible for water to get out through the fuel oil discharge

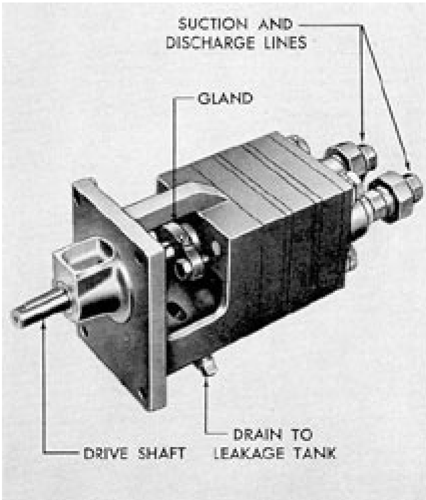


Figure 5-6. Attached fuel oil supply pump, F-M.

oil can be pumped to either fuel oil purifier by means of the transfer and purifier pumps and discharged to either clean fuel oil tank. Also, the transfer and purifier pump may be used to draw fuel oil from either clean fuel oil tank and supply any engine directly, during priming operation.

A hand pump is connected to the clean fuel oil tanks to provide a means of checking the contents of the tank for water, for testing the quality of the oil, and for removing residual oil in the tank when it is desired to clean it.

Each engine in a compartment is connected to the clean fuel oil

opening as long as the centrifugal force is in effect.

5C5. Clean fuel oil tanks. All fuel oil supplied to the engines is normally drawn from the clean fuel oil tanks. There are two clean fuel oil tanks, one in the forward engine room and one in the after engine room. Under normal operating conditions, the engines in each compartment draw their supply from the clean fuel oil tank in that compartment.

Each tank averages about 600 gallons capacity in fleet type submarine installations. By means of a system of valves and piping, fuel tank in the same compartment by a fuel line which goes from the bottom of the clean fuel oil tank up to the attached fuel oil pump on the engine. The attached fuel oil pump takes a suction from the clean fuel oil tank and delivers the oil to-the engine fuel system. If the attached fuel oil pump on one engine should become inoperative, it is possible to connect the fuel oil transfer and purifier pump so as to supply fuel up to the engine, thereby preventing a shutdown of the engine.

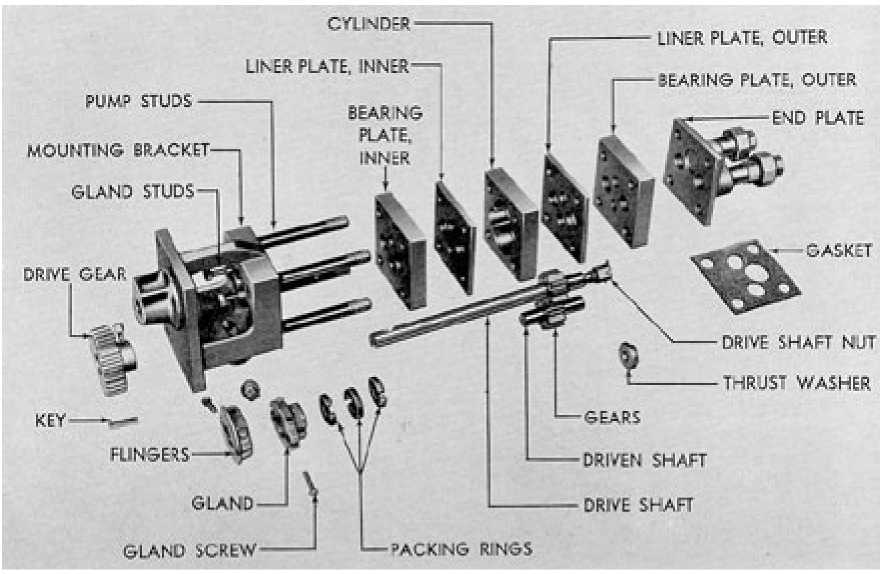


Figure 5-7. Exploded view of attached fuel all supply pump, F-M.

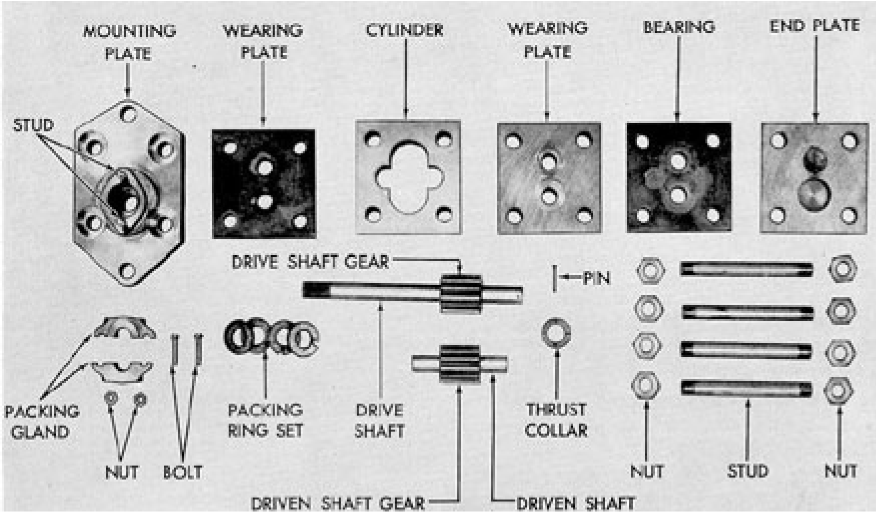


Figure 5-8. Exploded view of attached fuel oil supply pump, GM.

Each clean fuel oil tank is equipped with a liquidometer to measure the quantity of fuel oil in the tanks at all times.

5C6. Attached fuel oil supply pump, F-M. The attached fuel oil supply pump (Figures 5-6 and 5-7) draws fuel by suction from the clean fuel oil tank and delivers it through the strainer and filter units to the engine main fuel oil header.

The pump is a positive displacement type gear pump and is driven directly from the lower crankshaft of the engine through a flexible gear drive. A packing gland is provided on the fuel oil pump drive gear shaft to prevent fuel oil from leaking out around the shaft.

5C7. Attached fuel oil supply pump, GM. The function of the GM attached fuel oil supply pump is the same as that of the pump described in section 5C6 above. This pump is also of the positive displacement type, but it is driven directly from one of the engine camshafts instead of the crankshaft as on the F-M engine.

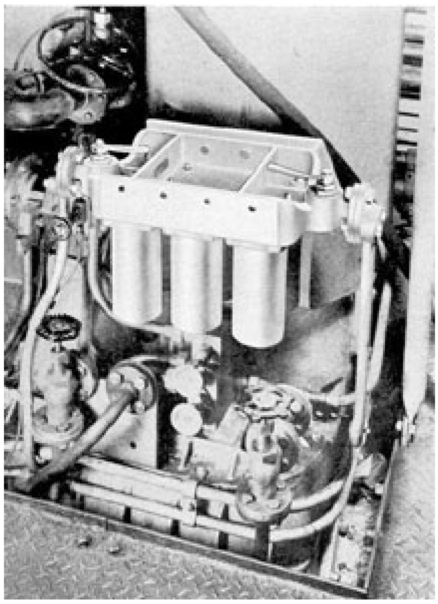


Figure 5-9. Fuel oil filter.

Each strainer consists of a body or case which is fitted with a metal ribbon wound element. A scraper device with long blades that contact the inside surface of the element is fitted into each strainer. A handle for turning the element extends through the top of the strainer so that the operator may occasionally turn the element, thereby cleaning accumulated dirt from the surface of the element. Dirt and sediment drop to the bottom of the case and should be removed at regular cleaning periods.

The pump drive shaft is provided with a packing gland to prevent fuel oil from leaking around the shaft.

Fuel oil is drawn from the clean fuel oil tank by suction created by the pump and fed into the pump housing through an inlet at the top of the pump. Oil is forced from the outlet at the bottom of the pump into the engine supply line. A pressure regulating valve in connection with the pump may be set to maintain a pressure of 40-50 psi in the engine fuel system. A pressure relief valve may be set at slightly above the desired pressure to bleed off excess fuel oil when the pressure exceeds the maximum setting. This oil returns to the clean fuel oil tank.

5C8. Duplex fuel oil strainer. All fuel oil delivered to the engine fuel header by pressure from the attached pump must pass through a duplex type strainer. This strainer actually consists of two strainer elements which may be used either individually or in pairs. The flow of fuel oil through either or both strainers is controlled by a manually operated valve. When the valve is set to bypass one strainer, the bypassed element may be removed and cleaned without disturbing the flow of fuel oil to the engine.

Each duplex strainer is equipped with a duplex pressure gage which measures the pressures of the fuel oil fed into the strainer and of the oil leaving the strainer. A drop of 10 psi between the inlet pressure and the outlet pressure indicates that the element or elements of the strainer needs cleaning. Each strainer has a small valve at the top of the case for venting air from the unit.

5C9. Duplex fuel oil filter. Most installations are equipped with duplex fuel oil filters as well as strainers. In function and operation the

filters are similar to the strainers. In the duplex filter, the element is a removable absorbent type cartridge which is removed and thrown away when it becomes particles of dirt and foreign matter. The filter elements are not equipped with scrapers. They should be examined when the pressure registered by the duplex

dirty. The absorbent type filter cartridge is a denser element than the strainer element and consequently filters out finer pressure gage drops a specified value. If found dirty, they must be removed and replaced by a new element.

D. FUEL INJECTION SYSTEMS

5D1. Basic requirements of a fuel injection system. The primary function of a fuel injection system is to measure accurately, vaporize, and inject the fuel at the proper time according to the power requirements of the engine.

In order to accomplish this there are certain basic requirements that any fuel injection system must fulfill.

a. It must measure or meter the fuel. The quantity of fuel injected determines the amount of energy available to the engine through combustion. The brake mean effective pressure and hence, economy, are dependent to a great extent upon the air to fuel ratio. Thus, it is important that the fuel injection system accurately measure the correct quantity of fuel according to engine requirements.

b. It must time the injection. The injection timing has a pronounced effect on engine performance. Early injection tends to develop high cylinder pressures, because the fuel is injected during the part of the cycle when the piston is traveling slowly and therefore the combustion takes place at nearly constant volume. Extremely early injection will cause knocking. Late rejection tends toward decreasing the mean effective start slowly so that a limited amount of fuel will accumulate in the cylinder during the initial ignition lag before combustion commences. It should proceed at such a rate that-the maximum rise in cylinder pressure is moderate, but it must introduce the fuel as rapidly as permissible in order to obtain complete combustion and maximum expansion of the combustion product.

d. It must properly atomize the injected fuel. The fuel must be injected into compressed air in the combustion chamber with sufficient force to accomplish thorough atomization. Atomization reduces the fuel to minute particles or globules. In general, the smaller the particles of fuel the shorter will be the delay period, that is, the interval between injection and ignition.

Opposed to this requirement is the fact that the smallest particles of fuel have a low penetrating quality. Therefore, with very fine atomization there is a tendency toward incomplete mixing of the fuel and air which leads to incomplete combustion.

e. It must inject fuel with sufficient force for effective penetration and distribution. Fuel must be atomized into sufficiently small particles to produce a satisfactory delay period. However, if the atomization process reduces the fuel to too small particles, they will

pressure of the engine and consequently lowering the power output. Extremely late injection tends toward incomplete combustion resulting in a smoky exhaust.

A more complete description of these fuel injection systems is contained in the Bureau of Ships publications entitled: Fairbanks-Morse Fuel Injection Systems Maintenance Manual, NavShips 341-5019; and General Motors Diesel Fuel Injector Maintenance Manual, NavShips 341-5018.

c. It must control the rate of feed during injection. The rate of injection is important because it determines the rate of combustion and influences the engine efficiency. Injection should

lack penetration. This lack of penetration results in igniting of the small particles before they can be injected far enough into the area of the combustion chamber. Consequently, injection pressure must be of sufficient force and the orifice properly proportioned to effect good penetration.

The fuel spray must also be directed by the spray tip to secure a uniform distribution of the spray charge over the entire combustion area.

High turbulence in the combustion chamber causes a more thorough mixing of the fuel and air and aids in complete combustion.

5D2. Types of fuel injection systems. The earlier diesel engine fuel systems utilized the air injection principle in their design. This method of injection consisted of furnishing both fuel oil and air to an injector valve. The high-pressure air carried the fuel into the cylinder where it was burned. Engines using air injection usually developed a high combustion efficiency because of the efficient mixing of fuel and air possible in such a system. However, a considerable amount of high-pressure air was necessary to inject the mixture into the cylinder against the compression pressure present in the cylinder. The necessary air pressure was usually supplied by an attached air compressor.

branch line. In the General Motors engines each pump is an integral part of the unit injector which includes, the injection nozzle. In both engines the injection pump meters the fuel, delivers it to the injection nozzle, and supplies the energy through hydraulic pressure of the fuel oil for the injection and atomization of fuel at the injection nozzle.

b. The fuel injection nozzle. The fuel injection nozzle contains a check valve which may be either needle type or spherical head type. The valve is opened for injection by hydraulic pressure from the injection pump which acts on the differential area of the valve. The pump plunger forces fuel oil through the orifices of the spray

These air compressors used a large percentage (10 to 15 percent) of the power developed by the engine, and, in addition, it was difficult to maintain them in proper operating condition.

In order to increase the reliability and compactness of these older engines, it became necessary to do away with the attached air compressor and shift to the solid injection system in which the fuel alone was injected into the cylinder in a fine atomized spray. This type of injection requires a higher grade fuel than did the air injection system. Solid injection engines are in general more powerful for their size, more simple in construction, and more reliable than their air injection predecessors. Also the total weight per horsepower of the engine is much less. All of our present modern submarine engines operate on the solid injection principle.

5D3. Components of the solid fuel injection system. The solid fuel injection systems under discussion may wary in design but they are alike in principle. The components of the mechanical fuel injection system are:

a. The fuel measuring or injection pump. These pumps are usually of the plunger type and are operated from cams on the engine camshafts through a rocker lever or push rod assembly.

A separate pump (or pumps) is used for each cylinder of the engine. In the Fairbanks Morse OP engines each pump is a tip, atomizing the fuel delivered into the combustion chamber. The injection is timed at the pump, not at the injection nozzle.

c. High-pressure fuel oil lines. Valve opening pressures up to 3,000 psi are encountered in many fuel injection systems, necessitating the use of high-pressure lines. Such tubing should meet the following requirements:

1. It should be of uniform inside diameter, otherwise the injection characteristics will be seriously impaired. For example, if the inside diameter of the tubing should occasionally run smaller than that specified, excessive pressures are likely to result. Where inside diameters exceed specifications, the pressures will drop and there is the possibility that the tubing will develop structural weakness.

2. It should possess sufficient and uniform strength to withstand pressures up to 9,000 psi without yielding.

3. It should have high ductility to permit easy bending to the desired shape and cold swaging without cracking. Bending of the tubing does not affect the injection characteristics as long as the bends do not have a radius of less than 1 1/2 inches.

4. It should have a smooth, accurate bore, absolutely free from scale, seams, laps, laminations, deep pits, or other serious defects which would weaken the structure of the metal or cause restrictions to the flow of the fluid.

separate unit connected to the fuel injection nozzle by a

E. GENERAL MOTORS ENGINE FUEL OIL SYSTEM

5E1. General. The attached fuel oil pump draws fuel oil from the supply tank and forces it through the fuel metering block, the strainer, and the filter. From the filter, the fuel oil flows to the fuel supply manifold, which is the bottom tube of the multiple manifold assembly on each cylinder bank, through a tube to a single jet filter on each cylinder head. This filter is a metal ribbon wound type with passages of approximately 0.001 inch in the element. From the filter, the fuel flows through the jumper tube that supplies the injector. The injector inlet contains a filter to further prevent solid matter from reaching the spray valve.

Two relief valves in the fuel metering block limit the fuel oil pressure in the system. Any excess oil is bypassed back to the clean fuel oil tank.

Surplus fuel from the injector flows through a filter in the outlet passage so that any reverse flow of fuel cannot carry dirt into the injector. The surplus fuel passes from the injector through a jumper tube to the bleed manifold which is the middle tube in the multiple manifold assembly on each cylinder bank. The fuel from the bleed manifold on each bank flows through a metering valve at the injector end of the rocker lever. The quantity of the fuel injected into each cylinder (and therefore the power developed in that cylinder) is varied by rotating the plunger by means of the injector control rack. A rack adjustment, called the micro-adjustment and located on the control linkage, permits balancing the load of each cylinder while the engine is running.

The unit injector is comprised of the various parts illustrated in Figure 5-11. Of these, the principal parts are the body, spray valve nut, bushing, plunger, needle valve or spherical type check valve (depending on the type of injector), valve spring, and the spray tip.

The injector body is a heat-treated, alloy steel forging with two flat surfaces extending on opposite sides for holding the injector in a vise when necessary. These surfaces are drilled in line to support a part of the injector control linkage. A small vent, just below the holding down clamp seat, allows leakage fuel which serves as the lubricant for the plunger and bushing to escape from the plunger spring chamber. This hole also serves as a breather opening to prevent pumping action by the plunger follower. On some injectors, plunger pump fuel leakage flows through a hollow

in the metering block, then back to the clean fuel oil tank.

5E2. The unit injector. a. Description. On the GM engine the fuel injection pump and spray valve are combined into a single compact unit called the unit injector, which meters the fuel and also atomizes and sprays it into the cylinder. The unit injector is held in position in a water-cooled jacket in the center of the cylinder head. At the lower end, the injector forms a gastight seal with the tapered seat in the cylinder head. All injectors in the engine are alike and interchangeable. Fuel is supplied through jumper tubes with spherical type gasketless connections.

The pumping function of the injector is accomplished by the reciprocating motion of the constant stroke injection plunger which is actuated by the injector cam on the engine camshaft through the injector rocker lever. The position of the plunger, and thereby the timing, is adjusted by means of the ball stud and lock nut

drain dowel, then through a drilled passage in the cylinder head and back to the clean fuel oil tank.

The bushing is the cylinder for the plunger pumping unit of the injector. It is located and held against turning in the body by a guide pin that fits into a groove at the upper end of the bushing. Two openings in the bushing wall, on opposite sides, serve as the inlet and bypass ports for the fuel oil. The bottom surface of the bushing is lapped to form an oiltight seal against the full injection pressure.

The unit injector pump plunger is made of a special steel, lapped to a close fit in the bore of the bushing. The clearance between the surface of the plunger and bushing is so fine that it is usually measured by forcing a specific amount of oil of a fixed viscosity between the surfaces and measuring the time consumed.

The lower end of the plunger is cut away to form a recess with an upper and a lower helical lip. These helical lips cover and uncover the

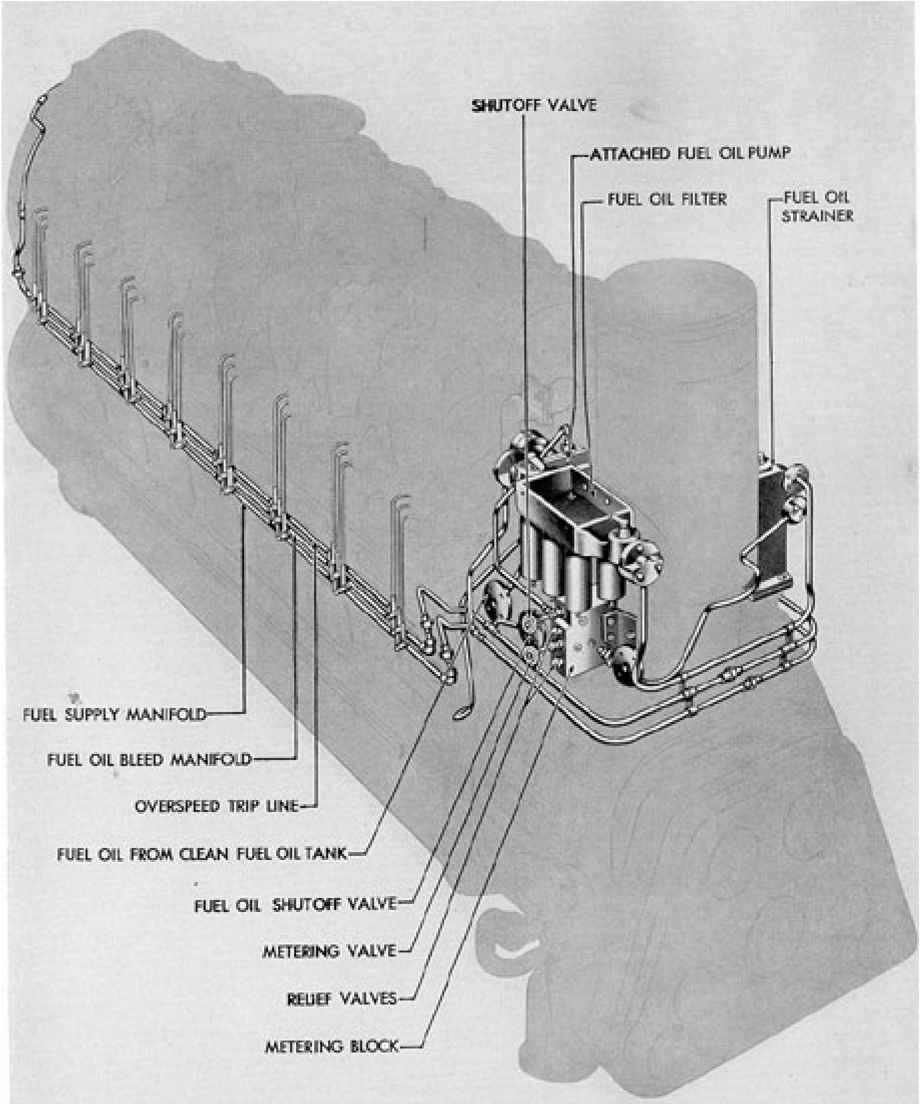


Figure 5-10. Isometric view of fuel injection system, GM.

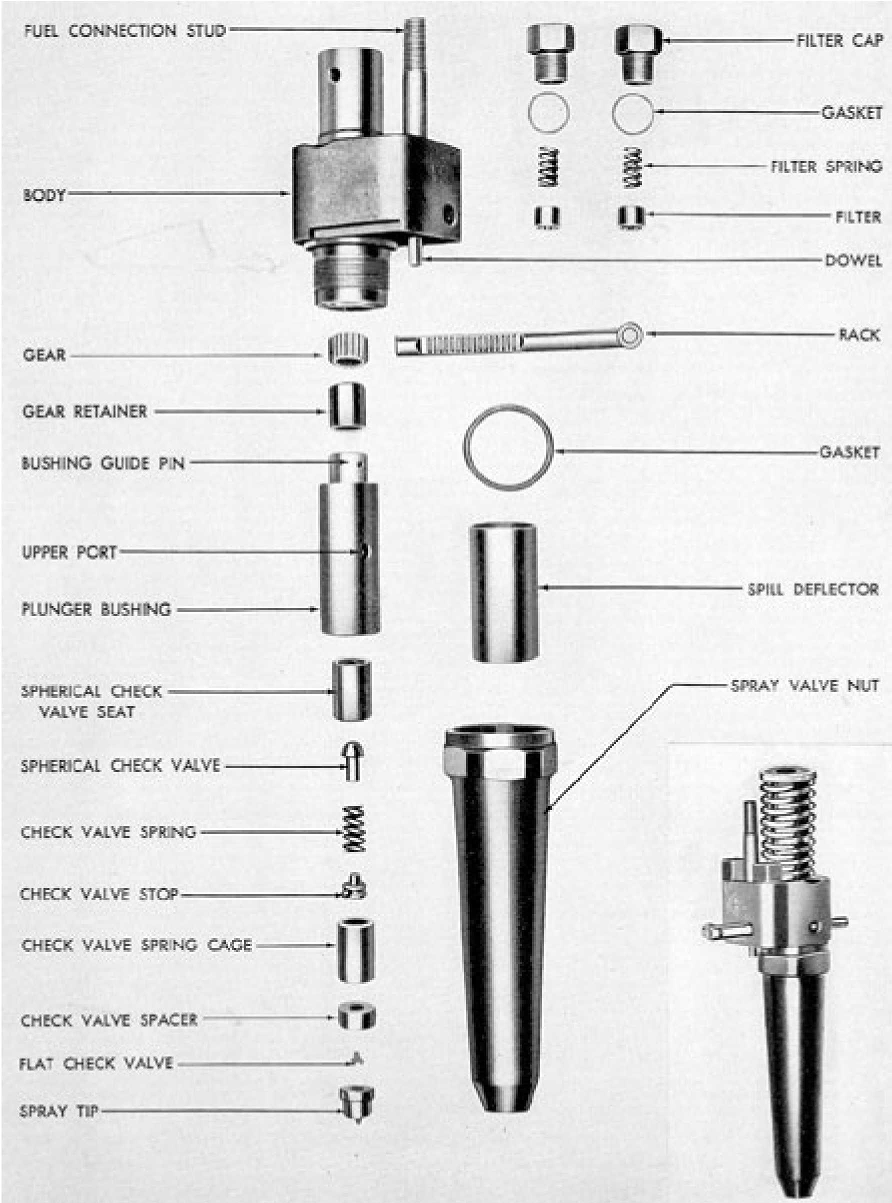


Figure 5-11. Relative arrangement of parts, spherical check valve type unit injector, GM.

inlet and bypass ports in the bushing to control the beginning and ending of the pumping part of the plunger stroke. An oil hole, drilled horizontally from one side of the recess, through the plunger to the other side, connects with a central oil hole extending vertically from the bottom of the plunger.

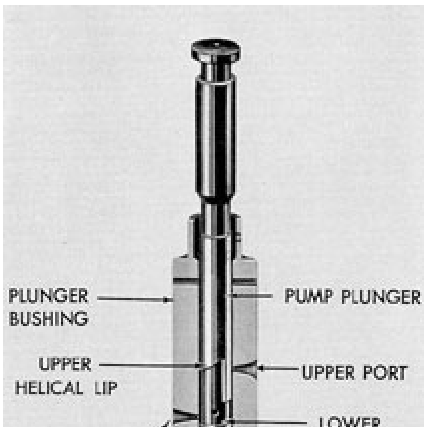


Figure 5-12. Unit Injector plunger and bushing, GM.

The plunger stroke remains constant at about 3/4 of an inch. However, the pump plunger does not pump fuel oil for the entire length of the stroke. The

effective pumping stroke begins when the upper helical lip covers the upper port and ends when the lower helical lip uncovers the lower or bypass port. The upper part of the plunger extends into the hub of the control gear. When this gear is turned, through the rack and linkage, the plunger rotates and changes the angular position of the helical lips with respect to the bushing ports, thereby changing the quantity of fuel injected and the timing of the injection.

In the two types of unit injectors used, one is equipped with a spherical check valve, the other with a needle valve. In the spherical check type valve, fuel, forced down by the pump plunger, passes through a drilled passage in the check valve seat and comes in contact with the spherical check valve. Pressure of the fuel acts on the differential area of the spherical check valve, forcing it off its seat against the check valve spring tension. The oil then goes past the spring and around the check valve stop, through the check valve spacer, around the flat check valve, and out through the openings in the spray tip.

In the needle valve type injector, fuel is forced through drilled passages in the spacer, down through a passage in the needle valve spring cage and needle valve seat, and into an annular groove at the bottom of this seat. It then is forced up through a short inclined passage leading to the needle valve against which the fuel pressure acts. When the fuel pressure is built up high enough to open the valve, the

The needle valve or spherical check valve spring, as well as the spray tip, are made of hardened, chrome-vanadium steel. The spring tension is such that it holds the valve on its seat to insure quick opening and cutoff until the fuel pressure is built up high enough to produce a fine spray when the oil is forced through the spray tip. The upper surface of the spray tip is lapped to affect a seal against this pressure.

b. Operation. Fuel oil enters the unit injector body through a filter and passes around the outside of the plunger bushing. From this supply chamber around the outside of the plunger bushing, the oil goes through the upper and lower ports of the bushing and into the pump chamber.

As the plunger is moved downward by the rocker lever, fuel in the pump chamber is first displaced through both ports into the supply chamber around the bushing, until the lower edge of the plunger closes the lower port. Fuel

fuel passes around the flat check valve and out through the spray tip.

NOTE. On some injectors the spacer has been eliminated and the fuel passes directly from the pump plunger into the needle valve spring cage.

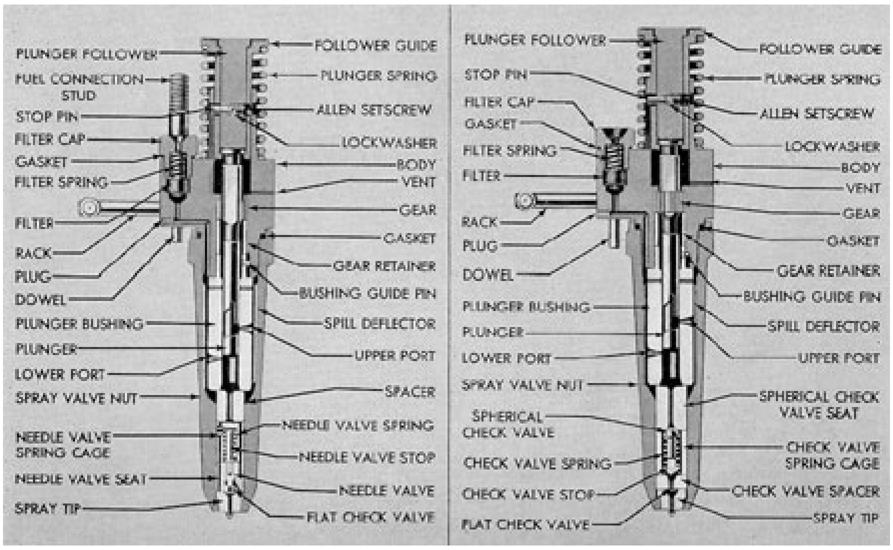


Figure 5-13. Cross sections of needle valve and spherical check valve type unit injectors, GM.

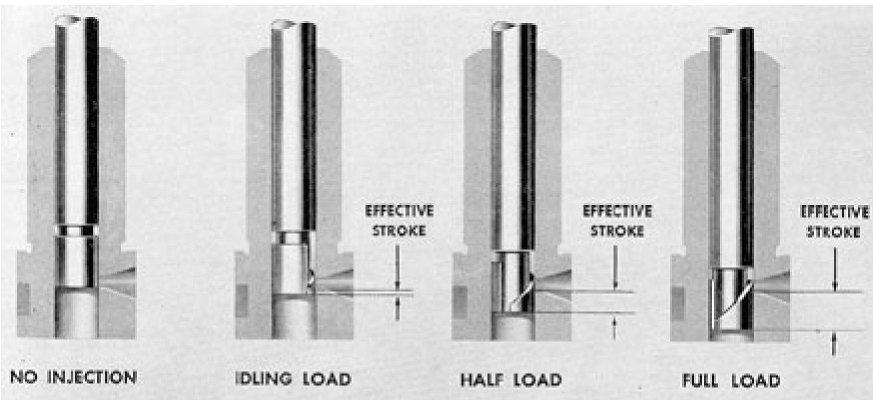


Figure 5-14. Plunger position at no Injection, idling, half load, and full load.

in the pump chamber is then displaced through connecting central and transverse holes in the lower port of the plunger and through the upper port into the supply chamber. Further downward movement of the plunger locates the helical plunger lips with respect to the port openings in the plunger bushing and thereby controls the amount of fuel injected into the cylinder. Figure 5-14 illustrates the plunger position and effective stroke for

plunger causes the upper lip to cover the upper port at which point the effective pumping stroke begins and the fuel in the pump chamber is then forced down through the spray valve. Injection continues until the lower lip on the plunger uncovers the lower port in the bushing at which point the effective pumping stroke ends. The fuel then bypasses upward through the holes in the plunger and through the lower port into the supply chamber. This immediately lowers the pressure of the fuel remaining in the pump chamber so that the valve snaps shut to prevent dribble. On the return stroke, the upward movement of the plunger uncovers the ports and allows fuel to enter the chamber.

The cylinder load, that is, the amount of fuel sprayed into the cylinder, is controlled by the rotation of the pump plunger. Rotating the no injection, idling load, half load, and full load.

In addition to measuring the amount of fuel, the injector pump plunger varies the timing of injection. This is accomplished by means of the upper helix on the pump plunger. The angularity of this helix causes injection to be advanced for a longer effective stroke of the plunger (more fuel) and retarded for a shorter effective stroke of the plunger (less fuel).

In the pressure chamber, fuel oil under pressure works on the differential area of the needle valve. The pump plunger creates a hydraulic pressure on the fuel oil in the pressure chamber that is greater than the pressure of the needle valve spring. This pressure working on the differential area of the needle valve overcomes the spring tension and raises the needle valve, opening the passage to the spray tip.

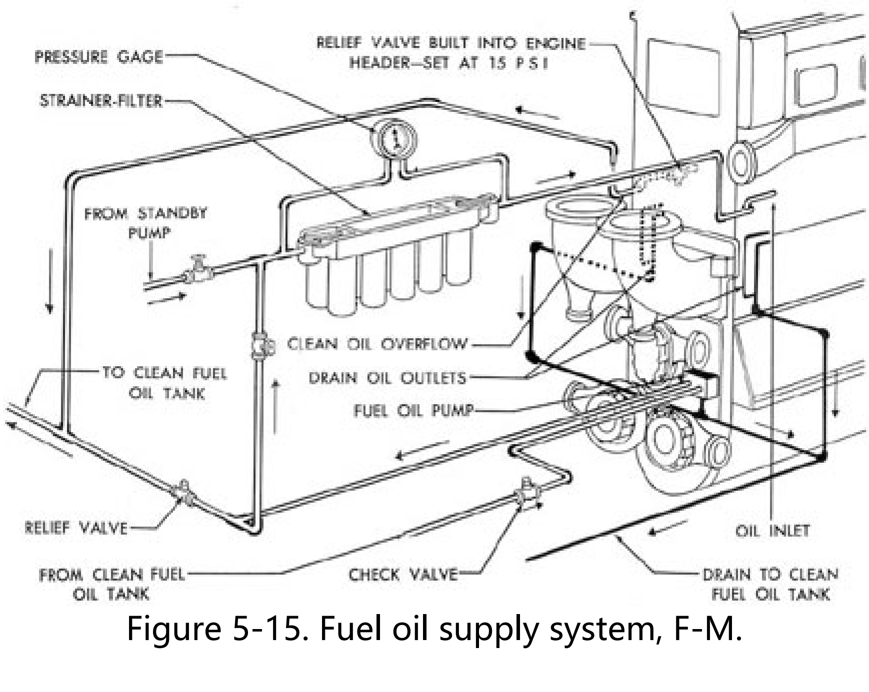


Figure 5-15. Fuel oil supply system, F-M.

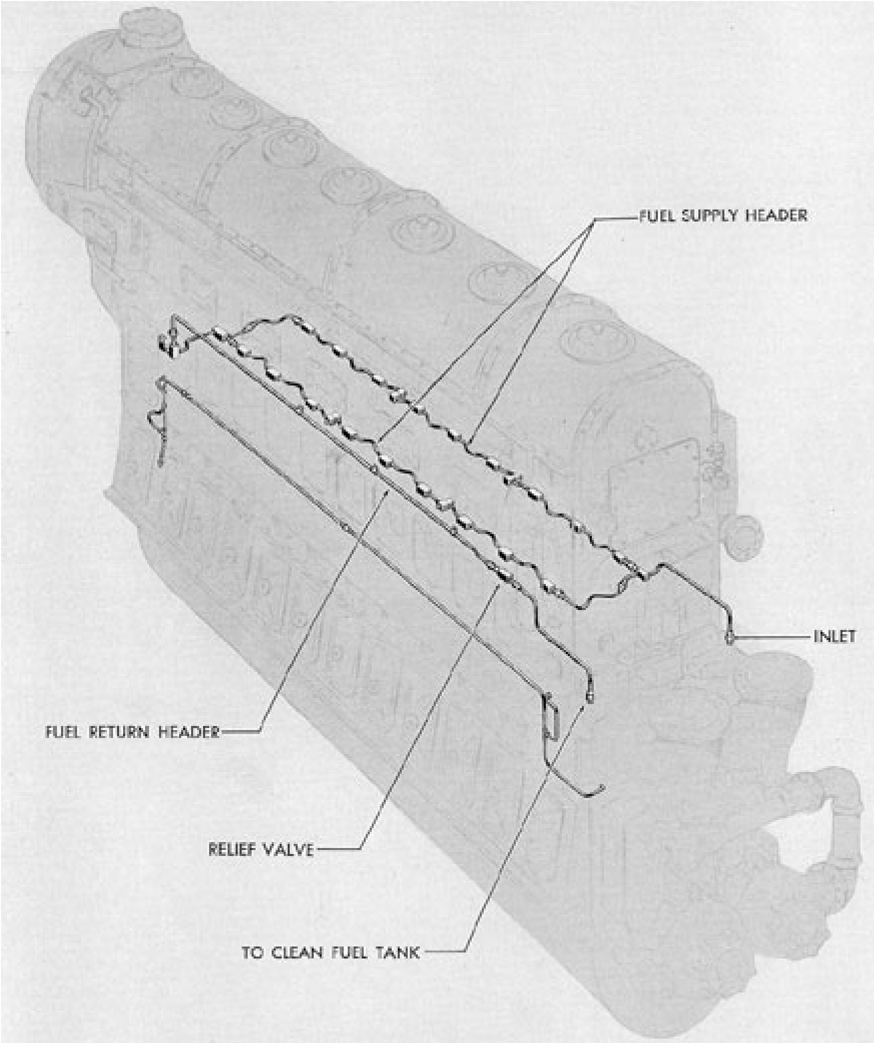


Figure 5-16. Isometric view of fuel injection system, F-M.

Fuel oil enters the spray tip and the force of the hydraulic pressure sprays the fuel oil through the orifices of the spray tip at a 75-degree angle with the centerline of the cylinder. Spray tips normally are marked to indicate number and diameter of orifices and angle of spray. For example: A spray tip marked 6-006-155 has 6 orifices, each measuring .006 inch, directing the spray at a 155-degree included angle.

F. FAIRBANKS-MORSE ENGINE FUEL OIL SYSTEM

5F1. General. The attached fuel oil pump draws fuel by suction from the clean fuel oil tank and delivers it through the strainer and filter units to the engine main fuel oil header. The pump has a greater capacity than is required to furnish fuel oil to the engine at maximum speed, it to the injection nozzles at the proper time. Each of the fuel injection pumps consists primarily of a tappet assembly, pump barrel, plunger return spring, discharge valve with its seat and spring, and the control rack.

A tappet assembly attached to the top of the pump body, transforms

therefore a pressure is built up in the supply line to the engine. A relief valve in the engine supply header prevents this pressure from being built up above a certain desired pressure, usually 15 psi. Pressure in excess of this amount is relieved by the relief valve which returns the excess oil to the clean fuel oil tank by gravity.

Fuel oil delivered to the engine inlet is piped along both sides of the engine through the supply header which in turn is connected to the fuel inlet ports of each injection pump.

Two injection pumps serve each cylinder, one from the left side, the other from the right. The pumps are actuated in proper sequence by the cams on the camshafts. Each injection pump delivers fuel oil to one of the injection nozzles, which, like the pumps, are arranged two to a cylinder, one on each side. The amount of fuel the pumps deliver to the nozzles is regulated by movement of the injection pump control racks which are actuated, through plungers and guides, by an injection pump control rod on each side of the engine.

A drip pan under each injection pump collects any fuel oil that drains from the top of the pump body. This oil is sent to the clean fuel oil tank through tubes extending from the drip pan at each end of the engine. Oil collecting on the bottom of the injection nozzle compartments is also drained into the clean fuel oil tank.

the rotary motion of the camshaft into up-and-down motion of the pump plunger. The assembly is comprised of a cam roller, a push rod, and a push rod spring. The push rod spring holds the push rod and cam roller against the camshaft cam. As the camshaft rotates, the cam acts against the cam roller to force the push rod down against the spring tension to actuate the injection pump plunger.

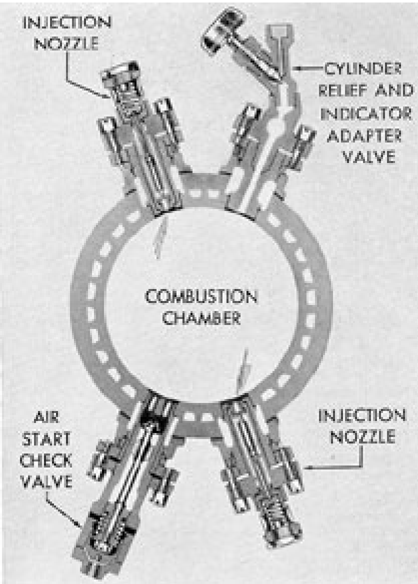


Figure 5-17. Arrangement of injection nozzles in F-M cylinder.

5F2. Fuel injection pump. a. Description. The injection pumps receive fuel oil at low-pressure, measure it into correct amounts for injection, build up a high pressure, and deliver

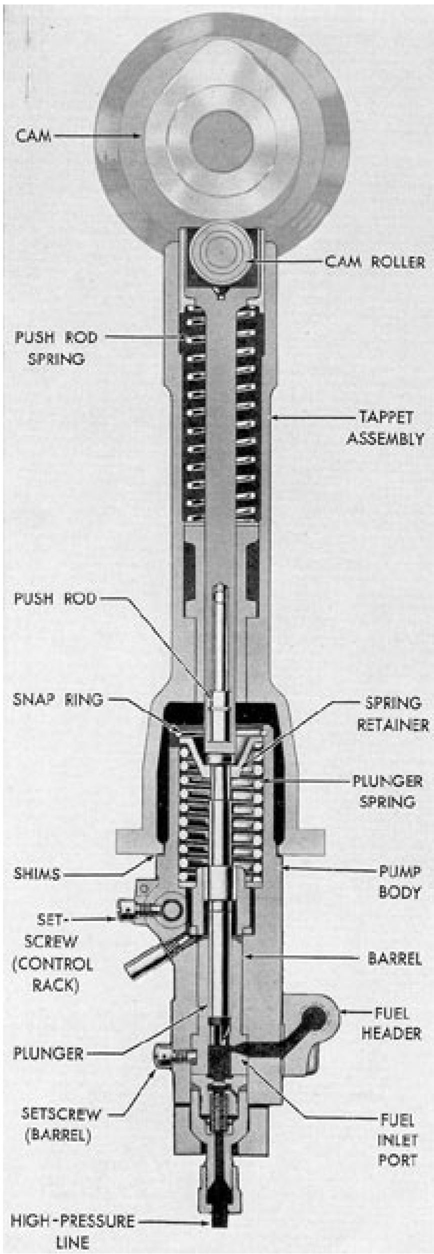


Figure 5-18. Cross section of fuel injection pump, F-M.

The injector pump plunger moves vertically in the pump barrel, delivering fuel to the injection nozzle by way of a discharge valve and the injection tube connecting the pump and the nozzle. At the base of the plunger is an annular recess. The lip formed between the annular recess and the bottom of the plunger has a slanting, or helical, edge. A vertical slot extends from the annular recess to the bottom of the plunger. It should be noted that, except at the slot, the edge of the helical lip at the bottom of the plunger is constant or even. Hence it is referred to as the constant beginning helical lip. The edge of the helical lip toward the recess in the plunger is helical, or slanting, and is referred to as the variable ending helical lip.

The plunger is lapped to an extremely close fit in the bore of the pump barrel. These two parts are always provided in pairs and should not be separated.

The pump barrel is positioned in the pump body by a setscrew which extends into an elongated slot at the lower part of the barrel. Fuel is delivered to the pump chamber of the barrel through a single inlet port which is also the bypass port.

The plunger spring returns the plunger to the starting position when the high point on the cam passes. The spring is held in position at the upper part of the pump body by a snap ring.

The pump discharge valve is held in its seat by a pump discharge valve spring. The spring returns the valve to its seat at the end of an effective pumping stroke.

The amount of fuel delivered by the injection pump is controlled by rotating the pump plunger. The mechanism by which this is accomplished is known as the fuel injection pump control rack and the control gear. The control gear is splined to the pump plunger and meshes with teeth in the control rack. Any lateral movement of the rack is transmitted to the control gear, causing the pump plunger to rotate. The control racks have calibration scales for reference in checking operating conditions under various engine loads. Normally the control racks are set at the 0 marking, as indicated by a pointer, when in the no fuel position. Each of the control racks is adjustable to the correct calibration by means of the control rack adjusting screw.

The control racks are connected to the control rod at the control rack collar. The control

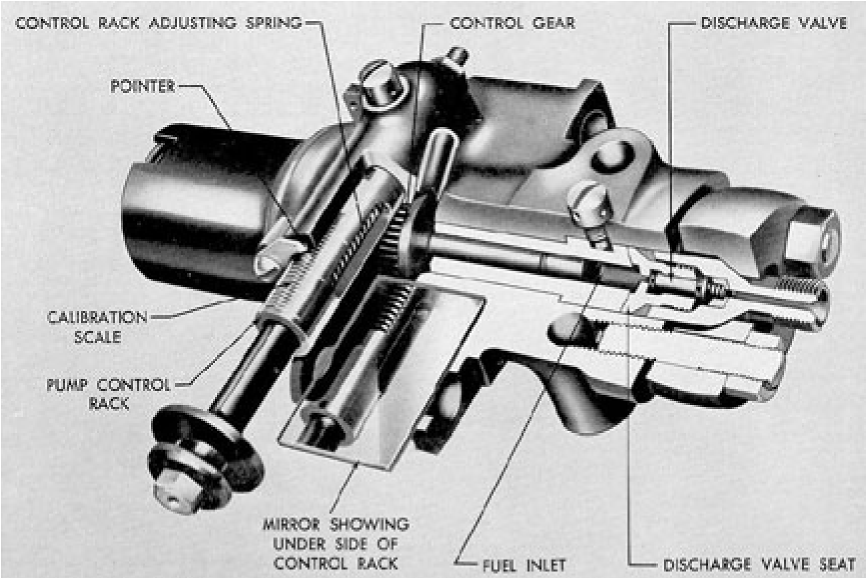


Figure 5-19. Cutaway of fuel injection pump, F-M.

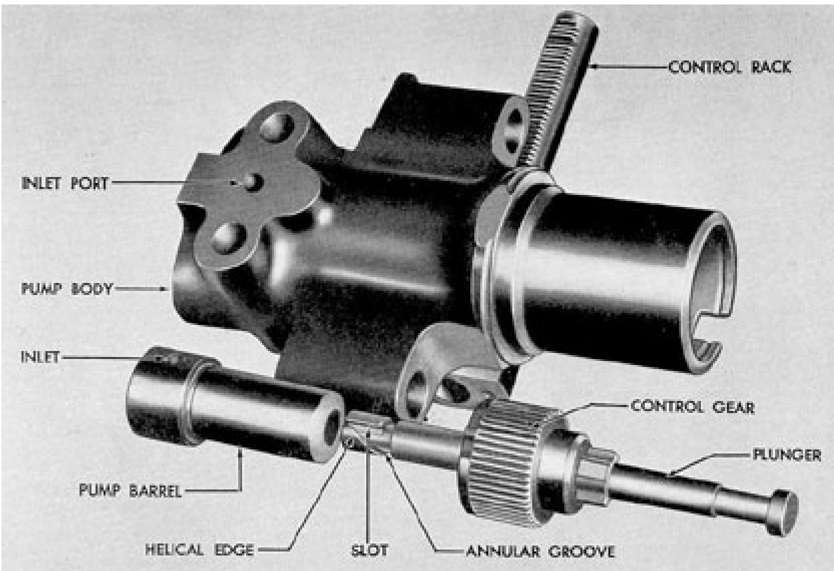


Figure 5-20. Fuel injection pump parts, F-M.

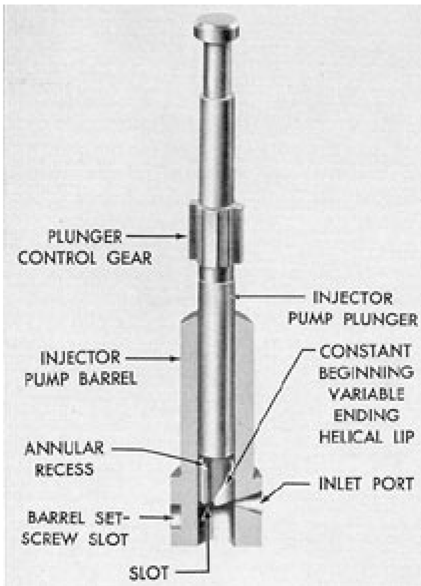


Figure 5-21. Details of injection pump plunger and barrel, F-M.

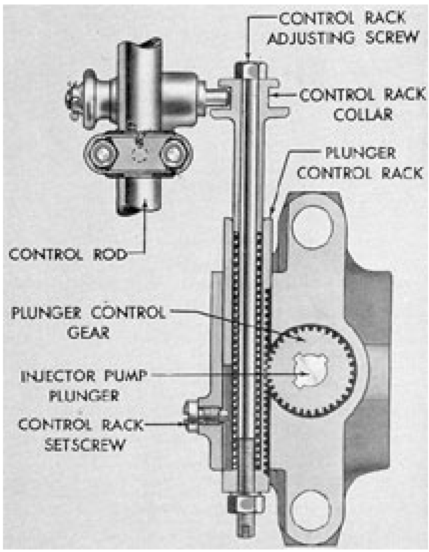


Figure 5-22. Cross section through control rack, F-M.

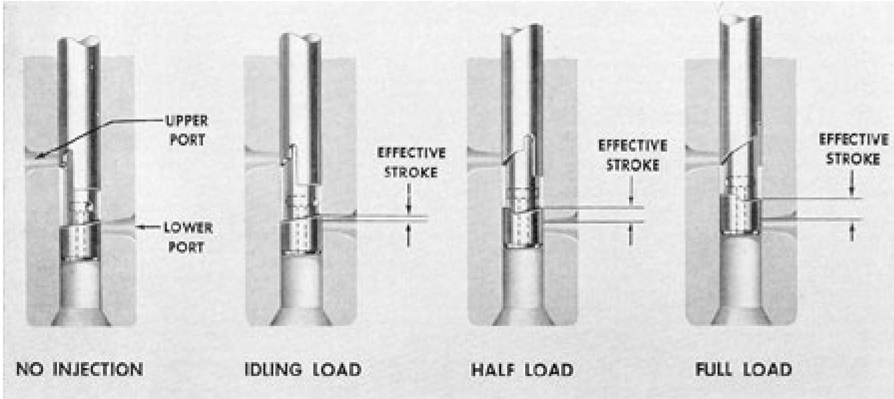


Figure 5-23. Position of GM fuel injection pump plunger at no injection, idling half load, and full load.

rod is connected to the engine governor by a linkage. Thus the rotary position of the pump plungers, and hence, the amount of fuel delivered, can be directly controlled by the governor.

It should be noted that there is no means provided for advancing or retarding the time of injection in the cylinder for a change in fuel supply as there is in the unit injector pump of the GM engine.

b. Operation. Figure 5-23 shows how the amount of fuel delivered through the discharge valve is varied by rotating the pump plunger. The plunger stroke remains constant at about 5/8 of an inch The effective pumping stroke begins when the constant beginning edge of the helical lip covers the fuel port. The pumping stroke ends when the variable ending edge of the helical lip uncovers the port, allowing the fuel oil remaining in the chamber to flow through the bypass area in the pump plunger.

The first picture in Figure 5-23 shows the pump plunger in a position in which the vertical slot the discharge valve seat, through the discharge valve cage, and into the high-pressure line to the injection nozzle.

Fuel enters the annular groove in the injection nozzle and is directed down through longitudinal grooves comprising the edge type filter. The clearance between the grooves of the filter and the injector nozzle body is approximately 0.0015 inch. The fuel oil is forced from the filter, down through flutes on the outside of the needle sleeve, then through needle sleeve holes at the bottom of the flutes to enter a fuel chamber in the needle sleeve. In this chamber, the hydraulic pressure of the fuel, acting on the differential area of the valve, lifts the valve from the needle valve seat. The oil is then discharged into the combustion chamber through the nozzle tip. As soon as the pressure from the fuel injection pump diminishes, the spring in the nozzle forces the needle valve closed.

The three orifices in the nozzle tip are 0.0225 of an inch in diameter and are positioned to direct the spray at a 15-degree angle for

is aligned with the fuel port. This is the no fuel position. Any downward movement of the pump plunger in this position allows the fuel to pass from the chamber beneath the pump plunger, through the slot and out through the inlet port. Thus no fuel is delivered through the discharge valve.

When the pump plunger is rotated slightly, a relatively shallow depth of the helical lip is aligned with the fuel inlet port. The effective pumping stroke is short, and only a small amount of fuel is delivered through the discharge valve such as is required for an idling engine.

The third and fourth pictures in Figure 5-23 shows the rotary position of the pump plunger at half load and full load fuel delivery.

Fuel oil is supplied to the fuel injection pump through a line from the engine main fuel oil header. The fuel oil enters the fuel port in the pump bushing at approximately 15 pounds' pressure and fills the chamber below the pump plunger. At the proper time the pump plunger is actuated by the cam on the engine camshaft through the tappet mechanism on the pump. The force exerted on the fuel oil in the chamber beneath the pump plunger overcomes the spring tension on the discharge valve and opens the valve. The oil then passes through holes below thorough distribution in the combustion chamber.

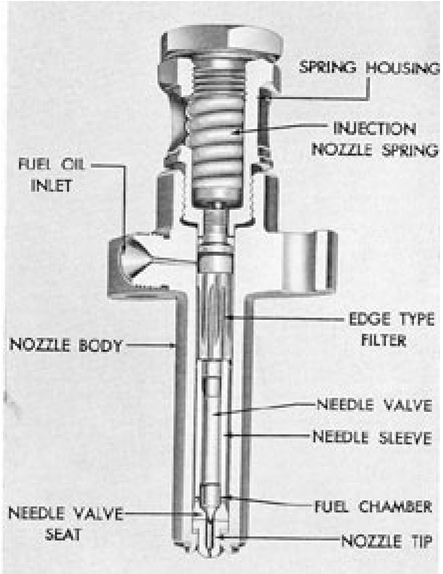


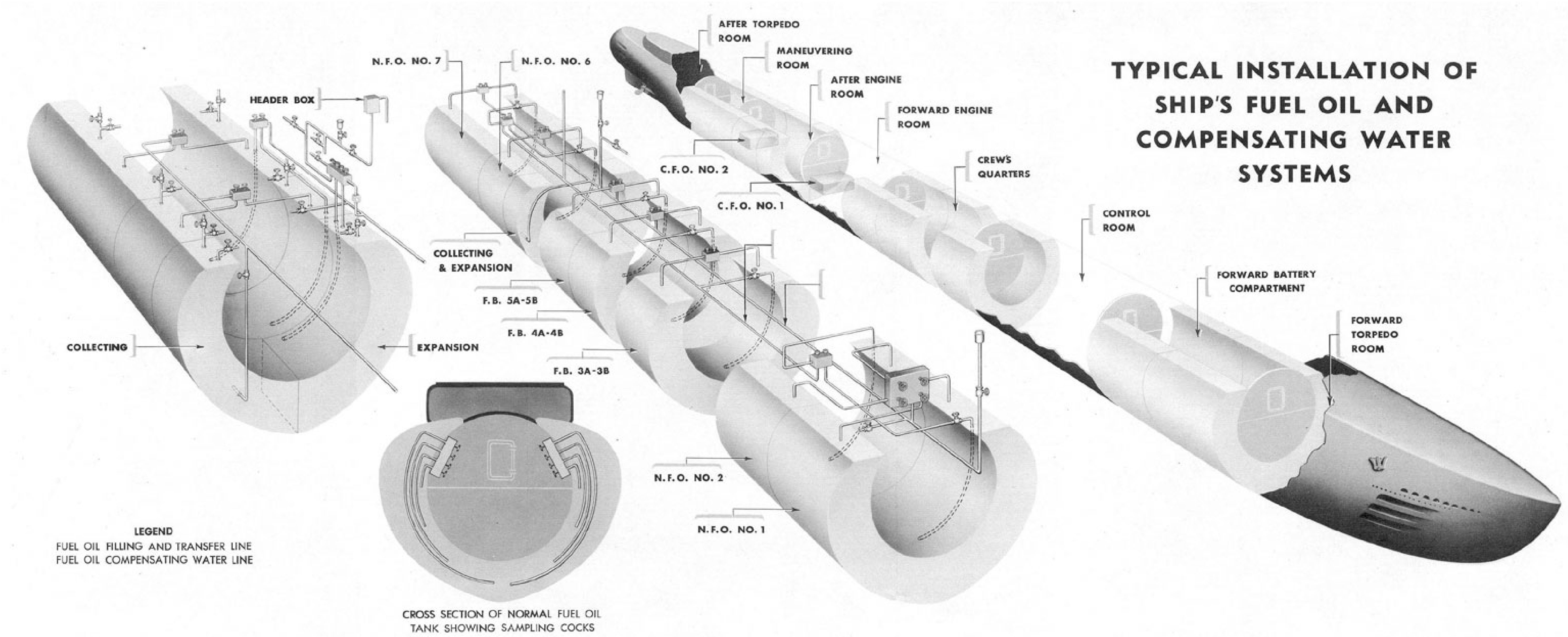
Figure 5-24. Cutaway of injection nozzle. F-M.

Previous Chapter
Sub Diesel Home Page
Next chapter

Version 1.10, 22 Oct 04

Figure 5-1. TYPICAL INSTALLATION OF SHIP'S FUEL OIL AND COMPENSATING WATER SYSTEMS.

Sub Diesel Home Page



6
INTAKE AND EXHAUST SYSTEMS

A. GENERAL

6A1. Intake systems. All of our modern submarine diesel engines are of the 2-stroke cycle type. The purpose of the intake systems in these engines is to force out the exhaust gases of combustion as effectively as possible and to recharge the cylinder with fresh air in order to support combustion for the next succeeding cycle. The supply of air must be in excess of that required to just support combustion since the fuel is thoroughly mixed with only part of the air compressed within the cylinder. The ratio of air to fuel in most diesel engines is approximately 20 to 1 at full load.

6A2. Scavenging. The term scavenging is used to describe the process of ridding the cylinder of burned exhaust gases during the latter part of the expansion stroke and the early part of the compression stroke of the 2-stroke cycle engine. Scavenging is accomplished by admitting fresh air under a pressure of about 1 to 5 psi into the cylinder while the exhaust valves or ports are open. This pressure usually is developed by means of a scavenging air blower. These blowers are driven from the engines themselves and generally are of the lobed rotor

These methods are illustrated in Figures 6-1 to 6-4. In port direct scavenging, the exhaust ports are on one side of the cylinder and the scavenging ports on the other. In port loop scavenging, the exhaust and scavenging ports are on the same side of the cylinder. In uniflow port scavenging, the air enters at ports at the lower end of the cylinder and passes out through ports in the upper end of the cylinder.

In valve uniflow scavenging, air enters the cylinder through ports in the bottom and passes out through exhaust valves in the cylinder head, carrying the burned exhaust gases with it.

The ports used for the inlet of scavenging air are usually constructed so as to give the air a whirling motion or turbulence to clear out all possible exhaust gases and fill the entire cylinder with a charge of fresh air.

In scavenging air systems, it is possible to supercharge the cylinder during the air intake. This is done by closing the exhaust ports or valves slightly ahead of the inlet port closure. This allows the air pressure in the cylinder to build up to scavenging air pressure, increasing the amount of air, the air-fuel ratio, and the

type, the rotors revolving together in closely fitting housings. The process of scavenging must be carried out in an extremely short period of time, depending upon the speed of the engine. The burned gases must be blown out of the cylinder and a fresh charge of air admitted during the time that the ports or valves are open. For example, in an engine making 750 rpm with the exhaust ports open for 140 degrees of crank angle, the elapsed time the ports are open each revolution is only (140/360) x (60/750) or approximately 1/32 of a second.

The scavenging air must be so directed as to remove the burned gases from the remote parts of the cylinder. The methods used may be classified as follows: port scavenging (direct, loop, and uniflow), and valve scavenging (uniflow).

combustion efficiency. If the amount of fuel injected is increased to give the same air-fuel ratio as before supercharging, the effect of supercharging is to give more power output to the cylinder. In the present submarine type engines, the F-M engine is supercharged, but the GM engine is not.

6A3. Intake system components. The intake systems consist of the following parts:

a. Air intake silencers and strainers. Intake air for submarine engines is drawn from the engine room compartments by the scavenging air blower through air silencers and strainers. If some type of air silencer were not used, the noise of the intake air would be almost unbearable because of its high-pitched whistling sound. Strainers are installed to remove any dirt or other foreign matter that would otherwise enter the scavenging blower or engine and cause damage.

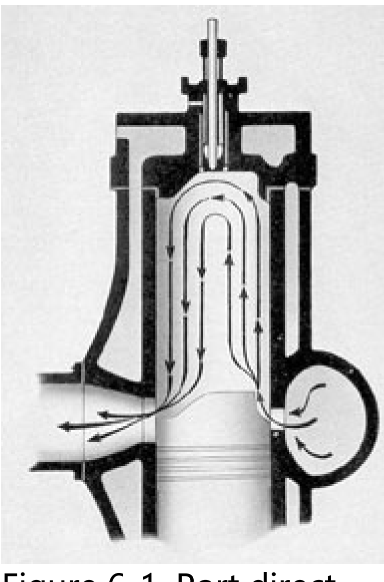


Figure 6-1. Port direct scavenging.

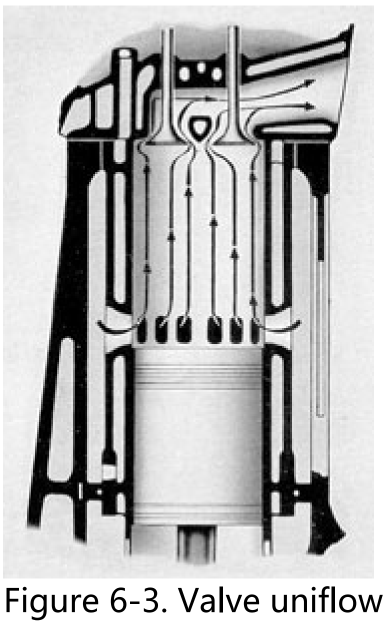


Figure 6-3. Valve uniflow scavenging.

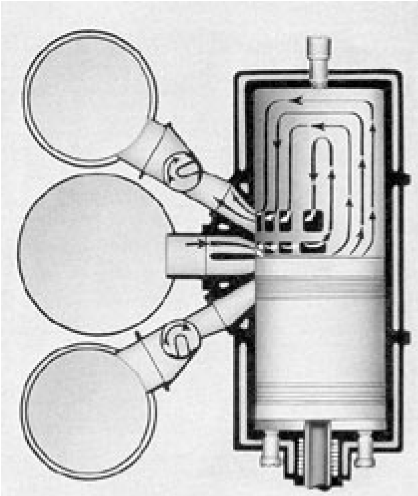


Figure 6-2. Port loop scavenging.

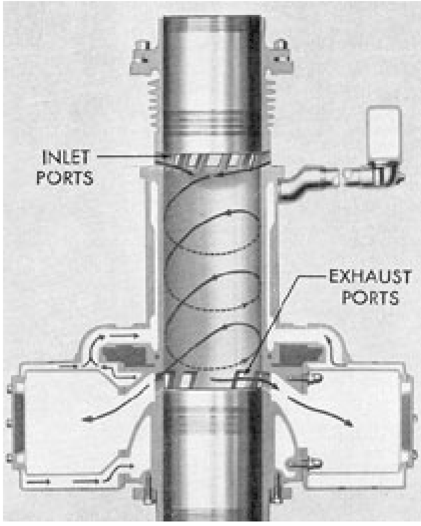


Figure 6-4. Cross section of F-M cylinder with uniflow port scavenging.

b. Scavenging air blower. The scavenging air blower furnishes air under pressure to the intake headers and receivers and eventually to the cylinder inlet ports.

c. Air intake headers, receivers, and necessary piping. The air headers and receivers carry the air from the scavenging air blowers to the inlet ports of the cylinders. In most installations, scavenging air headers and receivers are built into the cylinder block. Drains are placed in the scavenging air headers to drain off any liquids that may have accumulated. Spring-loaded covers are also furnished in the scavenging air header to allow the venting of excess pressure in case of emergency.

d. Intake air ports. The intake air ports are in the cylinder liner and permit the scavenging air to pass from the scavenging air receivers into the cylinder when the ports are open. The ports are usually tangentially constructed so as to the reciprocating motion of the pistons.

The exhaust headers or belts conduct the exhaust gases from the exhaust valves or ports to the atmosphere through an inboard and an outboard exhaust valve and muffler. The exhaust manifold and exhaust elbows (if used) are usually water jacketed to permit cooling of the piping and manifolds. The cooling water normally comes from the engine fresh water system. Cooling of these parts keeps down the temperature of the metal, thus prolonging its life and reducing its expansion to a minimum. In most exhaust systems, drains are provided to allow drainage of any accumulated liquids from the exhaust belts.

In submarine installations, the gases of combustion are piped from the exhaust headers to the outside of the submarine through an in board and outboard main engine exhaust valve and muffler. The inboard exhaust valve is inside

give the air a whirling motion as it enters the cylinders. They are usually opened and closed by the reciprocating motion of the piston.

6A4. Exhaust systems. The purpose of the exhaust system is to convey the burned exhaust gases of combustion from the cylinders to the atmosphere as silently as possible. The system includes exhaust valves and ports, headers and pipes, main inboard and outboard exhaust valves, and engine mufflers.

The exhaust valves or ports, as the case may be, are properly timed so as to permit the gases of combustion to escape from the cylinder at the correct point of the cycle. In the GM engine, this is accomplished by means of exhaust valves; in the F-M engine, by means of exhaust ports. Due to the heat that must pass through these exhaust valves or ports, they must be made of special material or be thoroughly cooled to prevent distortion and pitting. Valves are usually made of a high silicon heat-resistant alloy steel. In some large engine installations, the exhaust valves may be water or sodium cooled. A thermocouple is usually placed at the exhaust elbow to measure the exhaust temperature of each cylinder. When exhaust valves are used, they are opened and closed by means of rocker arm and camshaft assemblies. The exhaust ports, if used, are opened and closed by the pressure hull of the submarine and is hand operated. The outboard exhaust valve is located outside the pressure hull and is operated either by hand or by hydraulic power, the controls for the valve being at the throttleman's station at the engine. Both inboard and outboard exhaust valves are water cooled, the former usually by water from the engine fresh water system, the latter by water from the engine salt water system.

Mufflers are placed in the exhaust system. This is necessary, because in a 2-stroke cycle engine the uncovering of the exhaust ports releases a pressure of 20 to 40 psi in the exhaust system and this produces a noise that can be heard for miles if not muffled by some form of silencer. These mufflers are usually of cast or sheet iron construction with a system of baffles that break up the noise without producing back pressure. There are two general types of mufflers in use, the wet type and the dry type. In both types, circulating water is used to reduce the temperature of the exhaust gases as much as possible. The difference between the two is that in the dry type the exhaust gases do not come in contact with the cooling water, whereas in the wet type the gases are expanded in the muffler in the presence of a water spray. The exhaust gases in passing through the water spray are cooled, condensed, decreased in volume, and

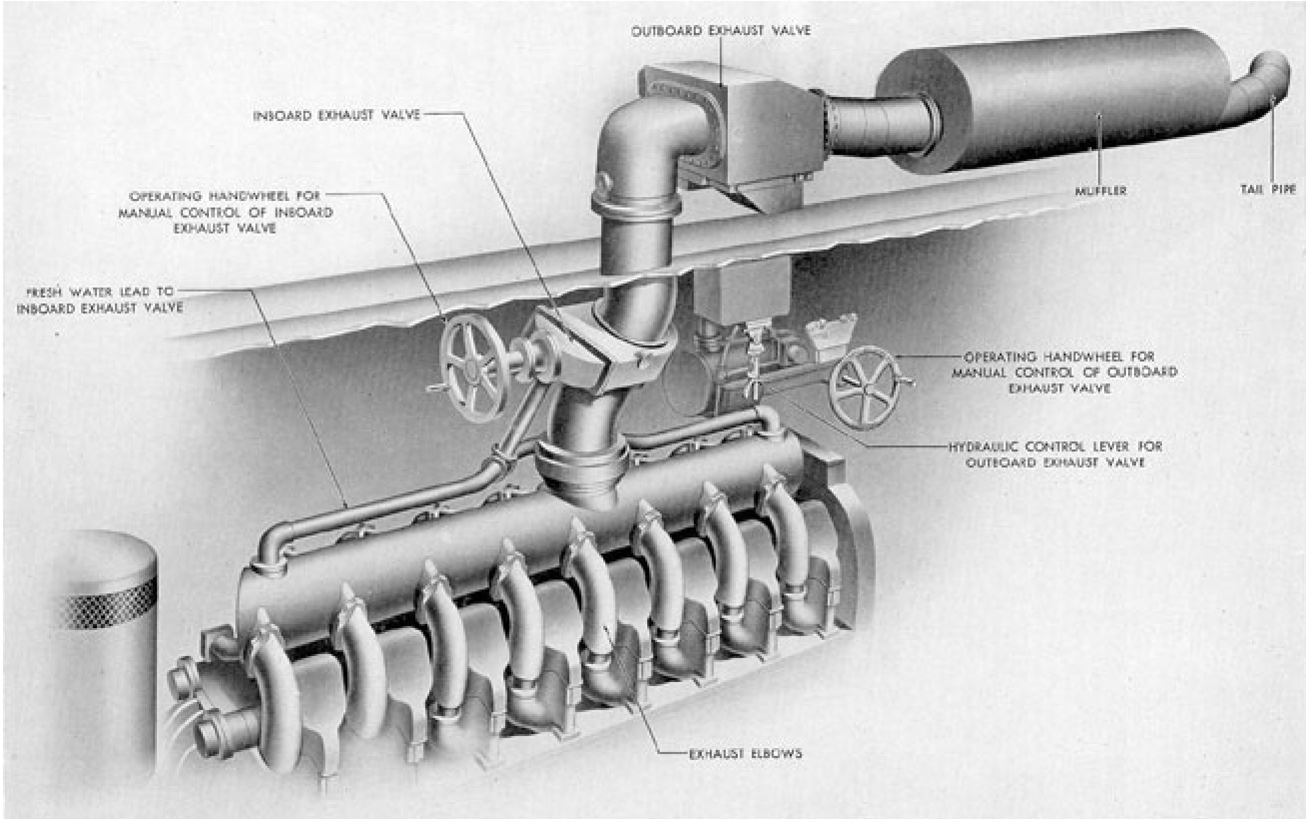


Figure 6-5. Typical exhaust system piping.

effectively silenced. Under normal operation, the smoke is also eliminated. Submarine installations use the wet type of muffler. From the muffler, the exhaust gases are passed out into the atmosphere through a section of piping known as the tail pipe.

B. GENERAL MOTORS INTAKE AND EXHAUST SYSTEM

6B1. General description. The General Motors engine employs the uniflow valve method of scavenging. The blower, mounted at the forward end of the engine crankcase and driven by the engine, takes air from the atmosphere through an attached silencer and forces it under pressure into the air box. The air box consists of the frame space in the engine included between the two legs of the V-construction and the open space between the upper and lower deckplates of each bank. The air from the air box goes through the cylinder inlet ports whenever The exhaust gases are released from the cylinder when the exhaust valves are opened by action of the camshaft and rocker arm assembly. The exhaust valves are opened ahead of the inlet ports to allow the pressure of the exhaust gases to be partially released before the low-pressure scavenging air is admitted to the cylinder. The exhaust gases pass through the exhaust valves into the water-cooled cylinder head and thence into the exhaust elbow connecting each cylinder head with the main exhaust manifold. This manifold extends longitudinally along the top

the individual pistons uncover the ports at the end of the expansion or power stroke. This scavenging air forces out the exhaust gases and charges the cylinder with fresh air.

centerline of the engine with elbow connections into each cylinder head. Thermocouples for measuring the temperature of the exhaust gases for each cylinder are located in each exhaust elbow. Both exhaust elbows and exhaust

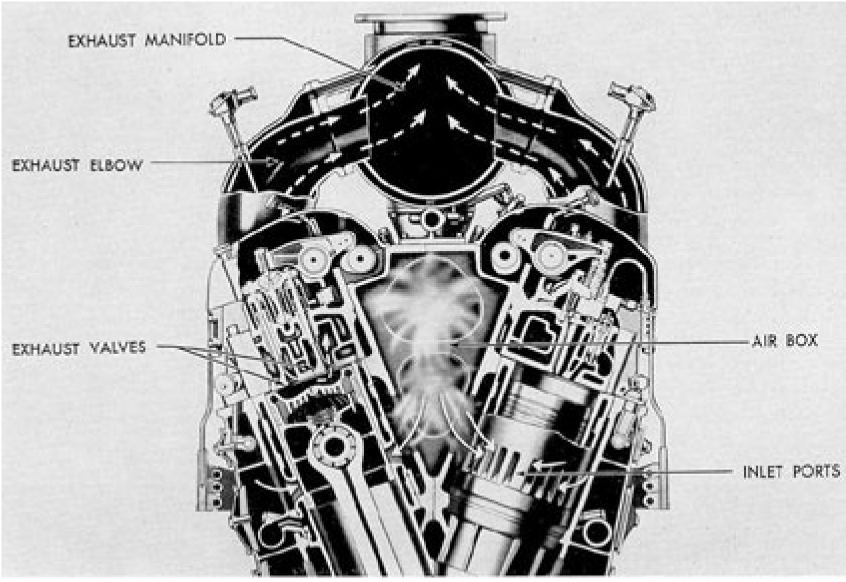


Figure 6-6. GM cylinder intake and exhaust.

manifold are water jacketed for cooling purposes. From the main exhaust manifold, the gases pass into a vertical pipe which leads to the inboard exhaust valve. From this valve, the gases pass outside the pressure hull, through exhaust piping which leads to the hydraulically operated main engine outboard exhaust valve, and thence to the atmosphere by way of the muffler and tail pipe. The inboard exhaust valve is cooled by water from the engine fresh water system, while the outboard valve is cooled by the engine salt water system.

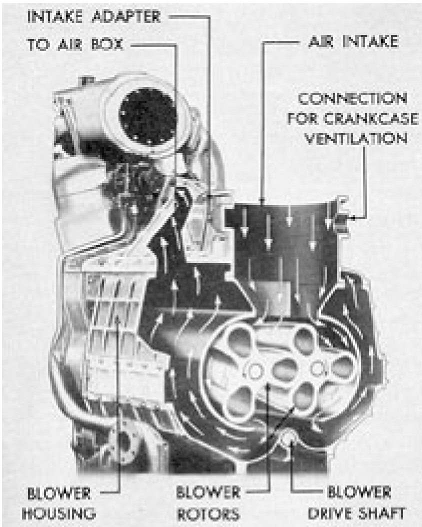


Figure 6-7. Cutaway of blower assembly, GM.

Drains are provided in the piping between the inboard and outboard exhaust valves so that any salt water that may have leaked past the outboard exhaust valve can be drained

into the engine room bilges. On a submarine it is extremely important that this space be drained before starting an engine after surfacing from submerged operations, otherwise the engine may be flooded.

6B2. Scavenging air blower. The scavenging air blower is of the positive displacement type consisting of a pair of rotors revolving together in a closely fitted housing. Each rotor has three helical lobes which produce a continuous and relatively uniform displacement of air. The rotors do not touch each other or the surrounding housing. Air enters the housing at the top and fills the spaces between the rotor lobes as they roll apart. The air is carried around the cylindrical sides of the housing, in the closed spaces between the lobes and the housing. It is forced under pressure to the bottom of the housing as the lobes roll together. Then the air passes through the space between the inner and outer wall of the blower housing and into the air box around the cylinder liners.

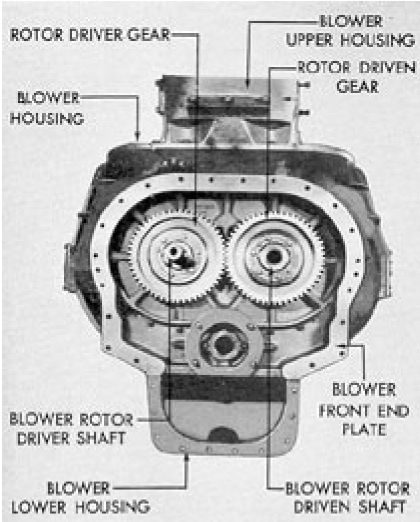


Figure 6-8. Front view of blower, GM.

Each rotor is carried on a tubular serrated shaft. Endwise movement is prevented by two taper pins. No gaskets are used between the end plates and the housing due to the importance of maintaining the correct rotor end clearance.

A fine silk thread around the housing and inside the stud line, together with a thin coat of nonhardening gasket compound, provides an air tight seal.

Babbitted bearings in the end plates locate the rotors in the two half-bores of the housing. These bearings permit clearances to be held to

a minimum between the rotor tips and the housing bores. Both ends of the rotor bearings have thrust surfaces at the gear end of the blower. The thrust surfaces locate the rotors endwise and prevent contact between the rotors and the end plates.

Figure 6-9. Air silencer.

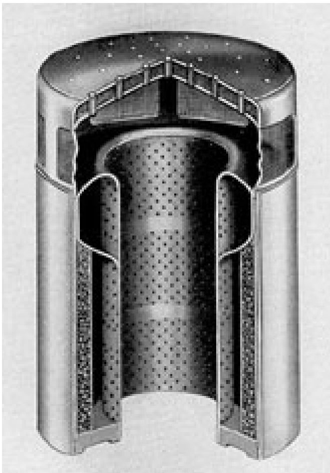


Figure 6-10. Cutaway of typical air silencer.

The blower is driven from the crankshaft through a quill shaft and through a train of helical spur gears. The quill shaft is driven through a serrated quill shaft coupling on the crankshaft, and drives the main driving gear in the train through a serrated connection in the gear hub. The main drive gear transmits power directly to the blower rotor driver gear. The quill shaft coupling is fastened to the end of the crankshaft and is driven through large dowel pins. The rotor driver and driven gear are closely fitted and rigidly attached to both rotor shafts to prevent the rotors from touching as they revolve. Each gear hub is pressed on the serrated rotor shaft. A hexagon head lockscrew in the rotor shaft holds a thrust collar as a spacer between the gear hub and the end of the rotor. The collar maintains clearance between rotors and blower end plate.

The blower rotor gears are bolted to the gear hub flanges and are located angularly by

dowel pins. Due to the importance of having the rotors roll together without touching, yet with the least possible clearance, it is necessary to locate the dowel pins during assembly for a given set of gears and hubs.

Oil passages in the end plates conduct lubricating oil under pressure to the blower bearings. Oil seals are provided at each bearing to prevent oil from entering the rotor housing.

6B3. Intake silencer. The air is drawn into the blower through an intake silencer mounted on the blower intake adapter. The silencer is a double sheet metal case with screened openings at the top. Felt padding is cemented between the double layers of metal at the top and sides of the case. To minimize the noise caused by the entering air, a perforated metal tube is welded through the center of the case, and the upper space between the outer shell and intake tube is filled with sound-deadening material.

6B4. Air maze. A breather system is used to prevent contamination of the engine room atmosphere by heated or fume-laden air which

would otherwise escape from the crankcase. This ventilation of the crankcase also reduces the formation of sludge in the oil and prevents any accumulation of combustible gases in the crankcase and oil pan. removes the oil from the vapor being drawn into the blower.

The air maze element consists of a number of fine steel and copper wire screens that remove the oil from the oil-laden air as it is drawn through the air maze screens. The

Atmospheric air for the breather system enters the engine through the cylinder head cover breathers. The blower suction draws air from the crankcase through the air maze which oil deposited on the wire drips to the bottom of the air maze housing, and then drains back to the sump tank through a tube.

C. FAIRBANKS-MORSE INTAKE AND EXHAUST SYSTEM

6C1. General description. The inlet or scavenging air system supplies the fresh air that blows the exhaust gases out of each cylinder at the end of the power stroke and recharges and supercharges the cylinder for the next compression stroke. The air is drawn from the engine room into the scavenging blower through an air intake silencer. From the scavenging air blower, the air is forced into two exhaust belts and receivers, one extending along each side of the engine. These receivers conduct the air up to the cylinder block compartments which surround the cylinder liners at their inlet ports. These ports direct the scavenging air tangentially into the cylinder when the upper piston uncovers the scavenging air ports. This air clears out the exhaust gases of combustion and fills the cylinder with a charge of fresh air. As the lower crank leads the upper crank by 12 degrees, the exhaust ports are uncovered by the lower piston before the inlet ports are uncovered by the upper piston. The delay allows most of the pressure of the cylinder to escape through the exhaust ports before the relatively low pressure of the scavenging air is admitted. The lower crank lead also causes the lower piston to cover the exhaust ports before piping which leads the exhaust gases up to the inboard exhaust valve. The exhaust belts and exhaust nozzles are cooled by fresh water from the engine fresh water system. From the inboard exhaust valve, the gases pass outside the pressure hull through the outboard exhaust valve, muffler, and tail pipe to the atmosphere. As in the GM installation, a drain is placed in the exhaust piping between the outboard and inboard exhaust valves. Both inboard and outboard exhaust valves are cooled by water from the engine salt water system.

6C2. Scavenging air blower. Scavenging air is supplied to the cylinders under a pressure of from 2 to 5 psi by a positive displacement type blower. The blower consists of the housing which has inlet and outlet passages and encloses two three-lobe spiral impellers. The impellers are interconnected by timing gears driven by a gear drive from the upper crankshaft.

Scavenging air from the atmosphere is drawn through the air silencer and enters the inlet passage of the blower. It is moved by the lobes along the walls of the blower housing and forced through the outlet passages and through piping to the air receiver

the upper piston has covered the inlet ports. This allows the inlet air to be built up in the cylinder to the scavenging air pressure, resulting in a certain degree of supercharging.

From the exhaust ports, the exhaust gases pass into the exhaust belt which encloses the lower part of each cylinder liner to a height slightly above the liner exhaust ports. The gases then pass into two exhaust manifolds, one on each side of the engine, along the manifolds to the control end of the engine, thence through two exhaust nozzles or elbows to the exhaust compartments on each side of the cylinder block.

Due to the design of the impeller lobes, the scavenging air is discharged from the blower at a uniform velocity. Efficient operation is possible due to the small clearances between the impellers, the impellers and the blower housing, and the impellers and the bearing plates. Oil should never be allowed to leak into the blower housing or the air receivers. To permit removal of any water that may enter the blower air passages through the air silencer, or indirectly through the exhaust manifold, drain tubes and a

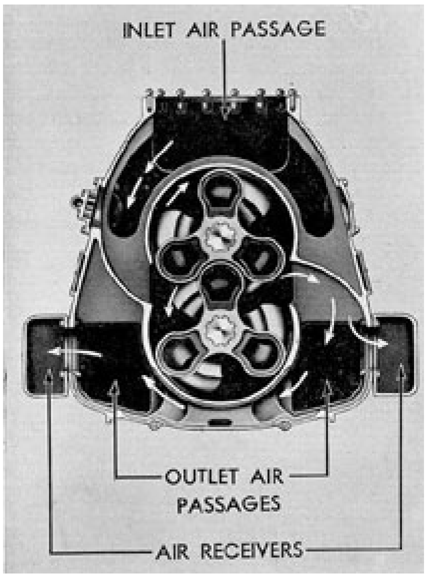


Figure 6-11. Cross section through F-M scavenging air blower.

drain tube cock are provided. This cock should be opened before starting the engine if any abnormal condition is suspected. Opening of this cock will drain the outlet air passages of the blower and the lowest part of the housing.

Each impeller is cast on a splined shaft. Each shaft turns in two roller bearings, the outer bearing taking the shaft thrust. The bearings are held by retainer rings in the end plates which also locate the impellers with radial relation to each other. The thrust bearings prevent contact between the ends of the impellers and the housing.

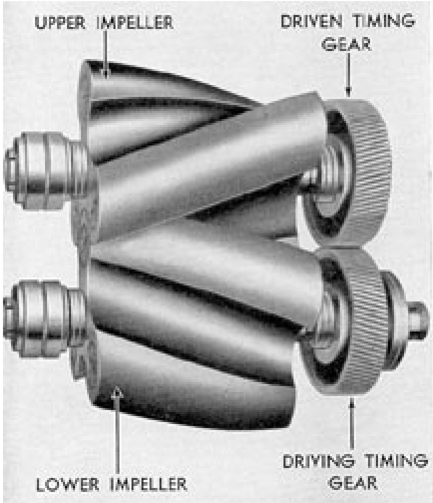


Figure 6-12. Blower impellers and timing gears, F-M.

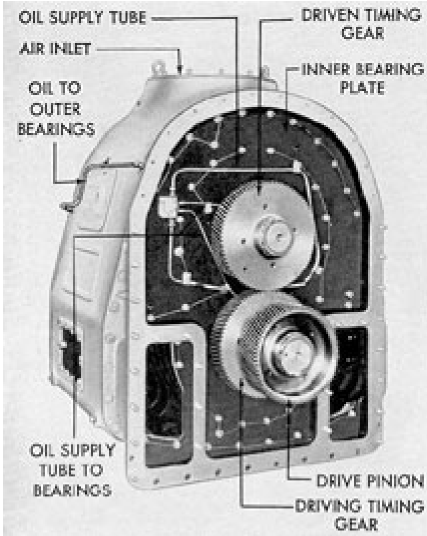


Figure 6-13. Blower assembly, timing gear end, F-M.

Power to drive the blower is transmitted from the upper crankshaft through a flexible gear drive that meshes with a drive pinion. The drive pinion drives the blower driving timing gear, on the end of the lower impeller, which in turn transmits power to the driven timing gear on the end of the upper impeller. The flexible drive gear and drive pinion on the upper crankshaft are lubricated by oil sprayed through nozzles from the engine lubricating system. The blower timing gears and the inner and outer bearings are lubricated by oil through tubes from the engine lubricating system. Oil is

collected between the end cover and the inner housing of the blower and drained to the vertical drive housing from which it returns to the engine oil sump. Gaskets between the bearing plates and blower housing form an oiltight seal.

6C3. Intake silencer. The intake silencer, through which air is drawn before entering the

6C4. Oil separator. The upper crankshaft and the lower crankcase compartments are vented by means of a pipe connected to the suction side of the blower. In the vertical drive compartment this vent line passes through an oil separator, in which a copper ribbon screen prevents oil from being carried into the blower with air from the crankcase. Any leakage of lubricating oil from

scavenging air blower, is similar to the GM unit in design and construction. It is mounted directly over the inlet opening of the blower.

the side covers of the lower crankcase is an indication that the separator needs cleaning.

7
LUBRICANTS AND LUBRICATION SYSTEMS

A. GENERAL

7A1. Purpose of a lubricant in a diesel engine. Lubricating oil in a diesel engine is used for the following purposes:

1. To prevent metal-to-metal contact between moving parts.

2. To aid in engine cooling.

3. To form a seal between the piston rings and the cylinder wall.

4. To aid in keeping the inside of cylinder walls free of sludge and lacquer.

A direct metal-to-metal moving contact has an action that is comparable to a filing action. This filing action is due to minute irregularities in the surfaces, and its harshness depends upon the finish and the force of the contacting surfaces as well as on the relative hardness of the materials used. Lubricating oil is used to fill these minute irregularities and to form a film seal between the sliding surfaces, thereby preventing high friction losses, rapid engine wear, and many operating difficulties. Lack of this oil film seal results in seized, or frozen pistons, wiped bearings, and stuck piston rings. The high-pressures of air and fuel in diesel engines can cause blow-by of exhaust gases circulate as much as 40 gallons of lubricating oil per minute. This illustrates how much of the lubricating oil is used for cooling purposes.

Lubricating oil that is used to form a seal between piston rings and cylinder walls or on any other rubbing or sliding surface must meet the following requirements:

1. The oil film must be of a sufficient thickness and strength, and must be maintained under all conditions of operation.

2. The oil temperature attained during operation must be limited.

3. Under normal changing temperature conditions the oil must remain stable.

4. The oil must not have a corrosive action on metallic surfaces.

It is important not only that the proper type of oil be selected but that it be supplied in the proper quantities and at the proper temperature. Moreover, as impurities enter the system, they must be removed. Diesel engines used in the present fleet type submarines use a centralize pressure feed lubrication system. In this system is incorporated an oil cooler or heat exchanger in

between the piston rings and cylinder liner unless lubricating oil forms a seal between these parts.

Lubricating oil is used to assist in cooling by transferring or carrying away heat from localized hot spots in the engine. Heat is carried away from bearings, tops of the pistons, and other engine parts by the lubricating oil. It is the volume of lubricating oil being circulated that makes cooling of an engine possible. For example, under average conditions, an 8-inch by 10-inch cylinder requires about 24 drops of oil per minute for lubrication of the cylinder wall. About 30 drops of oil per minute normally will lubricate a large bearing when the engine is running at high speed. Yet some engines

which the hot oil from the engine transfers its heat to circulating fresh water. The fresh water is then cooled by circulating sea water inside the fresh water cooler. The heated sea water is then piped overboard.

In order to maintain a strong oil film or body under varying temperature conditions, a lubricating oil must have stability. Stability of the oil should be such that a proper oil film is maintained throughout the entire operating temperature range of the engine. Such a film will insure sufficient oiliness or film strength between the piston and cylinder walls so that partly burned fuel and exhaust gases cannot get by the piston rings to form sludge.

7A2. Chemistry of lubricating oils. As explained in Chapter 5, lubricating oil is the product of the fractional distillation of crude

petroleum. Lubricating oils obtained from certain types of crude petroleum are better adapted for diesel engine use than others, therefore it was formerly highly important that the oils be manufactured from crudes that contained the smallest possible percentage of undesirable constituents. Modern refining methods, by employing such processes as fractionation, filtration, solvent refining, acid treating, and hydrogenation have, however, made it possible to produce acceptable lubricating oils from almost any type of crude oil.

5. Corrosion. The tendency of an oil to corrode the engine parts is known as the corrosive quality of the lubricating oil. The appearance of a strip of sheet copper immersed in oil at 212 degrees F for 3 hours formerly was thought to indicate the corrosive tendency of an oil. This test, however, is not necessarily a criterion of the corrosive tendency of the newer compounded oils, some of which do darken the copper strip but are not corrosive in service. Corrosive oil has a tendency to eat away the soft bearing metals, resulting in serious damage to the bearing.

7A3. Properties of lubricating oils. To insure satisfactory performance a lubricating oil must have certain physical properties which are determined by various types of tests. These tests give some indication of how the oil may perform in practice, although an actual service test is the only criterion of the quality of the oil. Some of the tests by which an oil is checked to conform to Navy specifications are as follows:

1. Viscosity. The viscosity of an oil is the measure of the internal friction of the fluid. Viscosity is generally considered to be the most important property of a lubricating oil since friction, wear, and oil consumption are more or less dependent on this characteristic.

2. Pour point. The lowest temperature at which an oil will barely pour from a container is the pour point. High pour point lubricating oils usually cause difficulty in starting in cold weather due to the inability of the lubricating oil pump to pump oil through the lubricating system.

3. Carbon residue. The amount of carbon left after the volatile matter in a lubricating oil has been evaporated is known as the carbon residue of an oil. The carbon residue test gives an indication of the amount of carbon that may be deposited in an engine. Excessive carbon in an engine leads to operating difficulties.

4. Flash point. The lowest temperature at which the vapors

6. Water and sediment. Water and sediment in a lubricating oil normally are the result of improper handling and stowage. Lubricating oil should be free of water and sediment after leaving the purifier and on arriving at the engine.

7. Acidity or neutralization number. The neutralization number test indicates the amount of potassium hydroxide, in milligrams, necessary to neutralize one gram of the oil tested. It is, therefore, proportional to the total organic and mineral acid present. The results are apt to be misleading or subject to incorrect interpretation, since the test does not distinguish between corrosive and noncorrosive acids, both of which be present. The chief harm resulting from the presence of organic acid, which is noncorrosive, is its tendency to emulsify with water. This emulsion picks up contaminants and is a sludge which may interfere with proper oil circulation. The neutralization number of new oils is generally so low as to be of no importance.

8. Emulsion. The ability of an oil to separate from water in service is known as the emulsibility of the lubricating oil. The emulsibility of a new oil has little significance. Two oils that have different emulsifying tendencies when new, may have the same emulsion tendency after being used in an internal combustion engine for a few hours. The emulsibility of an oil that has been in use for some time is important.

9. Oiliness or film strength. The ability of a lubricating oil to

of a heated oil will flash is the flash point of the oil. The flash point of an oil is the fire hazard measure used in determining storage dangers. Practically all lubricating oils have flash points that are high enough to eliminate the fire hazard during storage in submarine, tender, or base stowage facilities.

maintain lubrication between sliding or moving surfaces under pressure and at local high temperature areas is known as the oiliness or film strength of the oil. Film strength is the result of several oil properties, the most important being viscosity.

10. Color. The color of a lubricating oil is useful only for identification purposes and has nothing to do with lubricating qualities. If the color of a nonadditive oil is not uniform, it may indicate the presence of impurities; however, in additive lubricating oils, a nonuniform color means nothing.

11. Ash. The ash content of an oil is a measure of the amount of noncombustible material present that would cause abrasion or scoring of moving parts.

12. Gravity. The specific gravity of an oil is not an index of its quality, but is useful for weight and volume computation purposes only.

13. Sulphur. The test for sulphur indicates the total sulphur content of the oil and does not distinguish between the corrosive and noncorrosive forms. A certain amount of noncorrosive sulphur compounds is allowable, but the corrosive compounds must be eliminated because of their tendency to form acid when combined with water vapor.

wear. In the bearings, however, the temperatures are lower and the rotation tends to create a fluid film permitting a lighter oil to be used. When a single lubricating system supplies oil to cylinders and bearings, it is necessary to compromise on an oil that will do the best job possible in both places. All modern submarine diesel engines are of the latter type, having a single lubricating system.

Temperature, however, is not the only consideration in selecting an oil of the proper viscosity. Clearances, speed, and pressures are also important factors. Their effects on required viscosity may be summarized as follows:

1. Greater clearances always require higher viscosity.

2. Greater speed requires lower viscosity.

3. Greater load requires higher viscosity.

The oil selected for a diesel engine is therefore a compromise between a high- and a low-viscosity oil. Most high-speed engines run better using low-viscosity oils, but the viscosity

14. Detergency. The ability of an oil to remove or prevent accumulation of carbon deposits is known as its detergent power.

7A4. Viscosity of lubricating oils. The viscosity of a lubricating oil at the operating temperature in the engine is one of the most important considerations in selecting oil, since viscosity is the characteristic that determines film thickness and the ability to resist being squeezed out. The viscosity of an oil changes with temperature. Therefore, the viscosity should be measured at the operating temperatures of that particular part of the engine which the oil is to lubricate. From the viewpoint of lubrication, engines can be considered in two classes, those in which the cylinders and bearings are lubricated separately, and those in which only one lubricating system is used. If there are separate lubrication systems for cylinders and bearings, it is possible to use two grades of oil, the heavy one for cylinders and a medium one for bearings. The operating temperature to which the oil is subjected in the cylinders is naturally much higher than in the bearings. Also the motion in a cylinder is sliding, and a heavier oil is required to provide sufficient body to prevent metallic contact and must not be so low that the oil film wedge is too thin for efficient lubrication. On the other hand, oil of a greater viscosity than necessary should not be used because:

1. An oil of too great a viscosity increases starting friction.

2. Increased friction raises oil temperatures, and thereby promotes oxidation.

3. The more viscous oils usually have a higher carbon residue.

4. An oil of too great a viscosity places an overload on the lubricating oil pump with a possible inadequate supply reaching some moving parts.

For practical purposes the viscosity is determined by noting the number of seconds required for a given quantity of oil to flow through a standard orifice at a definite temperature. For light oils the viscosity is determined at 130 degrees F, and for heavier oils at 210 degrees F. The Saybolt type viscosimeter with a Universal orifice is used for determining the viscosity of lubricating oils. The longer it takes an oil to flow through the orifice, at a given temperature, the heavier or more viscous the oil is considered.

7A5. Tests. Viscosity tests are frequently conducted on board ship to determine the amount of dilution caused by leakage of fuel oil

into the lubricating oil system. The test is made with a Visgage

(Figure 7-1), a small instrument consisting of two glass tubes, each of which contains a steel ball, and a scale calibrated to indicate seconds Saybolt Universal (SSU) at 100 degrees F. One of the glass tubes is sealed and contains oil of a known viscosity. The other has a nozzle at one end and contains a plunger with which the oil to be tested is drawn into the tube. The instrument should be warmed by hand for a few minutes so that the temperature of the sample oil will be the same as that of the oil sealed in the master tube. Then, starting with both steel balls at the zero marking on the scale, the instrument is tilted so that the balls will move through the oil. On the instant that the leading ball reaches the 200 marking at the end of the scale, the position of the other ball in relation to its scale is noted. That reading indicates the viscosity of the sample oil in SSU at 100 degrees F direct.

The percentage of dilution of the lubricating oil by the diesel fuel oil is determined by use of the viscosity blending chart. This chart is essentially a graph of oil viscosity against percentage. Both right and left vertical boundary lines are marked in terms of viscosity SSU. The horizontal lines are divided into percentages from 0 to 100 percent. In using the viscosity blending chart, a line is drawn between the lubricating oil viscosity marked on the left vertical boundary line and the diesel fuel oil viscosity marked on the right vertical boundary line. This line represents only one

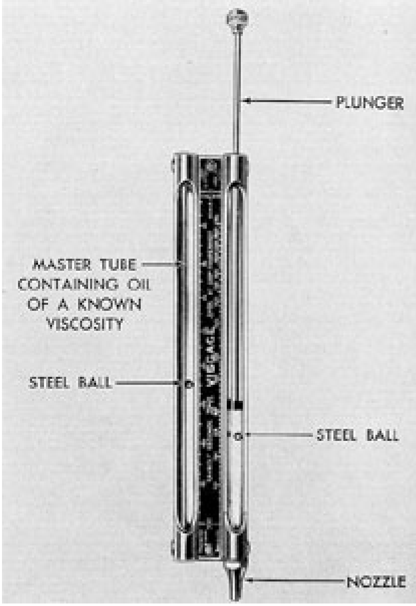


Figure 7-1. Visgage.

As shown on the chart, the dilution is approximately 5 percent.

7A6. Detergent lubricating oils. Detergent or additive oils as they are usually called, consist of a base mineral oil to which chemical additives have been added. The additive agent has the following beneficial effect on the performance of the base lubricant:

1. It acts as an oxidation inhibitor.

2. It improves the natural detergent property of the oil.

3. It improves the affinity of the oil for metal surfaces.

For Navy use, heavy duty detergent lubricating oils of the 9000 series are used in most diesel installations. The use of these oils in a diesel engine results in a reduction in ring sticking and gum or varnish formation on the piston and other parts of the engine. In dirty engines, a heavy duty detergent oil will gradually remove gummy and carbonaceous deposits. This material being carried in suspension in the oil will

particular lubricating oil viscosity. Figure 7-2 is an expanded portion of one section of a viscosity blending chart with lines drawn in for Navy symbol lubricating oils most commonly used. To determine the percent dilution of a lubricating oil, the viscosity of a test sample of the used oil is obtained, usually with a Visgage. The intersection of this valve on the chart with the line representing the Navy symbol oil in use gives a direct reading of the percentage of dilution on the horizontal scale.

Example:

	SSU at 100 degrees F
New lubricating oil, viscosity 9250	550
Diesel fuel oil	37
Used lubricating oil (measured by Visgage)	420

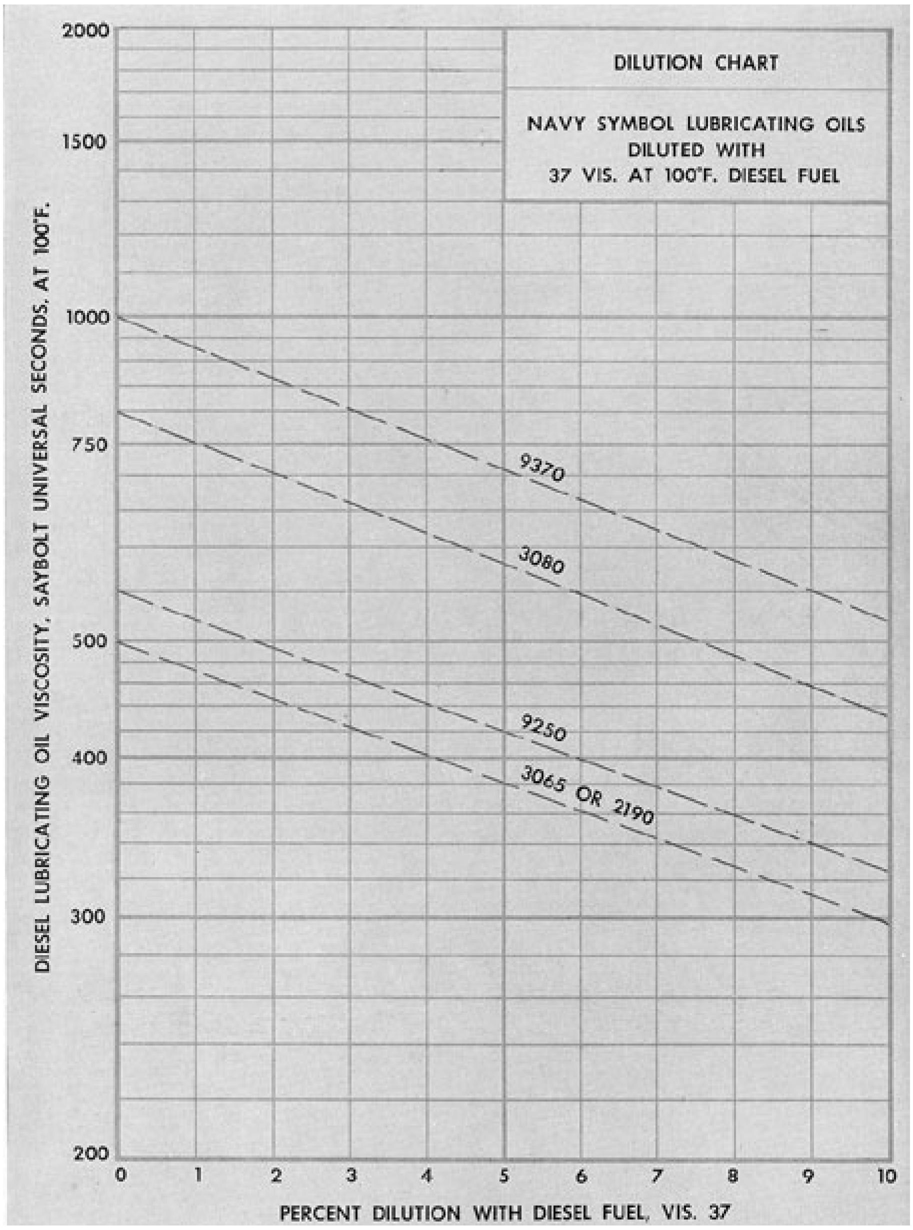


Figure 7-2. Section of viscosity blending chart.

tend to clog the oil filters in a relatively short time. Normally, a dirty engine will be purged with one or two fillings of the sump, depending upon the condition of the engine and the quantity of the oil used. During the cleaning-up process, the operator should drain the sump and clean the filter if the oil gage indicates an inadequate oil flow.

In using additive or detergent type oils the following points should be considered:

2. Carbon caused by the evaporation of oil on a hot surface, such as the underside of a piston.

3. Gummy, partially burned fuel which gets past the piston rings.

4. An emulsion of lubricating oil and water which may have entered the system.

Sludge is often attributed to the breaking down of lubricating oil, but generally this is not true.

Sludge gathers many dangerous ingredients, such as dust from the

1. All Navy approved oils are miscible. However, to obtain the maximum benefit from additive oils, they should not be mixed with straight mineral oils except in emergencies.

2. Detergent oils on the approved list are not corrosive. Should ground surfaces be found etched, or bearings corroded, it is probable that contamination of the lubricant by water or partially burned fuel is responsible. It is important that fuel systems be kept in good repair and adjustment at all times. The presence of water or partially burned fuel in lubricating oil is to be avoided in any case, whether mineral oil or detergent oil is used. However, small quantities of water in the Navy symbol 9000 series oils are no more harmful than the same amount of water in straight mineral oils. They will not cause foaming nor will the additives in the oils be precipitated

7A7. Sludge. Almost any type of gummy or carbonaceous material accumulated in the power cylinder is termed sludge. The presence of sludge is dangerous for several reasons:

1. Sludge may clog the oil pump screen or collect at the end of the oil duct leading to a bearing, thereby preventing sufficient oil from reaching the parts to be lubricated.

2. Sludge will coat the inside of the crankcase, act as an insulation, blanket the heat inside the engine, raise the oil temperature, and induce oxidation.

atmosphere, rust caused by water condensation in the engine, and metallic particles caused by wear, which contribute to premature wear of parts and eventual break down of the engine.

7A8. Bearing lubrication. The motion of a journal in its bearing is rotary, and the oil tends to build up a wedge under the journal. This oil wedge lifts the journal and effectively prevents metallic contact. The action of the oil film is explained in Figure 7-3 which illustrates the hydrodynamic theory of lubrication. This theory, involving the complete separation of opposing surfaces by a fluid film, is easily understood when the mechanism of film formation in a plain bearing is known. The diagram shows first the bearing at rest with practically all of the lubricant squeezed from the load area. Then, as rotation begins, an oil film is formed which separates the journal from the bearing. When rotation starts with the clearance space filled with oil there is a tendency for the journal to climb or roll up the bearing as a wheel rolls uphill. As the center of the bearing does not coincide with the center of the journal, the clearance space is in the form of a crescent with its wedge-shaped ends on either side of the contact or load area. Because of the fact that oil is adhesive and sticks to the journal, rotation causes oil to be drawn into the wedge-shaped space ahead of the pressure area. As the speed of rotation increases, more oil is carried into the wedge by the revolving journal, and sufficient hydraulic pressure is built up to separate completely the journal and bearing. When this

3. Sludge will accumulate on the underside of the pistons and insulate them, thereby raising piston temperatures.

4. Sludge in lubricating oil also contributes to piston ring sticking.

Sludge is usually formed by one or a combination of the following causes:

1. Carbon from combustion chambers.

film has formed, the load on the journal tends to

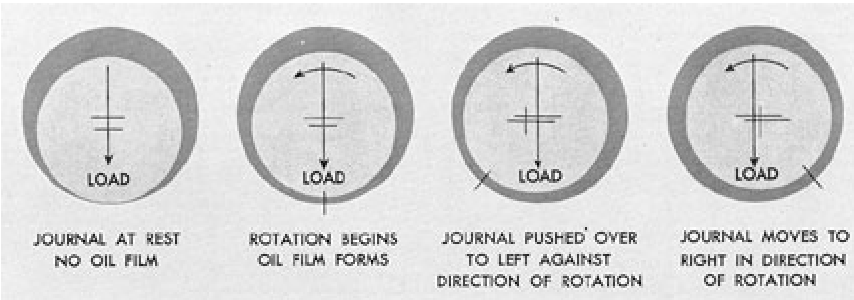


Figure 7-3. Formation of bearing oil film.

cause it to drop to the lowest point. However, the pressure built up in the converging film ahead of the pressure area tends to push the journal to the other side of the bearing. The wedging action of the oil builds up a film pressure of several hundred pounds per square inch. The oil pump pressure, however, need only be sufficient to insure an adequate supply of oil to the bearings. All oil openings should be in the low-pressure section of the bearing in order to keep the lubricating oil pump pressure to a minimum. Diesel bearing pressures normally are not much over 1000 psi, and an oil film of straight mineral oil will usually withstand pressures of over 5000 psi.

result from either a lack of sufficient lubricant or the use of an improper lubricant. Lack of lubricant may be due to excessive bearing wear, excessive bearing side clearance, low oil level, low oil pressure, and plugged oil passages. Failure, due to the use of an improper oil, results not only from incorrect original lubricant, but more frequently from continued use of an oil that should be replaced. Viscosity, in particular, is subject to change due to bearing temperature variation, dilution by unburned fuel, and oxidation. Bearing temperature variation is controlled by the proper operation of the cooling system. Lubricating oils may become corrosive in service, due to contamination by products of combustion or to inherent

The viscosity required to produce the proper oil film thickness depends on several factors. A rough or poor bearing needs a more viscous oil than a smooth, properly fitted bearing. Bearing clearances should always be enough to form an oil film of the proper thickness. Excessive bearing clearances reduce the oil pressure and only an excessively viscous oil will stay between the bearing surfaces. The greater the load on the bearing, the greater the oil viscosity required to carry the load. On the other hand, higher speeds permit a reduction in viscosity since the high shaft rotation helps build up the oil film pressure.

Bearing trouble and failure are usually attributable to improper lubrication. This may characteristics of the oil itself. Bearing corrosion is, of course, most likely to occur at high temperatures.

To insure against corrosion, the lubricating oil should be changed frequently, especially if oil temperatures are high or if easily corroded bearing materials are used. A pitted bearing usually indicates corrosion, which may be due to fuel, lubricant, or water.

7A9. Cylinder lubrication. The oil supplied to the cylinders must perform the following functions:

1. Minimize wear and frictional losses.

2. Seal the cylinder pressures.

3. Act as coolant.

If no lubricant were employed, the metal surfaces would rub on one another, wearing away rapidly and producing high temperatures. The cylinder oil must prevent, as much as possible, any metallic contact by maintaining a lubricating film between the surfaces. Since oil body, or viscosity, determines the resistance of the oil against being squeezed out, it might seem that the thicker the oil, the better. This holds true in regard to wear, but there are other factors to be considered The body of the oil which prevents the film from being removed from the rubbing surfaces also provides a drag, resisting motion of the piston and reducing the power output of the engine. In

The oil aids in cooling by transmitting heat from the piston to the cylinder wall. To fulfill this requirement the oil should be as light as possible, since with light oils there is more movement in the oil film, a condition which aids the transfer of heat.

7A10. Navy specifications and symbols for lubricating oil. The symbol numbers used in Navy lubricating oil classification tables are for the ready identification of the oils as to use and viscosity. Each number consists of four digits, of which the first classifies the oil according to its use, and the last three indicate its viscosity. For example, the symbol 2250 indicates that the oil is a force feed oil (viscosity measured at 130

addition, an oil that is too heavy does not flow readily, and spots on the cylinder walls remote from the point of lubrication may remain dry, causing local wear. Very heavy oils tend to remain too long on the piston lands and in ring grooves. While this condition may result in lower oil consumption, it will eventually cause gumming due to oxidation of the oil, and the final result will be sticky rings. For cylinder lubrication, therefore, it is desirable to use the lightest possible oil that will still keep the cylinder walls and piston lubricated. Use of a light oil will result in faster flow of the oil to the parts requiring lubrication, reduce starting wear, and minimize carbon deposits. This will result in lower fuel consumption, lower temperatures, longer periods between overhauls, and finally, lower total operating costs The lubricating oil consumption will probably be slightly higher, but the saving in fuel alone will more than make up for the additional lubricating oil expense.

The sealing function of the oil is tied in with its lubricating property. In order to make a good seal, the oil must provide a film that will not be blown out from between the ring face and the cylinder wall nor from the clearance space between the ring and the sides and back of the ring groove. The effectiveness of this seal depends partly upon the size of the clearance spaces. With a carefully fitted engine, in which clearances are small, a light oil can be used successfully. If the oil is heavy enough to provide a good seal, it will have a degrees F) and has a viscosity of 250 seconds Saybolt Universal. The following is a list of the classification of lubrication oils as to use:

Series	Classification	Navy Symbol Examples
1000	Aviation oils	1065, 1080, 1100, 1120, 1150
2000	Forced feed oils (viscosity measured at 130 degrees F)	2075, 2110, 2135, 2190
3000	Forced feed oils (viscosity measured at 210 degrees F)	3065, 3080, 3100
4000	Compound marine engine oils	4065
5000	Mineral marine engine and cylinder wall oils	5065, 5150, 5190
6000	Compounded steam cylinder oil (tallow)	6135
7000	-	-
8000	Compounded air compressor cylinder oils	8190
9000	Compounded or additive type heavy duty lubricating oils (viscosity measured at 130 degrees F)	9110, 9170, 9250, 9370, 9500

good margin of safety for the requirement usually stressed, that of preventing metallic contact.

The most common lubricating oil classification is that known as the SAE (Society of Auto motive Engineers) classification. Since the SAE numbers are more generally used outside of the Navy, a comparison showing the viscosity limits of the various numbers is given in the accompanying table.

SAE No.	Viscosity Seconds Saybolt	
	At 130 degrees F	At 210 degrees F
10	90-120	
20	120-185	
30	185-255	
40	255-	80
50		80-105
60		105-125
70		125-150

B. LUBRICATING SYSTEMS

7B1. Basic requirements of a lubricating system. Lubrication is perhaps the most important single factor in the successful operation of diesel engines. Consequently, too much emphasis cannot be placed upon the importance of the lubricating oil system and lubrication in general. It is not only important that the proper type of oil be used, but it must be supplied to the engine in the proper quantities, at the proper temperature, and provisions must be made to remove any impurities as they enter the system. In general, the basic requirements that a lubricating system must meet to perform its functions satisfactorily are:

1. An effective lubricating system must correctly distribute a proper supply of oil to all bearing surfaces.

2. It must supply sufficient oil for cooling purposes to all parts requiring oil cooling.

3. The system must provide tanks to

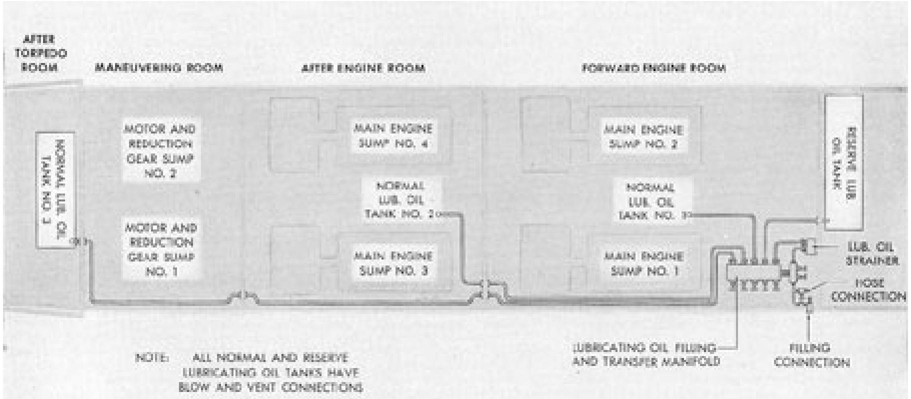


Figure 7-4. General arrangement of lubricating off tanks.

collect the oil that has been used for lubrication and cooling, so that it can be recirculated throughout the system.

4. The system must include coolers to maintain the oil temperature within the most efficient operating temperature range.

5. In order to exclude dirt and water from the working parts of the engine, filters and strainers must be included in the system to clean the oil as it circulates.

6. Adequate facilities must be provided on the ship for storing the required quantity of lubricating oil necessary for extensive operation and for transferring this oil to the engine lubricating systems as needed.

7B2. Ship's lubricating oil tanks and sumps. A typical lubricating oil system installation on recent submarines consists of three normal lubricating oil tanks and one reserve lubricating oil tank. These tanks are located inside the pressure hull adjacent to the engineering spaces and have approximately the following capacities:

A filling connection is provided on the main deck to a five-valve filling and transfer manifold located on the starboard side of the forward engine room. This manifold is connected not only to the filling connection, but also directly to each of the normal lubricating oil tanks and the reserve lubricating oil tank. The oil to fill the tanks normally is passed through a strainer before it reaches the filling and transfer manifold. This oil strainer may be bypassed. A drain from the bottom of the strainer makes it possible to drain out any salt water that might have leaked into the filling line through the outboard filling connection.

The tanks are provided with vents and air connections from the 225-pound air service lines. By the use of these lines, lubricating oil may be blown from any lubricating oil storage tank to any other lubricating oil tank. Oil to be discharged may be blown or pumped overboard through the deck filling connection or through a hose connection in the filling line.

7B3. Operation of engine lubricating oil system. Oil is drawn from the sump tank by the

Normal lubricating oil tank No. 1	1534 gallons
Normal lubricating oil tank No. 2	973 gallons
Normal lubricating oil tank No. 3	1092 gallons
Reserve lubricating oil tank	1264 gallons

In addition to these storage tanks, there is a sump tank under each main engine and under each of the two reduction gears. These tanks collect the oil as it drains from the engine oil pans. The sump tanks are always partially filled in order to insure a constant supply of oil to the lubricating oil pumps. As, the sump tanks are never completely filled with lubricating oil, their capacity is usually indicated as 75 percent of the actual total tank capacity. The approximate capacities of the various sump tanks (at 75 percent) are:

Main engine sump tanks Nos. 1, 2, 3, 4	382 gallons each
Motor and reduction gear lubricating oil sumps Nos. 1, 2	165 gallons each

attached lubricating oil pump. The discharge from this pump passes through the lubricating oil strainer. Between the discharge side of the pump and the strainer is a relief valve built integral with the pump. From the strainer the oil is carried to the lubricating oil cooler and thence to the engine main lubricating oil headers. The strainer is always placed forward of the cooler in the system because, if the temperature of the lubricating oil is higher, its filtering efficiency will be greater and the power necessary to force the oil through the strainer will be less.

In most installations the lubricating oil goes from the main lube oil headers to the engine main bearings and thence to the connecting rod bearings. The oil then passes through a drilled hole in the connecting rod up to the piston pin bearing which it lubricates and sprays out onto the under surface of the piston crown. Next, it drains down into the oil drain pan, carrying away from the piston much of the heat caused by combustion. From the oil pan, the oil drains to the engine sump tank from which it is recirculated

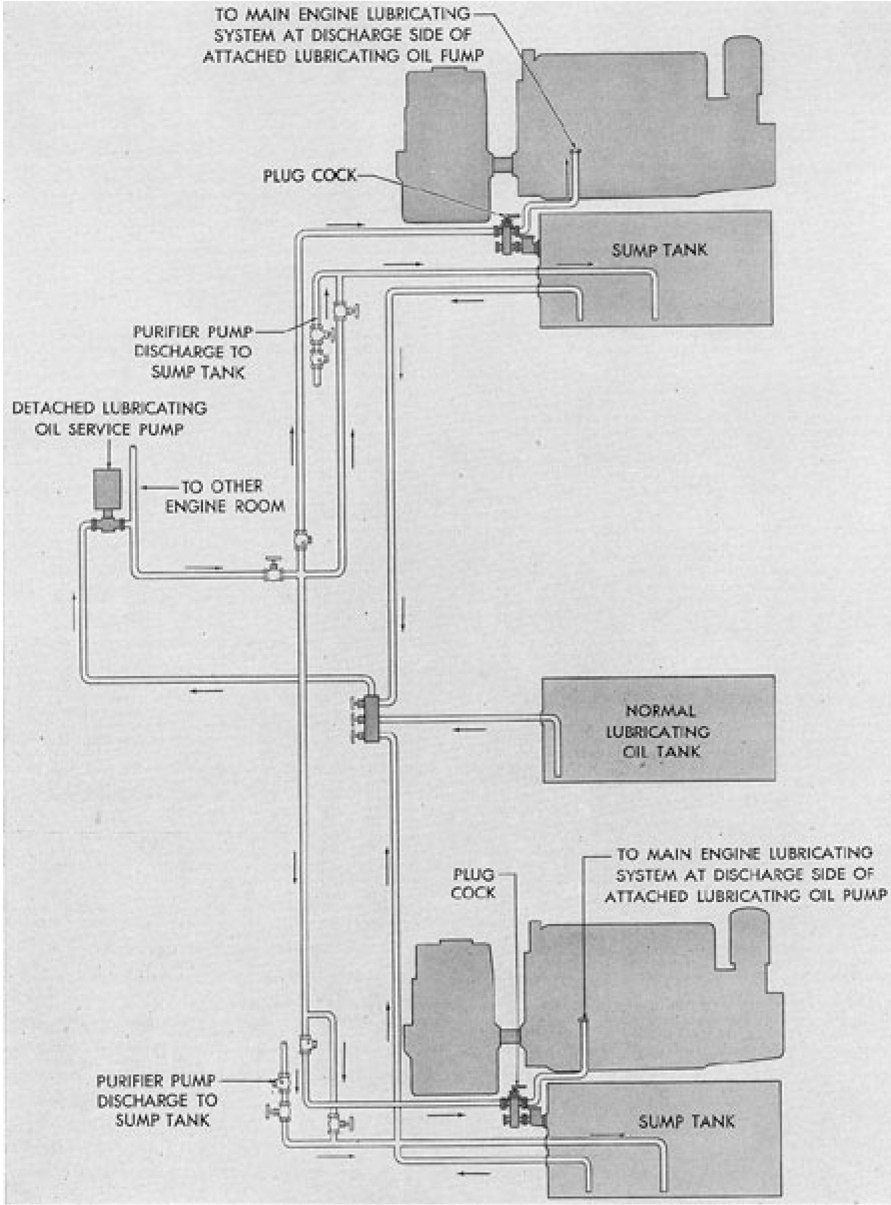


Figure 7-5. Typical lubricating oil flushing and filling system.

Between the oil pan and the sump tank, screens and basket type strainers may be inserted to prevent small metallic particles from draining down into the sump tank.

Lubricating oil for the main generator bearings is also provided by the main lubricating oil system. The oil used for this purpose is piped from the main lubricating oil line, at a point just before it enters the engine oil header, to the tops of the generator main bearings. From the bottoms of the bearings, the their respective sump tanks are shown, together with the piping that connects these units with the auxiliaries necessary for lubricating oil purification. These include a lubricating oil purifier, detached lubricating oil service pump, lubricating oil heater, and lubricating oil filters. The normal path of the oil during purification is from the sump tanks to the lubricating oil service pump thence to the oil heater, the purifier, the filters, and then back to the sump tanks. In actual installations, the filling and flushing and the purifying systems are combined in

oil drains back to the sump tank, either directly or through the engine oil system.

The attached lubricating oil pumps are driven directly by the engines and therefore cannot be used for priming the lubricating oil system before starting. For this purpose, detached lubricating oil service pumps are provided, one in each engine room. These pumps should be started approximately five minutes before starting an engine. When an engine has been started and its attached pump is supplying oil to the engine system, the service pump may be shut down. The service pumps may also be used to circulate lubricating oil to cool an engine after it has been stopped.

Figure 7-5 shows a typical lubricating oil flushing and filling system in one engine room. In this system the detached lubricating oil service pump may be used to prime the engine lubricating oil systems prior to starting, to replenish the sump tanks from the normal lubricating oil stowage tank, and possibly to flush out the engine lubricating oil system when necessary. When the system is used for priming, the detached service pump takes a suction from the sump tank and discharges the oil into the engine lubricating oil system at the discharge side of the attached lubricating oil pump. When the detached pump is used for replenishing the sump tanks, it takes a suction from the normal lubricating oil tank and one system. For clarity the systems are separated as shown in Figures 7-5 and 7-6.

The lubricating oil pumps are designed to deliver considerably more oil than is normally required to pass through the engines. This insures sufficient lubrication when changes in the rate of oil flow occur because of cold starting, changes in speed, changes in viscosity of the oil due to heat, or increases in bearing clearances.

Pressure gages are placed in the system to indicate the pressures of the lubricating oil entering the strainer, leaving the strainer, and entering the engine. Through a change in pressure readings at these gages, troubles such as air binding of pumps, broken supply lines, or dirty strainers may be localized and remedied.

The lubricating oil is cooled by fresh or salt water circulating through an oil cooler. The pressure of the lubricating oil is higher than the pressure of the water so that, in the event of a leak, water cannot enter the oil system.

7B4. Detached lubricating oil service and standby pumps. All fleet type submarines use a detached lubricating oil service pump in each engine room for the purpose of supplying the purifier, filling the sump tanks from the storage tanks, and for flushing and priming the engine lubricating oil system. These ships also have a standby pump located in the maneuvering room for the purpose of filling the lubricating oil storage tanks, discharging used oil from the ship, and for transferring oil from one tank to

discharges the oil to either sump tank as necessary.

Lubricating oil may be purified by drawing the oil from the sump tanks with the service pump and discharging the oil back to the sump tanks through a purifier. Figure 7-6 illustrates a typical main engine lubricating oil purifying system for one engine room. Two engines and an other. This pump also serves to supply the main motor bearings and reduction gears in the event that the oil pressure in that system drops below the safe operating limit, or the reduction gear sump pumps become inoperative. Both the

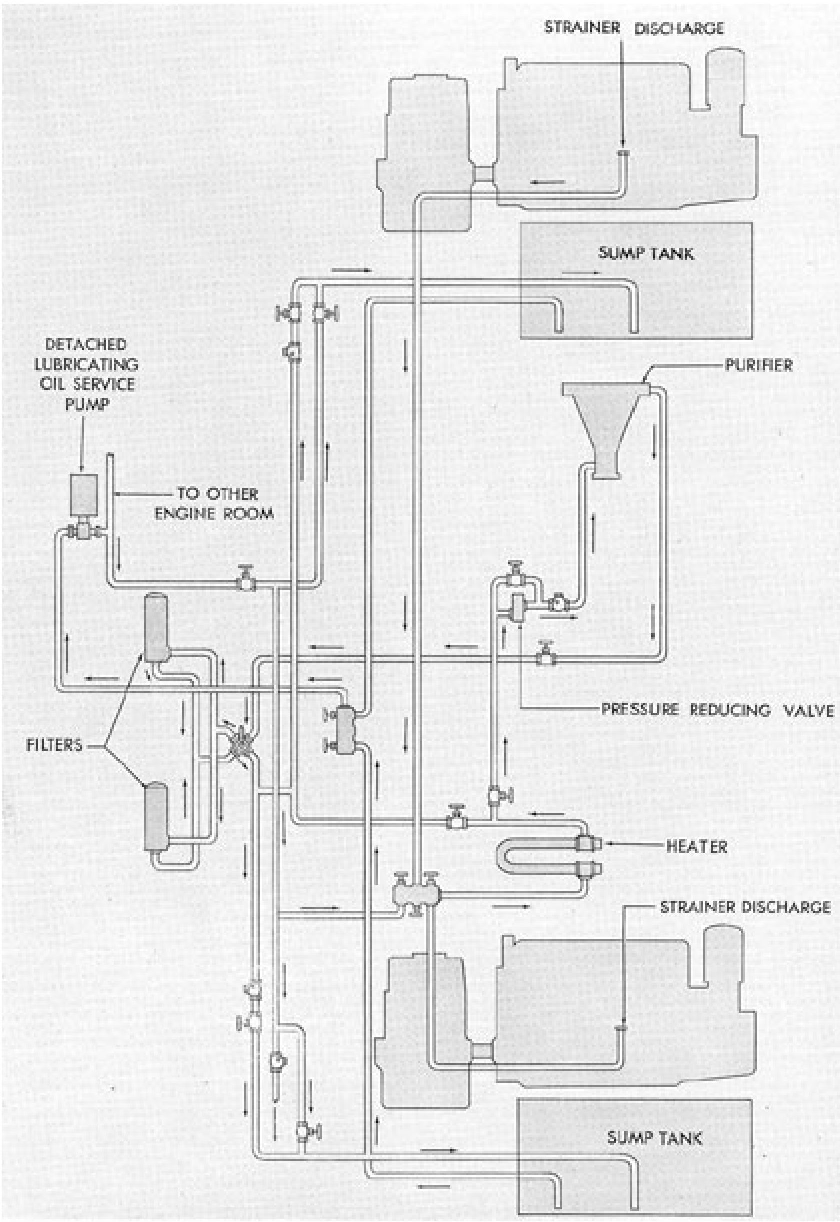


Figure 7-6. Typical main engine lubricating oil purifying system in one engine room.

detached service and standby pumps are of the positive displacement type, driven by electric motors.

7B5. Lubricating oil coolers. The oil cooler is a Harrison radiator heat exchanger. This cooler is made up of a tube bundle or core and an enclosing case. The tubes are oblong and each tube encloses a baffled structure which forms a winding passage for the flow of oil. The tubes are fastened in place with a header plate at each end and with an intermediate reinforcement plate. These plates are electroplated with tin. The tube and plate assembly is mounted in a bronze frame by means of which the tube bundle is fastened to the covers on each end of the casing.

The header plates, at the end of the tubes, separate the water space in the casing from the lubricating oil ports in the end covers. The lubricating oil flows through the tubes in a straight path from one cover port to the other. The intermediate tube plate acts as a baffle to form a U-shaped path for the water, which flows around the outside of the tubes, from one opening in the bottom of the casing to the other.

All the lubricating oil coolers are provided with zincs which act as electrodes. Electrolytic action is always present in all water systems on a submarine, and these electrodes allow the zinc rather than the cooler tubes to be eaten away. Zincs are mounted on removable plates

lubricating oil purifying system on the discharge side of the purifier. Various types of strainers and filters may be found in service. Some strainers consist of an element of edge-wound metal ribbon, others use a series of edge type disks. Filters may employ absorption type cellulose, waste, or wound yarn elements which are replaced when dirty. A few of the commonly used strainers and filters are described in the following paragraphs.

b. Edge disk type strainer. The edge disk type of lubricating oil strainer consists of an assembly of thin strainer disks separated slightly by spacer disks. The lower end of this assembly is closed and the upper end is open to the strainer discharge. The oil comes into the strainer and is forced through the strainer disks into the center of the strainer assembly. The oil then passes up through the assembly and out the top of the strainer. In passing through the strainer, the oil must pass through the slots between the strainer disks. In the bottom of the strainer element a relief valve is provided to avoid the possibility of excess pressure building up in the strainer should the slots become filled with foreign matter. This relief valve bypasses the oil up through the center of the strainer element and out the strainer discharge. The valve is set to open when the differential pressure reaches 10 psi. The disadvantage of this relief valve is that its functioning allows any foreign matter that may have collected in the bottom of the strainer to pass to the discharge side of the strainer and into the lubricating oil system.

and should be replaced when they show marked deterioration.

In all cooling systems it is a universal rule that the pressure of the liquid cooled be greater than that of the cooling agent. In a lubricating oil cooler this means that the pressure of the lubricating oil should be greater than the pressure of the fresh or salt water, whichever is used. If a leak should develop in the system, the water would then be prevented from leaking into the lubricating oil.

7B6. Lubricating oil strainers and filters. a. General. Strainers and filters are incorporated in the lubricating oil system for removal of foreign particles. In most installations the oil is passed through two strainers located forward of the cooler. Filters generally are located in the

When the assembly is turned by means of the external handle, the solids that have lodged against or between the disks are carried around until they meet the stationary cleaner blades. The stationary cleaner blades comb the solids clear of the strainer surface. The solids are compacted by the action of the cleaner blades and fall into the sump where they are filtered out of the stream of incoming oil. To keep the strainer in its clean and free filtering condition, the external handle is given one or more complete turns in a clockwise direction at frequent intervals. It is therefore not necessary to break any connections or interrupt the flow of oil through the strainer to clean the strainer unit.

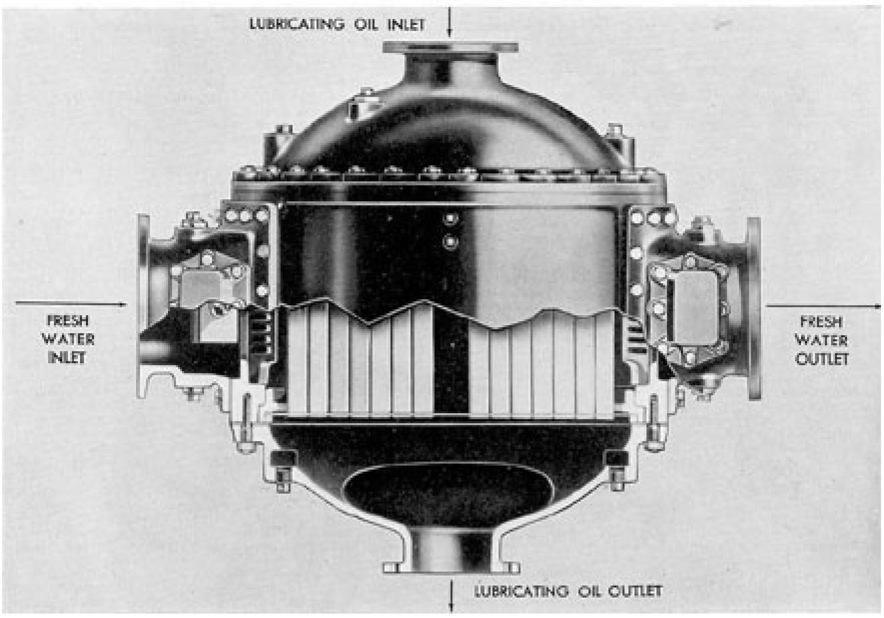


Figure 7-7. Cutaway of latest type Harrison heat exchanger.

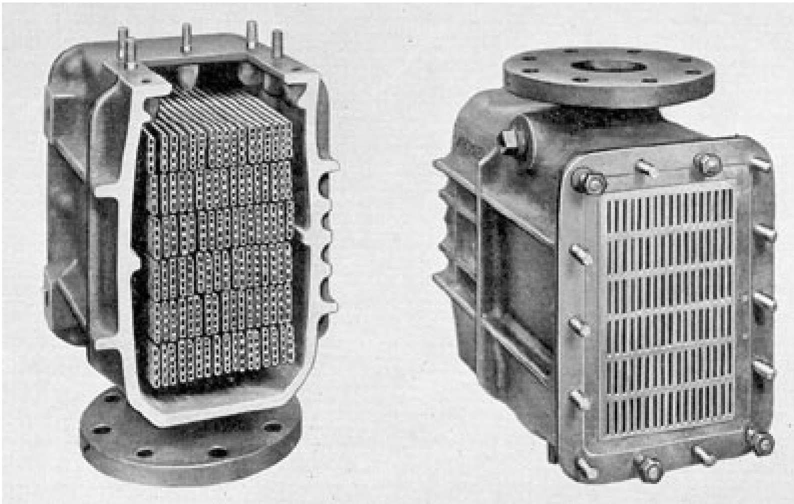


Figure 7-8. Cutaway of older type Harrison heat exchanger showing internal construction.

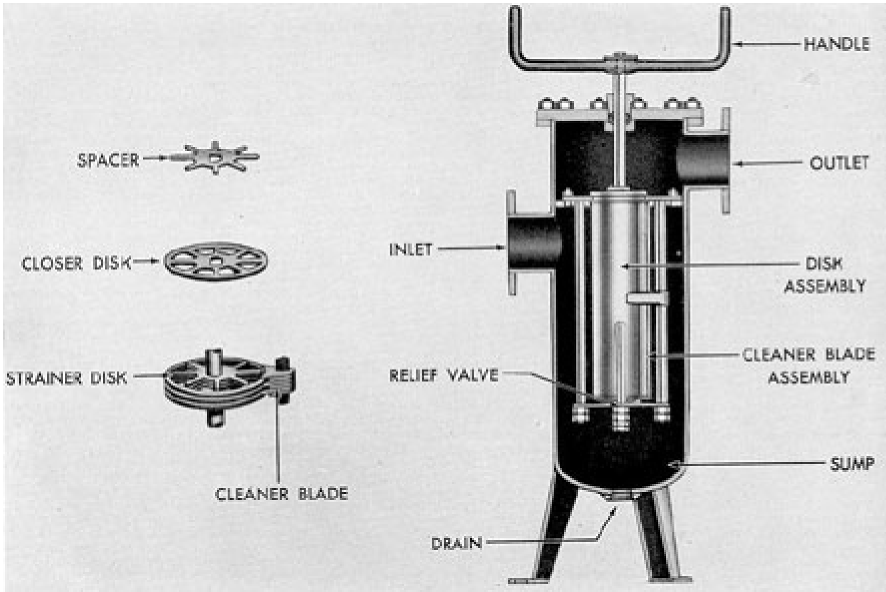


Figure 7-9. Edge disk type oil strainer.

If the handle turns hard it indicates that the strainer surfaces have heavy deposits of solids on them. The handle should be turned frequently; there is no danger of turning the handle too often as there are no parts to wear out. If the strainer cannot be cleaned by turning, the head and disk assembly must be removed and soaked in a solvent until the solids have been removed. A wrench or other type of tool should never be used to turn the strainer handle. During periods of overhaul the head and disk

This strainer, manufactured by the Purolator Company, is so constructed that the oil required by the engine is continuously filtered except when its filtering element must be removed for cleaning or servicing. When this is done, the control valve handle is turned to the bypass position. This shunts the oil flow through the filter head, permitting removal of the element without interruption of oil flow to the engine. Under normal conditions the oil comes into the strainer and surrounds the ribbon element. It then passes through and up the center of the

assembly should be removed and the disk assembly rinsed in a clean solvent. The disk assembly should never be disassembled. If it is in such a condition as to warrant disassembly it should be replaced with a new unit. When cleaning the disk assembly, the strainer body and sump should be thoroughly drained and cleaned. Extreme care is necessary when cleaning the strainer, to prevent injury to the strainer element and the introduction of dirt and foreign material into the clean side of the strainer.

c. Edge-wound metal ribbon type strainer.

strainer element to the outlet passage. One complete turn of the cleaning handle on top of the element rotates the element winding, and foreign material is removed from the element. The element consists of a cage of accurately spaced slots or perforations covered with a continuous, closely compressed coil of stainless steel wire. The wire is passed between rollers to produce a wedge-shaped wire or ribbon, one edge thicker than the other. On one side, projections are spaced at definite intervals while the other

side is smooth. The projections on one side of the wire touch against the smooth side of the wire on the next coil to provide a spacing of approximately 0.005 inch. The thick edge of the wire is on the outside of the coil so that a tapered slot is formed through the coil, with the narrowest part of the slot on the outside. This insures that the dirt particles small enough to pass the outside, or narrowest point will not become stuck halfway and clog the oil flow. The dirt removed from the oil remains on the outside and can readily be removed by rotation of the cleaning handle.

The control valve handle on the strainer operates the bypass valve. When the handle is

The pressure drop through the strainer is an indication of the condition of the straining element. When the pressure drop becomes abnormal and cannot be reduced by turning the cleaning handle, the strainer element should be removed and cleaned with an approved solvent. Care must be taken to prevent entrance of dirt to the inside of the element while it is being washed. The strainer element should not be cleaned with a wire brush or a scraper. The drain plug may be removed when the element is bypassed, thereby making it possible to drain out sludge and foreign material from the bottom of the strainer.

Most filters of this type have a relief valve installed in the lower end of the element. This valve lifts when there is a differential pressure of 7 to 10 psi. This design makes it possible for dirt to be

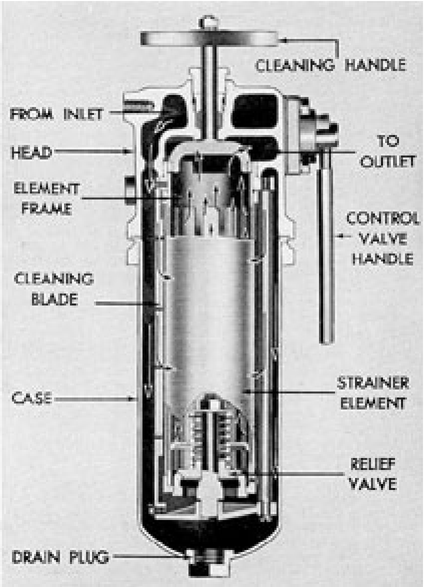


Figure 7-10. Cutaway of edge-wound metal ribbon type oil strainer.

in the ON position, the lubricating oil is flowing through the strainer. When the handle is in the BYPASS position, the oil is flowing directly through the head of the unit, and the strainer case and element can be removed and cleaned. The ON and BYPASS positions are indicated on the strainer head.

bypassed to the clean side of the filter; therefore foreign matter must not be allowed to accumulate in the filter housing.

d. Absorption type filter. The absorption type filter consists of a number of cellulose, waste, or wound yarn filter elements supported in a steel container. The steel container is partitioned so that oil entering the tank completely surrounds all filter elements. A pressure relief valve mounted in the partition is permanently set to maintain the correct pressure differential across the filter for proper clarification. Oil in excess of the set pressure (usually about 20 psi) is discharged through the valve and the filter outlet from which it returns to the sump tank for recirculation.

Filters of this type vary considerably in design and construction but are similar in operating principle. Some designs employ only two large filter elements, while others may have over twenty. The location of the partition and the position of the relief valve and the inlet and outlet openings also vary depending upon the make and model of the filter.

7B7. Lubricating oil clarifier. a. General. Clarification of the lubricating oil is accomplished by the Sharples centrifuge which also serves as the fuel oil purifier (Section 5C4). The machine is set up as a clarifier by installing

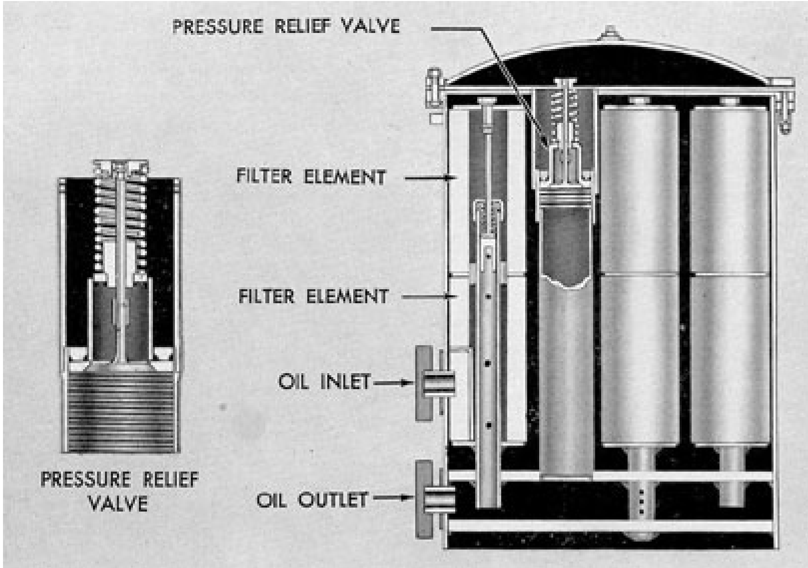


Figure 7-11. Absorption type filter.

a clarifier sleeve, or ring dam, on the top of the bowl, thus closing the outlet passage through which the water is discharged. The term clarifier is applied to the machine when it is set up to discharge a single liquid from which solid matter has been removed by centrifugal force. If the machine is set up to separate two liquids from solid matter and from each other (such as oil and water in a fuel oil purifier) it is called a separator. The machine is usually set up as a separator for fuel oil purification and as a clarifier for lubricating oil purification.

The lubricating oil purifier consists essentially of a rotor, or bowl, which rotates at high speeds. It has an opening in the bottom to allow the dirty lubricating oil to enter and two sets of openings to allow the oil and water or the water by itself to discharge. The bowl, or hollow rotor, of the centrifuge is connected by a coupling unit to a spindle which is suspended from a ball bearing assembly. The pulley of this bearing assembly is driven by an endless belt from an electric motor Tension on the belt is maintained by an idler pulley.

The lower end of the bowl is entered into a drag bushing mounted in the drag assembly. This is a flexibly mounted guide bushing. In side the bowl is a three-wing partition consisting of three flat plates equally spaced radially. The three-wing partition rotates with the bowl and its purpose is to force the liquid in the bowl to rotate at the same speed as the bowl. The liquid to be centrifuged is fed into the bottom of the bowl through the feed nozzle under pressure so that it jets into the bowl in a stream. For lubricating oil clarification the three-wing partition has a cone on the bottom against which the feed jet strikes to bring the liquid up to speed smoothly without making an emulsion. This cone is not necessary for fuel oil separation since fuel does not have the tendency to emulsify.

b. Operation. When a mixture of oil, water, and dirt stands undisturbed, gravity tends to effect a separation into an upper layer of oil,

mounted on the rear of the frame.

an intermediate layer of water, and a lower layer of the solid. When the mixture is placed in a rapidly revolving centrifugal bowl, the effect of gravity is negligible in comparison with that of centrifugal force, which acts at a right angle to the vertical axis of rotation of the bowl. The mixture tends to separate into a layer of solids against the periphery of the bowl, an intermediate layer of water, and a layer of oil on the inner surface of the water. The discharge holes of the bowl may be so arranged that water can be drawn off and discharged into the upper cover. The solids will deposit against the wall of the bowl, to be cleaned out when necessary or as operations permit.

If an oil contains no moisture, it need only be clarified, since the solids will deposit in the bowl, and the oil will discharge in a purified state. If, however, the oil contains some moisture, the continued feeding of wet oil to the bowl

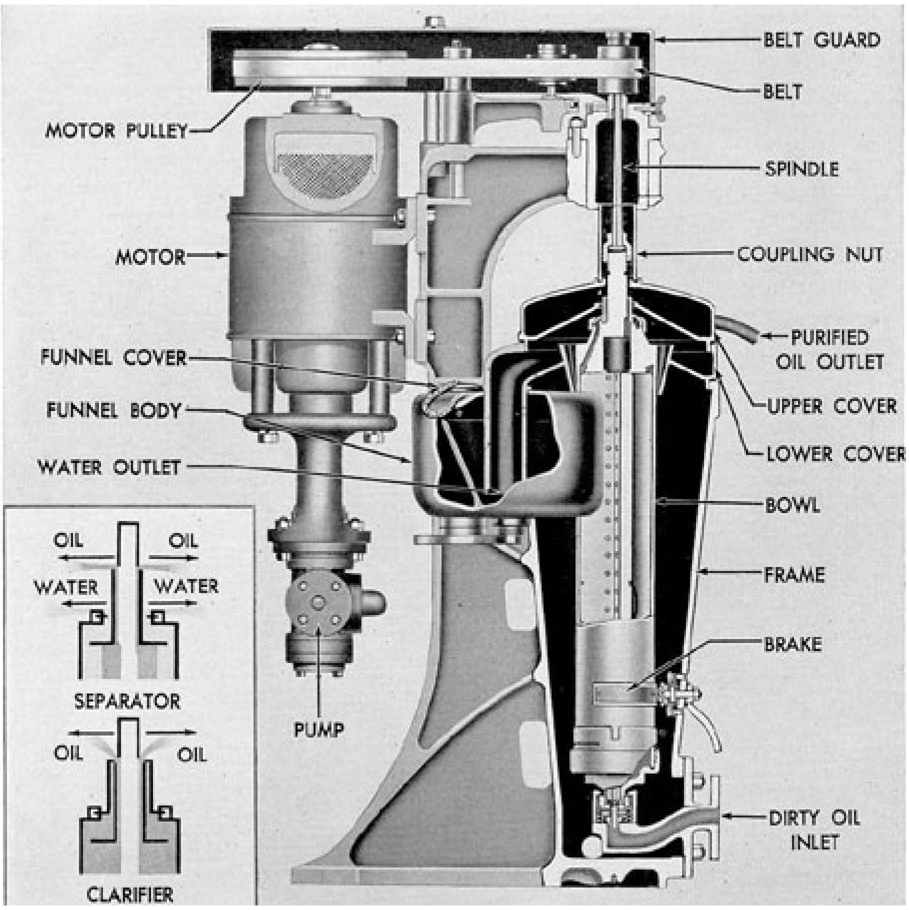


Figure 7-12. Cross section of Sharpies purifier.

results eventually in a bowl filled with water, and from that time through the purifier will result merely in less efficient purification.

on, the centrifuge is not accomplishing any separation of the water from the oil. Even before the bowl is completely filled with water, the presence of a layer of water in the bowl reduces the depth of the oil layer. As a result, the incoming oil passes through the bowl at a very high velocity. This higher velocity means that the liquid is under centrifugal force for a shorter time, and the separation of water from the oil is, therefore, not so complete as it would be if the bowl were without the water layer, or if the water layer were a shallow one. Because of this, the centrifuge should not be operated as a clarifier unless the oil contains very little or no water. A small amount of water can be satisfactorily accumulated, together with the solids, to be drained out when the bowl is stopped for cleaning, but if there is any appreciable amount of water in the oil, the bowl should be operated as a separator.

The length of time required to clarify lubricating oil is determined to a great extent by the viscosity of the oil. The more viscous the oil, the longer it takes to purify it to a given degree of purity. The use of a pressure in excess of that normally used to force a high-viscosity oil

Decreasing the viscosity of the oil by heating is therefore one of the most effective methods of facilitating purification.

The capacity rating of the centrifuge is based on the use of 2190 oil at 130 degrees F, which represents a viscosity of approximately 200 SSU. For good results no oil should be purified at a higher viscosity than this and other oils may need to be heated above 130 degrees F to reach 200 SSU. (See temperature table below.)

A reduction in the pressure at which the oil is forced into the centrifuge will increase the length of time the oil is under the influence of centrifugal force, and therefore will tend to Improve results. The effective output of the machine in any case will depend on viscosity, pressure, the size of the solid particles, the difference in specific gravity between the oil and the water, and the tendency of the oil to emulsify. If a used lubricating oil contains no water, but merely metallic particles, it may be cleaned at a higher rate (high input pressure). If the same oil contains a large percentage of water, and has a tendency to emulsify, the input pressure will necessarily have to be lower to obtain the required degree of purity.

TEMPERATURE TABLE					
Oil, Navy symbol	**Temperature* in degrees F**	**Oil, Navy symbol**	**Temperature* in degrees F**	**Oil, Navy symbol**	**Temperature* in degrees F**
1042	89	2135	116	5065	143
1047	102	2190	129	5150	190

1065	137	2250	142	5190	209
1080	151	3050	119	6135	192
1100	166	3065	135	7105	173
1120	179	3080	154	8190	128
1150	190	3100	163	9170	123
2075	92	3120	180	9250	140
2110	95	4065	140	9370	158
* Minimum temperature of oil to obtain viscosity of 200 SSU					

C. GENERAL MOTORS LUBRICATING SYSTEM

Oil for the GM lubricating system is circulated by the positive displacement attached lubricating oil pump driven through the camshaft drive gear train. This pump draws oil from the sump tank, passes it through a safety relief valve at the discharge side of the pump, then through a lubricating oil strainer and a cooler. From the cooler, the oil enters the engine's main lubricating oil manifold. After circulating through the various

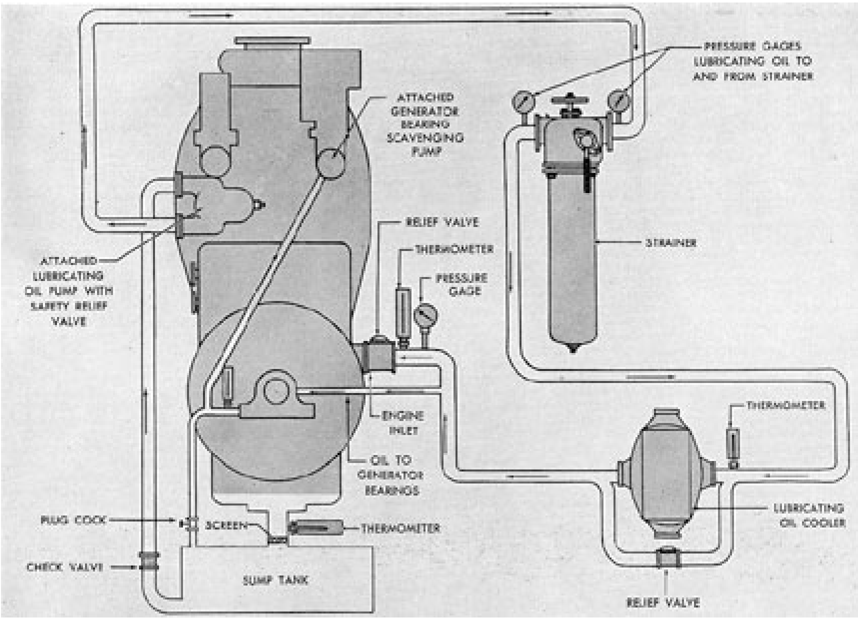


Figure 7-13. Lubricating oil system, GM.

passages in the engine, the oil drains into the engine oil pan and then back into the sump tank from which it is recirculated.

Oil for the generator bearings is taken from the lubricating oil piping between the cooler and the engine inlet. When the engine is running, the bearing inlet. A bypass line with a relief valve is provided to bypass the cooler when for any reason the cooler cannot handle the full flow volume. This bypass is also used when cooling of the oil is not required, as when starting an engine in cold weather.

drains are placed under a suction head by the attached generator bearing scavenging pump to prevent flooding of the main generator bearings. The plug cock in the gravity drain line leading from the generator bearings to the sump tank is closed and the oil is drawn from the bearing drains into the engine lubricating system, whence it drains into the oil pan and back to the sump tank. Before starting, or when flushing the engine with the detached lubricating oil service pump, the plug cock in the gravity drain line is opened, permitting the oil to drain directly into the sump tank.

Mercury type thermometers are located at the lubricating oil cooler inlet and at the engine Engines on SS 313 to 318 have a relief valve set at 80 psi. All other engines for the SS 313 Class have a 30-pound differential pressure relief valve. The 80-pound relief valve is not considered satisfactory for this service.

Duplex type pressure gages are provided in the system to register the oil pressure at the engine inlet and at the inlet and outlet of the lubricating oil strainer. The system is provided with a low-pressure alarm consisting of a pressurestat at the engine inlet which energizes a horn and light whenever the oil intake drops to 15 psi or less. Continuous reading type thermometers indicate the temperature of the oil drain from the engine and the generator bearings.

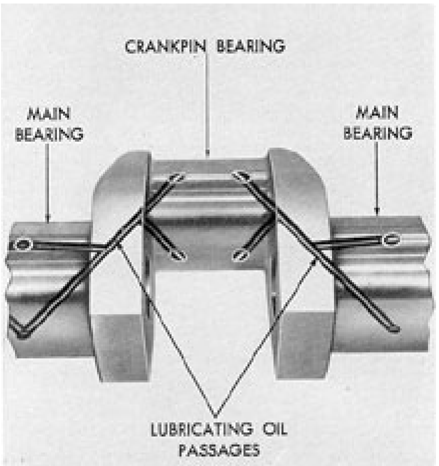


Figure 7-15. Crankshaft oil passages, GM.

7C2. Engine lubricating system. The lubricating oil enters the engine at a connection on the control side of the camshaft drive housing. The relief valve ahead of the inlet keeps the pressure of the oil at 40-50 psi. Any oil bypassed by the relief valve returns to the camshaft

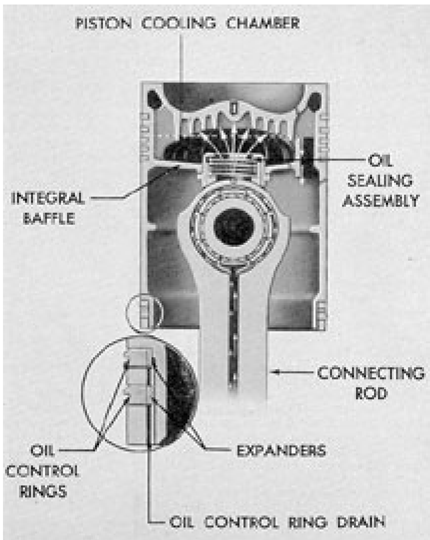


Figure 7-16. Piston and piston pin lubrication and cooling, GM.

chamber. The sealing assembly consists of a bronze oil seal saddle which rides on the machined top of the connecting rod and is held against the connecting rod by a spring. The heated oil overflows through two openings in the

drive housing from which it drains to the oil pan.

From the engine inlet connection, the oil flows to the main lubricating oil manifold which extends the length of the engine and is bolted to the bottom of the main bearing supports. The oil flows from the manifold up through drilled passages in the supports to each main bearing. The crankpin bearings are lubricated with oil that is received from the adjacent main bearings through oil passages in the crankshaft. A drilled passage in the connecting rod conducts this oil to the piston pin bearing and to the piston cooling chamber formed by an integral baffle under the piston crown. Lubricating oil under pressure flows from the top of the connecting rod, through an oil sealing assembly and into the cooling integral baffle and down to the oil pan from which it drains to the sump tank.

The lubricating oil for the camshaft drive gear train is supplied by branch lines from the main lubricating oil manifold. These branch lines conduct oil to the lubricating oil distributor block on each side of the camshaft drive housing. From each of the distributor blocks a pipe supplies oil to each camshaft drive gear bearing. The drilled camshafts are supplied with oil through passages in the camshaft gear hubs and the camshaft drive sleeves. The oil then passes through the hollow camshafts and supplies the camshaft bearings by radial holes through the camshaft bearing journals. Oil for lubricating the rocker levers and cam rollers flows through a tube from the camshaft bearing cap at each

Figure 7-14. ENGINE LUBRICATING SYSTEM, GM.

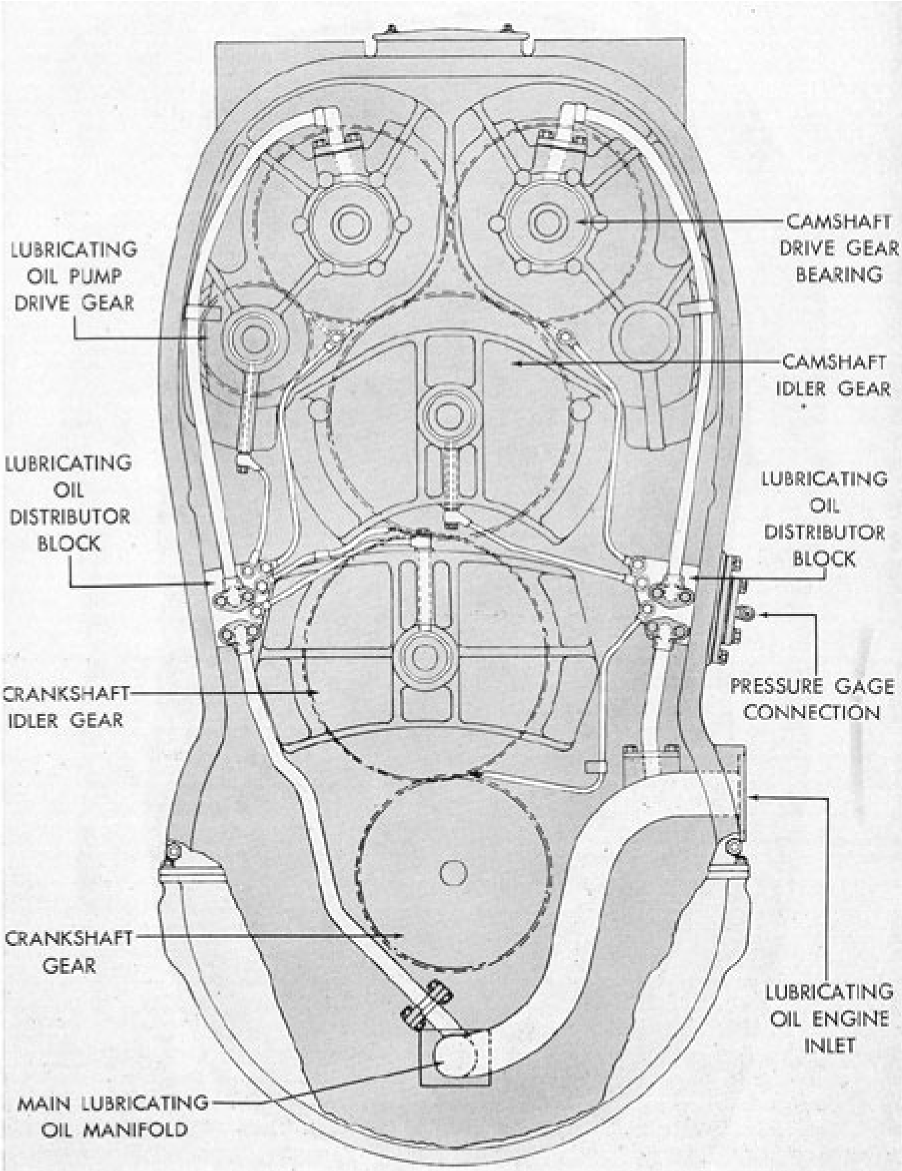


Figure 7-17. Camshaft drive lubrication, GM.

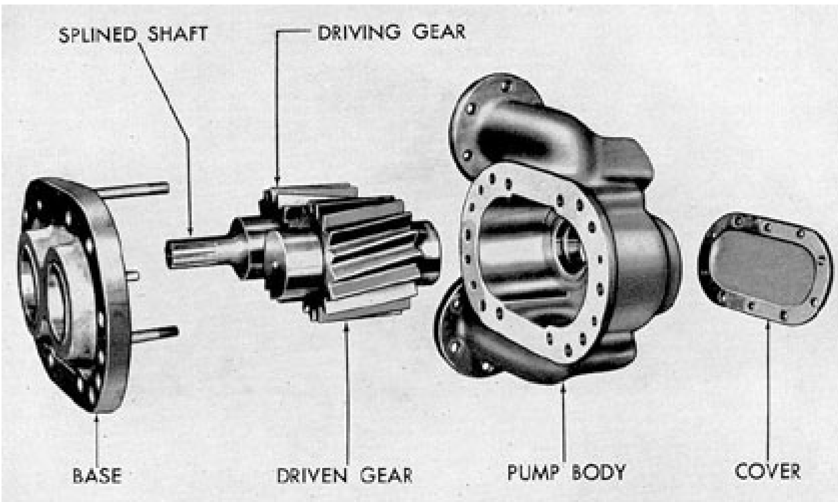


Figure 7-18. Attached lubricating oil pump, GM.

engine cylinder. This oil also lubricates the valve assemblies. The oil flows from the end of the

Oil from the cylinder heads and valve operating gear drains through the micrometer link passages to the control shaft

camshafts down the camshaft drain tubes to the engine oil pan.

Each rocker lever assembly is lubricated with oil that is received from an adjacent camshaft bearing. The oil flows from the top of the camshaft bearing through a tube to the plate connection that is fastened to one end of the rocker lever shaft. From this connection, the oil flows through drilled passages in the rocker lever shaft to the three bearings in the rocker lever hubs.

A drilled passage in each of the rocker lever forgings conducts the lubricating oil from a hole in the hub bushing to the camshaft end of the lever. The rocker lever motion permits oil to flow intermittently under pressure from the hole in the shaft, through one hole in the bushing and rocker lever to the cam roller. The bearing in each of the cam rollers receives oil through drilled holes in the roller pin and in the bearing bushings.

compartment and thence through tubes to the crankcase.

A manifold, bolted to the blower end of the main lubricating oil manifold, supplies the lubricating oil for the blower gears and bearings, and the accessory drive gears and bearings. The manifold carries oil to the blower rear end plate, to the blower front end plate, and the accessory drive housing. Steel tubing cast into the ribs of the end plates and the housing carries the lubricating oil to the blower drive gear bearings and to the gear bearings in the accessory drive. Excess oil drains through the lower blower housing into the oil pan.

The engine main lubricating system also furnishes lubricating oil to the overspeed governor. When the engine speed exceeds a predetermined limit, this lubricating oil is pumped under pressure to the overspeed injector lock on each cylinder head and prevents the injector from operating.

7C3. Attached lubricating oil pump. The lubricating oil pump used for the GM pressure lubricating system is mounted on the camshaft drive housing cover and is a positive displacement helical spur gear type pump.

The lubricating oil pump body and body base are bronze castings. The spur gears and shafts are integral forgings and the shafts revolve on bronze bushings which are pressed into the housing and cover, the cover the oil pump drive gear. The camshaft drive gear meshes with the oil pump drive gear to operate the pump.

A spring-loaded pressure safety valve is frequently attached to the lubricating oil pump to prevent the lubricating oil pressure in the system from exceeding a safe operating pressure, The spring pressure is adjusted with a regulating screw which is enclosed by a cover on the valve head. The regulating screw is adjusted so

being used to close the outside of the body.

The spur gear that does the driving has its shaft extended and splined to fit into the hub of that the valve opens when the lubricating oil discharged from the pump reaches a gage pressure of 90 pounds. The bypassed oil is returned to the suction side of the pump.

D. FAIRBANKS-MORSE LUBRICATING SYSTEM

7D1. General description. The lubricating oil system outside the engine in an F-M installation is similar to the GM system described in section 7C1. Lubricating oil is drawn from the sump tank by the attached positive displacement gear pump mounted on a plate at the control end of the engine and driven by the lower crankshaft through gears and a flexible coupling. From the attached pump, the oil is passed through a strainer and two coolers. Leaving the coolers, the oil is piped to the engine where it enters the lower lubricating oil header through an inlet flange. A pipe connection ahead of the engine inlet supplies lubricating oil to the generator bearings. After circulating through the engine, the lubricating oil drains into the engine oil pan and back to the sump tank for recirculation. Oil from the generator bearings returns directly to the sump tank.

An electrical resistance thermometer bulb is installed in the lubricating oil line between the pump and the strainer. Temperature of the oil at this point is indicated on a gage mounted on the engine gage board. This gage board also supports a duplex type pressure gage which indicates the a low-pressure alarm signal whenever the lubricating oil pressure drops to 15 psi or less.

The coolers may be bypassed if the operating conditions warrant. In this bypass line a spring-loaded pressure relief valve, set to open at a pressure of 45 psi, is installed. This bypass and relief valve insures circulation of the oil should its viscosity be such as to cause a restricted flow through the coolers.

7D2. Engine lubricating system. Entering the lower lubricating oil header from the inlet near the control end, the oil flows through the lower header toward the blower end. There, a vertical pipe carries the oil to the upper header. Both headers extend longitudinally the entire length of the engine.

Through supply pipes from both lower and upper headers, oil is forced to each main bearing, and thence, through tubes swedged into the crankshaft, to each crankpin bearing. From each crankpin bearing, oil passes through the drilled passage in the connecting rod to the piston pin bearings and to the piston oil cooling pockets.

The surfaces between the thrust shells and the crankshaft flanges are lubricated through openings in the thrust bearing shells.

pressure of the oil in the line between the strainer and the attached pump and in the line ahead of the engine inlet. Also near the lubricating oil inlet to the engine is a mercury bulb thermometer and a pressure static contact maker which closes a circuit to energize

The cooling oil from each lower piston is discharged through the lower piston cooling oil outlet into the oil pan. Oil from each upper

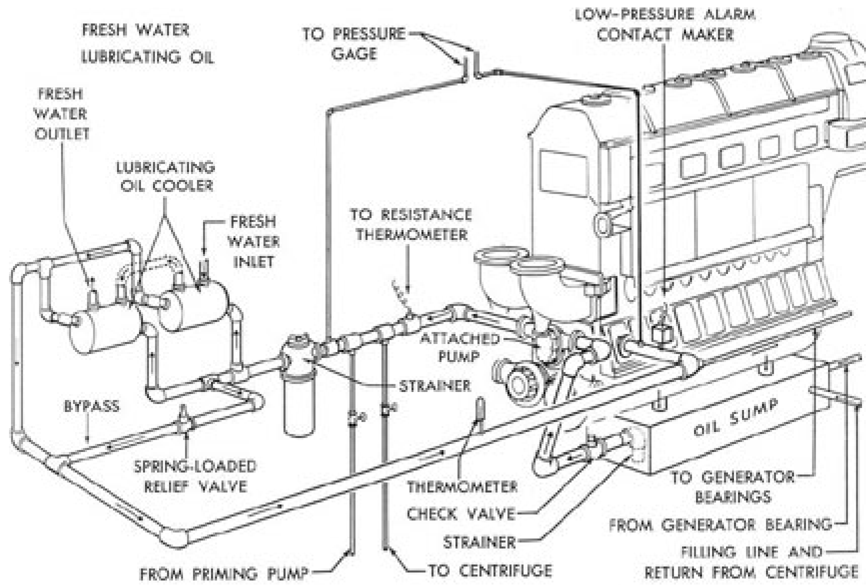


Figure 7-19. Lubricating system, F-M.

piston is discharged through the upper piston cooling outlet into the compartment around the tipper ends of the cylinders. This oil can drain either to the blower or to the control end of the engine and then down to the oil pan.

The two camshafts receive lubrication from the upper oil header. Oil enters the hollow camshafts through the camshaft bearing at the control end of the engine, and small openings at each bearing journal allow oil to reach the camshaft bearing surfaces. An opening in the end of each camshaft, and excess oil from the No. 1 main bearing, supply oil to the timing chain at

oil which is carried by the air to lubricate the air start check valves.

The drive bushings of the pump flexible drive (on the control end of the lower crankshaft) receive lubrication through an opening in the lower crankshaft from the control end main bearing.

Oil from the upper engine compartment enters the injection pump housing and lubricates the tappet assembly. The excess oil is drained through leads to a return header which conducts the oil to the control end compartment.

The blower drive gears are lubricated by sprays of oil from special nozzles located on each

the control end of the engine. The oil spray from the timing chain provides lubrication for the bearings of the idler sprockets, for the control mechanisms, drive gears, and bearings of the governor, and for the water, fuel, and lubricating oil pumps. Spray from the timing chain also lubricates the air start distributor and the air start control valve located in the lower part of the control end compartment. The air start distributor valves admit a minute quantity of

side of the centerline of the engine. These nozzles are attached to the oil piping connecting the lower and upper oil headers.

The blower flexible drive gear is lubricated through openings in the drive spider. Oil is brought to these openings from the nearest main bearing by means of drilled passages in the upper crankshaft.

Figure 7-20. ENGINE LUBRICATING OIL CIRCULATION. F-M.

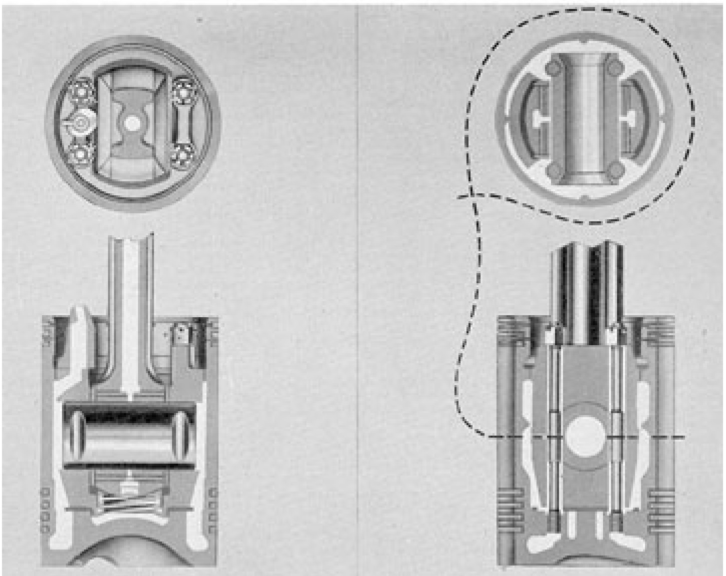


Figure 7-21. Sectional views of F-M piston showing oil passages.

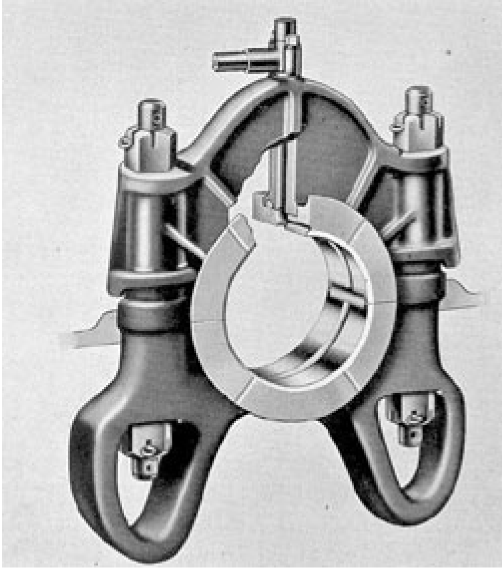


Figure 7-22. Thrust bearing oil passages, F-M.

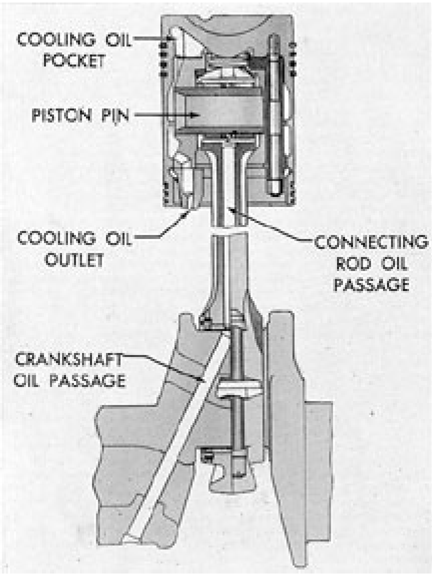


Figure 7-23. Piston assembly oil passages, F-M.

The inner and outer blower impeller bearings are lubricated by branches and oil tubes from the upper oil header.

The vertical drive gears and pinions are lubricated by a spray of oil from nozzles connected to the upper and lower oil headers by

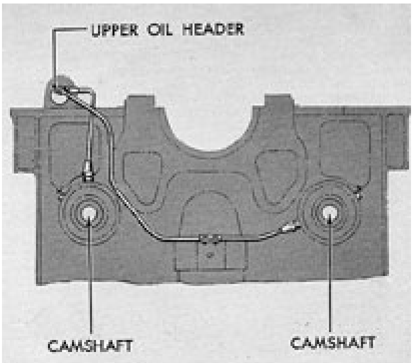


Figure 7-24. Oil supply to camshafts, F-M.

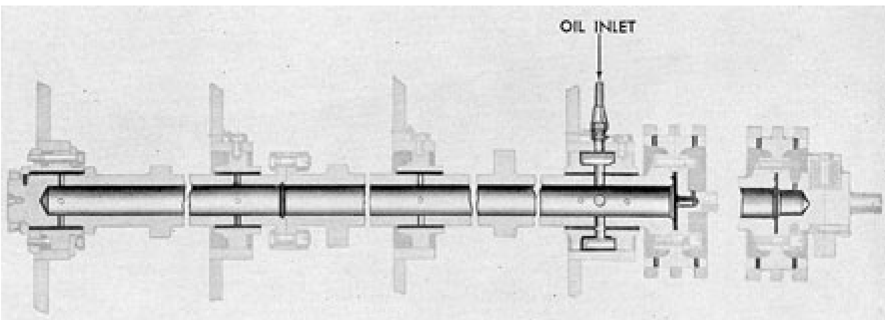


Figure 7-25. Camshaft and camshaft bearing lubrication, F-M.

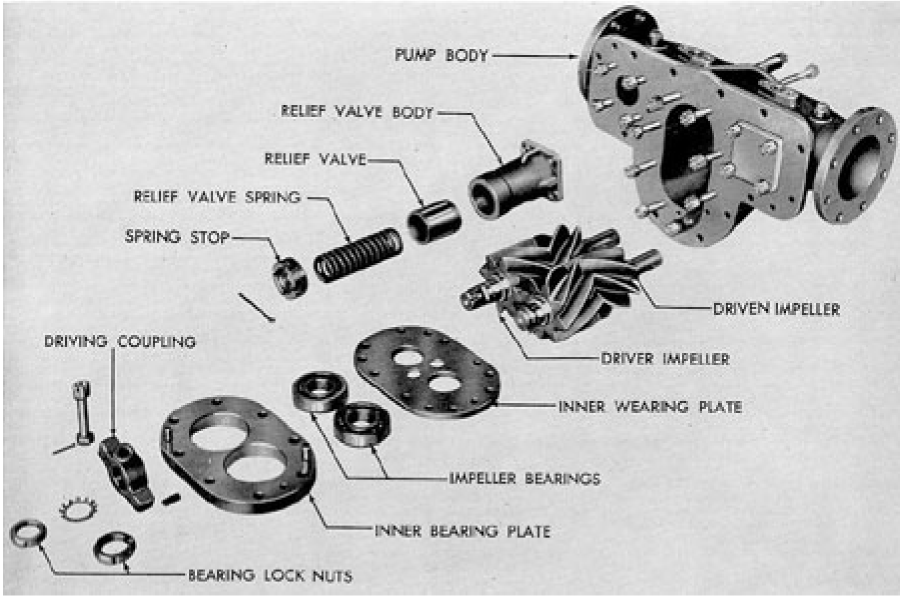


Figure 7-26. Drive end of attached lubricating oil pump, F-M.

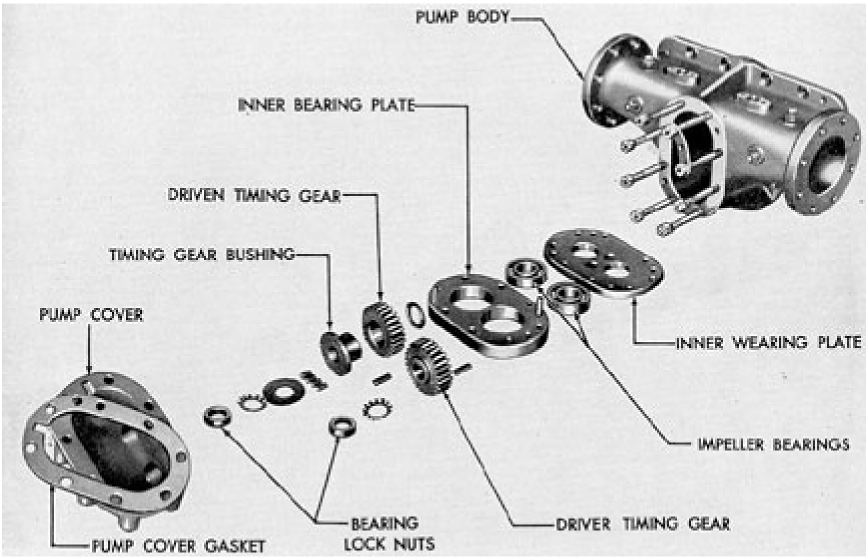


Figure 7-27. Gear end of attached lubricating oil pump, F-M.

tubes. Other tubes supply oil to the vertical drive pinion shaft roller and thrust bearings.

Having performed its various functions. the lubricating oil drips down into the oil pan below the lower crankshaft. From the oil pan, the oil drains into the sump tank from which it is recirculated.

7D3. Attacked lubricating oil pump. The attached lubricating oil pump used in the F-M spring-loaded relief valve, set at 50 to 60 pounds pressure, is located in the discharge opening of the pump body.

The impeller shafts are supported on bearings pressed into the bearing plates. The inner and outer wearing plates are located against the inner surfaces of the bearing plates and provide a clearance of 0.002 to 0.004 inch between the wearing plates and the impellers. The longitudinal clearance between the impellers

pressure lubricating system is mounted on a pump mounting plate on the control end of the engine. It is driven, through gears and a flexible coupling, by the lower crankshaft. The pump is a positive displacement herringbone (impeller) gear type and consists essentially of a pump body, a driver and driven timing gear, a driver and driven impeller, inner and outer wearing plates, and inner and outer bearing plates. A and the pump body is 0.003 to 0.0045 inch. The driver timing gear is pressed on the end of the driver impeller shaft and the driven timing gear is pressed on the end of the driven impeller. Both timing gears are enclosed within the pump cover. The drive coupling is attached to the end of the driver impeller outside the outer bearing plate.

Figure 7-14. ENGINE LUBRICATING SYSTEM, GM.

Sub Diesel Home Page

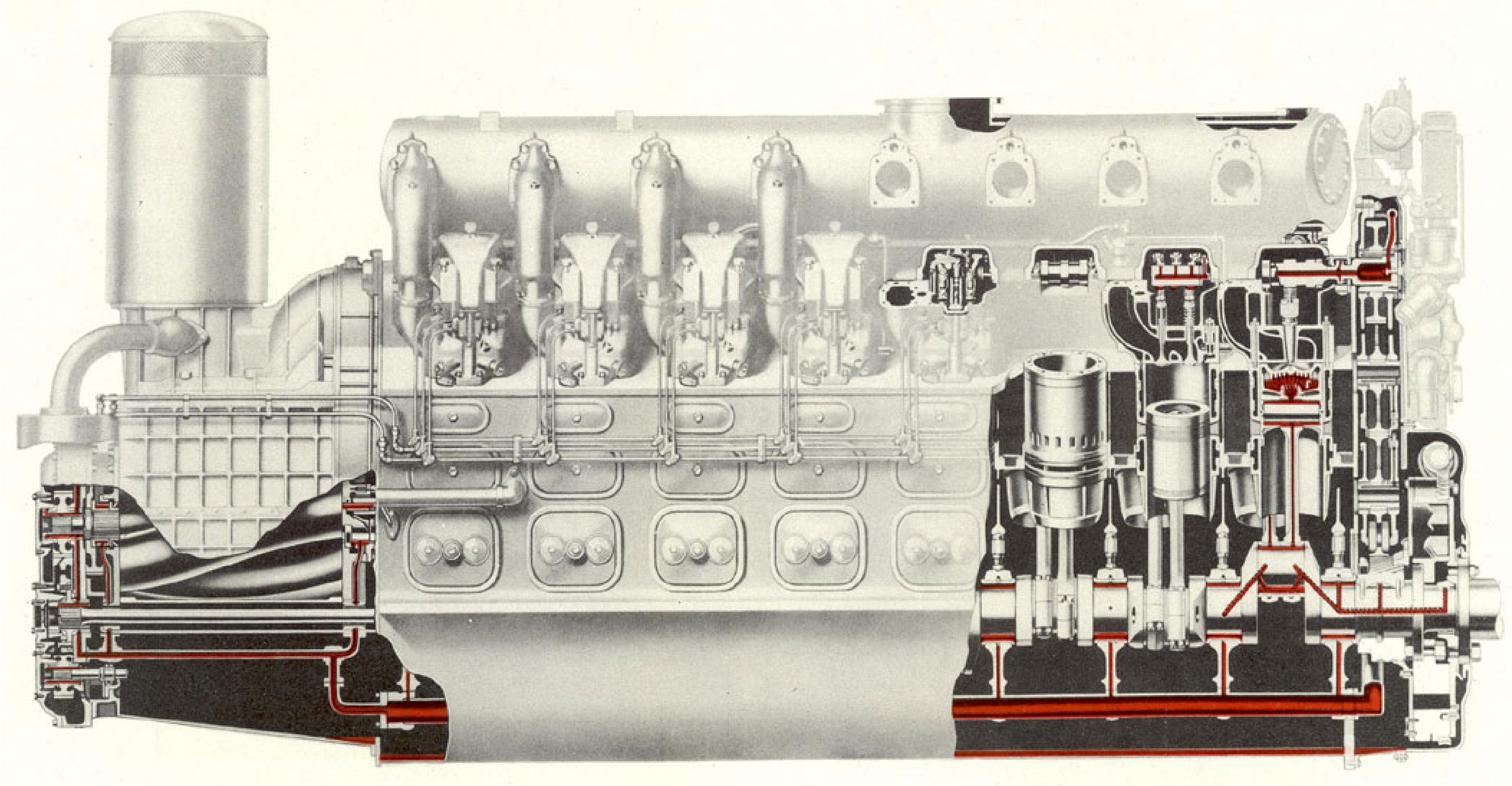
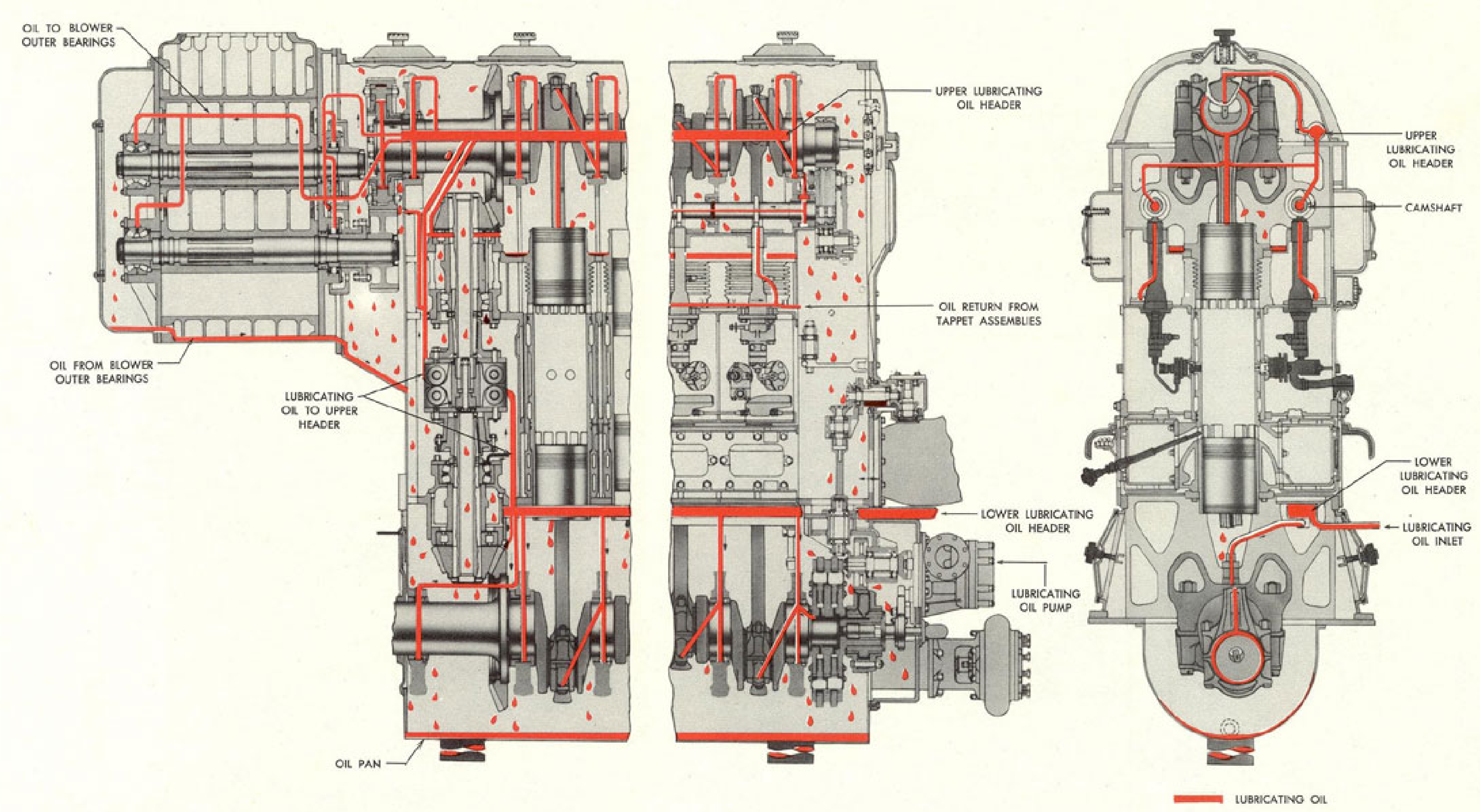


Figure 7-20. ENGINE LUBRICATING OIL CIRCULATION. F-M.

Sub Diesel Home Page


OIL TO BLOWER OUTER BEARINGS
OIL FROM BLOWER OUTER BEARINGS
LUBRICATING OIL TO UPPER HEADER
OIL PAN
UPPER LUBRICATING OIL HEADER
OIL RETURN FROM TAPPET ASSEMBLIES
LOWER LUBRICATING OIL HEADER
LUBRICATING OIL PUMP
UPPER LUBRICATING OIL HEADER
CAMSHAFT
LOWER LUBRICATING OIL HEADER
LUBRICATING OIL INLET
LUBRICATING OIL

8
COOLING SYSTEMS

A. GENERAL

8A1. Purpose of cooling systems. The high-speed high-output diesel engines of today are strictly limited as to the maximum temperature at which they can safely operate. To maintain the temperature below the maximum allowable limit, various types of cooling systems are used. The thermal efficiency of an engine would be greatly improved if it were not necessary to provide a cooling system. The cooling system losses, together with the loss of heat during the combustion, working, and exhaust periods, cut down the thermal efficiency of the engine to a relatively small percentage. Shown below are the percentages of useful work and various losses obtained from the combustion of a fuel oil in a diesel cylinder:

To useful work (brake thermal efficiency)	30-35 percent
To exhaust gases	30-35 percent
To cooling water and friction	30-35 percent
Radiation, lube oil, and so forth	0- 5 percent

There are three practical reasons for cooling an engine:

1. To maintain a lubricating oil film on pistons, cylinder walls, result in a variation of clearances between the moving parts. Under normal operating conditions these clearances are very small and any variation in dimension of the moving parts may cause insufficient clearances and subsequent inadequate lubrication, increased friction, and possible seizure.

3. To retain the strength of the metals used. High temperatures change the strength and physical properties of the various ferrous metals used in an engine. For example, if a cylinder head is subjected to high temperatures without being cooled, the tensile strength of the metal is reduced, resulting in possible fracture. This high temperature also causes excessive expansion of the metal which may result in shearing of the cylinder bolts.

Cylinder heads, cylinder jackets, cylinder liners, exhaust headers, valves, and exhaust elbows usually are cooled by water. Pistons may be cooled either by water or oil. In present fleet type submarine installations, the pistons are cooled by lubricating oil which is in turn cooled by engine cooling water. It is important to keep all parts of the engine at as nearly the same temperature as possible. This can be accomplished to some

and other moving parts as explained in Chapter 7. This oil film must be maintained to insure adequate lubrication. The formation of an oil film depends in large degree on the viscosity of the oil. If the engine cooling system did not keep the engine temperature at a value that would insure the formation of an oil film, insufficient lubrication and consequent excessive engine wear would result. If the engine is kept too cool, condensation takes places in the lube oil and forms acids and sludge.

2. To avoid too great a variation in the dimensions of the engine parts. Great differences between operating temperatures at varying loads cause excessive changes in the dimensions of the moving parts. These excessive changes also occur when there are large differences between the cold and operating temperatures of the parts. These changes in dimensions extent by engine design. For instance, the water jacket should cover the entire length of the piston stroke to avoid possible unequal expansion of various sections of the cylinder and cylinder liner.

It requires time to conduct heat through any substance, therefore the thicker the metal, the slower the conduction. This is one of the reasons the size of cylinders in diesel engines is limited, because the larger the cylinder, the thicker the material necessary for liners and cylinder heads in order to withstand the pressures of combustion. Thicker metals cause the inside surfaces to run hotter, because the heat is not conducted so rapidly to the cooling water.

8A2. Operation of a cooling system. One of the principal factors affecting the proper

cooling of an engine is the rate of flow of water through the system. The more rapid the rate of flow, the less danger there is of scale deposits, and the formation of hot spots, since the high water velocity has a scouring effect upon the metal surfaces of the jackets, and the heat is carried away more quickly. When the velocity of the circulating water is slower, the discharge temperature is higher and more heat per gallon of water circulated is carried away. When the circulation is speeded up, each gallon of water carries

8A3. Types of cooling systems. Two types of cooling systems are in common use, the open system and the closed system. In the open system the engine is cooled directly by salt water. In the closed system the engine is cooled by fresh water and the fresh water is then cooled by salt water. The closed type of cooling system is in common use today in all modern medium- and high-speed diesel engines.

The open type of cooling system has many disadvantages, the most important being the exposure of

away less heat and the discharge temperature of the water drops, resulting in a relatively cool running engine.

The temperature of the engine can be controlled by the discharge temperature of the cooling water. This can be done in two ways, depending upon the arrangement of the piping and the type of pump used. A common and simple method is to control the amount of water pumped, by means of a throttling valve in the pump discharge to the engine cooling system. The water can then be made to pass more slowly through the engine and be discharged at a higher temperature, or to pass more rap idly at a lower temperature. If the pump is driven separately by an electric motor, the same effect can be obtained by slowing down or speeding up the pump. The other method to control the temperature is to bypass some of the warm discharged water around the cooler and directly to the suction side of the pump. This method gives a more uniform temperature throughout the cooling system and keeps the passage of water at a higher velocity.

In all modern engines, the latter method is used and accomplished automatically by means of a temperature regulator. These regulators may be set to give any desired temperature at the engine outlet. They are used not only to regulate the fresh water but also to regulate indirectly the temperature of the lubricating oil leaving the lubricating oil cooler.

the engine to scale formation, marine growth and dirt deposits in the piping, and fluctuating sea water temperature. Scale or deposits not only restrict water flow in the engine water passages but also act as a blanket and hinder heat transfer to the cooling water. This prevents adequate cooling of engine parts which may result in serious difficulties.

8A4. Open type cooling systems. The term open system is used because salt water is drawn directly from sea, passed through the system, and then discharged overboard.

In a typical system the salt water is drawn through sea valves and a strainer by a centrifugal pump and then discharged through the lubricating oil heat exchanger or cooler where it cools the lubricating oil. The water then passes to the cylinder liner jackets, exhaust manifold jackets, exhaust uptake jackets, the inboard exhaust valve, overboard sea valves, and to the outboard exhaust valve jackets and sprays. Part of the water may be piped to the fuel compensating water system. The remaining water passes through the muffler jackets and then overboard.

The open type of cooling system is used only on engines in the older types of submarines, particularly the O, R, and S classes. All of the later fleet type submarine engines are designed with cooling systems of the closed type.

8A5. Closed type cooling systems. Closed type, or fresh water cooling systems consist basically of two entirely separate

This is possible because the fresh water that is passed through the regulator and fresh water cooler is used as the cooling agent in the lubricating oil cooler. This permits the maximum amount of controllability of fresh water and lubricating oil temperatures with the use of the minimum amount of equipment.

systems-the fresh water cooling system and the salt water cooling system. In the fresh water cooling system the same fresh water is reused continuously for cooling the engine. The water is circulated throughout the engine cooling spaces

by an attached circulating fresh water pump. The water is then led to a fresh water cooler, where it is cooled by the salt water of the salt water cooling system. After it leaves the cooler, the fresh water may or may not, depending on the installation, go through the lubricating oil cooler to act as cooling agent for the lubricating oil. The water then returns to the fresh water pump, completing the circuit.

Fresh water temperature is usually regulated by means of an automatic regulating valve which maintains the fresh water temperature at any desired value by bypassing the necessary amount of water around the fresh water cooler.

An expansion tank is provided which aids in keeping the fresh water system filled at all times by keeping available a ready supply of water. A vent usually is provided in the high point of the line to keep the system free of air, thereby preventing the water pump from becoming air bound. The expansion tank also is equipped with a gage glass by which the level of water in the tank may be constantly

the salt water circulating water overboard discharge.

On generator type engines the attached salt water pump furnishes salt water to the generator air coolers and returns the water to the overboard discharge. Throttling valves frequently are placed in lines to the fresh water cooler and generator air coolers to control the flow of water through these heat exchangers.

Thermometers and pressure indicators are placed in the system at various places. Salt water temperatures should not exceed 122 degrees F. Fresh water temperatures should be between 140 degrees F and 180 degrees F, with a minimum of 140 degrees F at the engine inlet. Outlet fresh water temperatures should be between 160 degrees F and 180 degrees F. Cooling water temperatures should not be allowed to drop below 140 degrees F, otherwise excessive engine wear and corrosion may result if the temperature drops, below the dewpoint.

8A6. Detached fresh water circulating pumps. Earlier General

observed. If the level of water in the tank becomes too low, the system may be replenished from the ship's fresh water service system through a make-up line into the suction side of the attached fresh water pump. Any large rapid fluctuation in the level of water in the expansion tank signifies some type of leak into or out of the fresh water system. It usually indicates a cracked cylinder liner.

The salt water section of the closed type cooling system consists of an attached salt water pump, usually similar to the fresh water pump which draws salt water from sea through a sea chest, a stop and check valve, and a strainer, and discharges it through the fresh water cooler and then overboard. The overboard discharge performs varying functions, depending upon the individual installation. Normally it is used to cool the outboard exhaust valve, outboard exhaust piping, and muffler. The ship's compensating water and header box discharge lines also receive their water from Motors and Fairbanks-Morse models (the GM 16-248 and the F-M 38D 8 and the 9-cylinder 38D 8 1/8) were equipped only with attached fresh water pumps. This design made it impossible for fresh water to be circulated in the engine for cooling purposes after the engine had been stopped. During normal operations in peacetime this is not too great a disadvantage because before stopping, the engine can be idled until it is properly cooled. During the war, however, emergency dives were a common occurrence and lack of a detached pump resulted in very high engine and engine room temperatures immediately after diving. This was not particularly good for the engine and imposed a hardship on engine room crews, especially in tropical climates. This condition resulted in the installation in all new submarines of detached fresh water pumps for circulation of the water after the engine has been stopped. An authorized alteration provides for the same installation in older fleet type submarines.

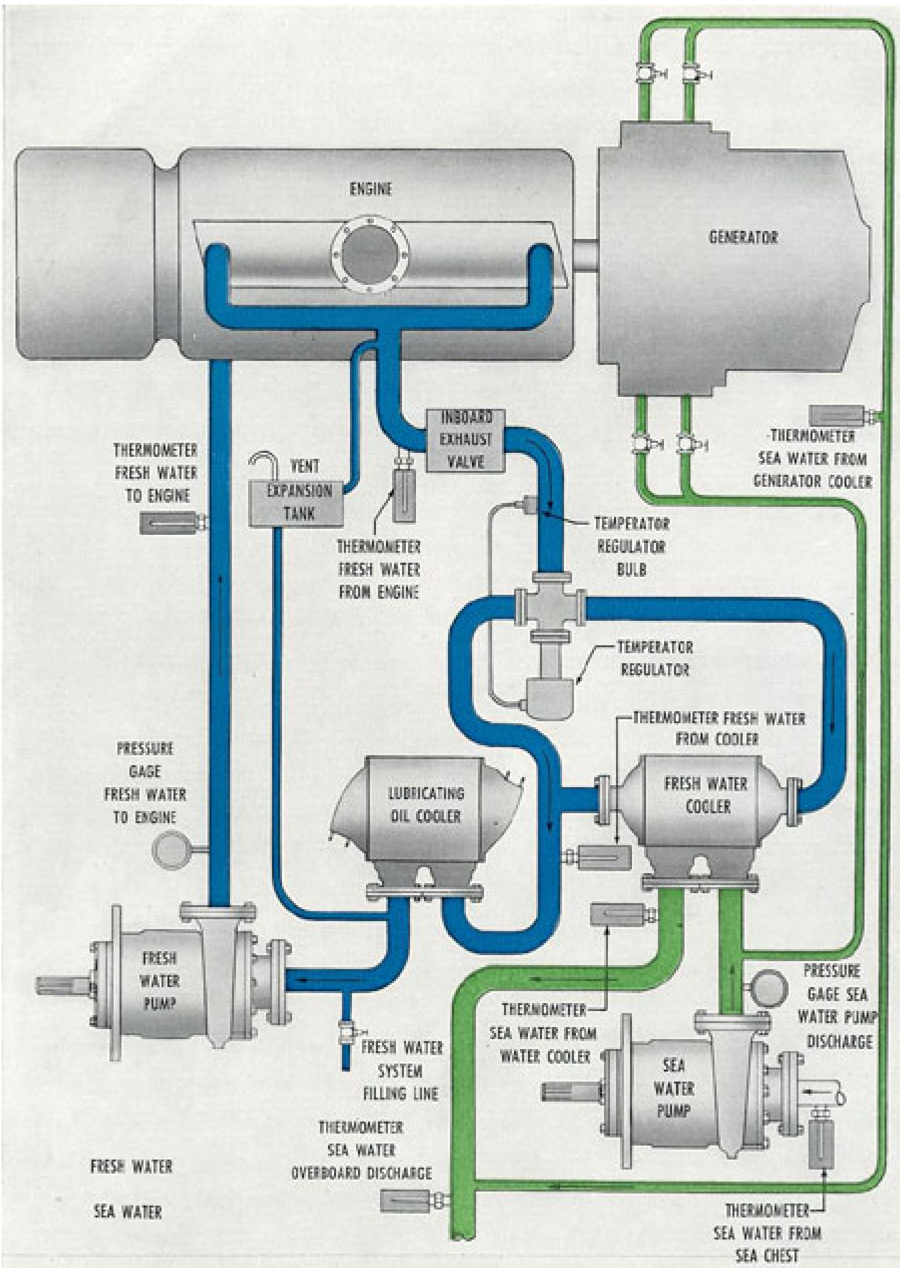


Figure 8-1. Typical fresh and salt water cooling systems.

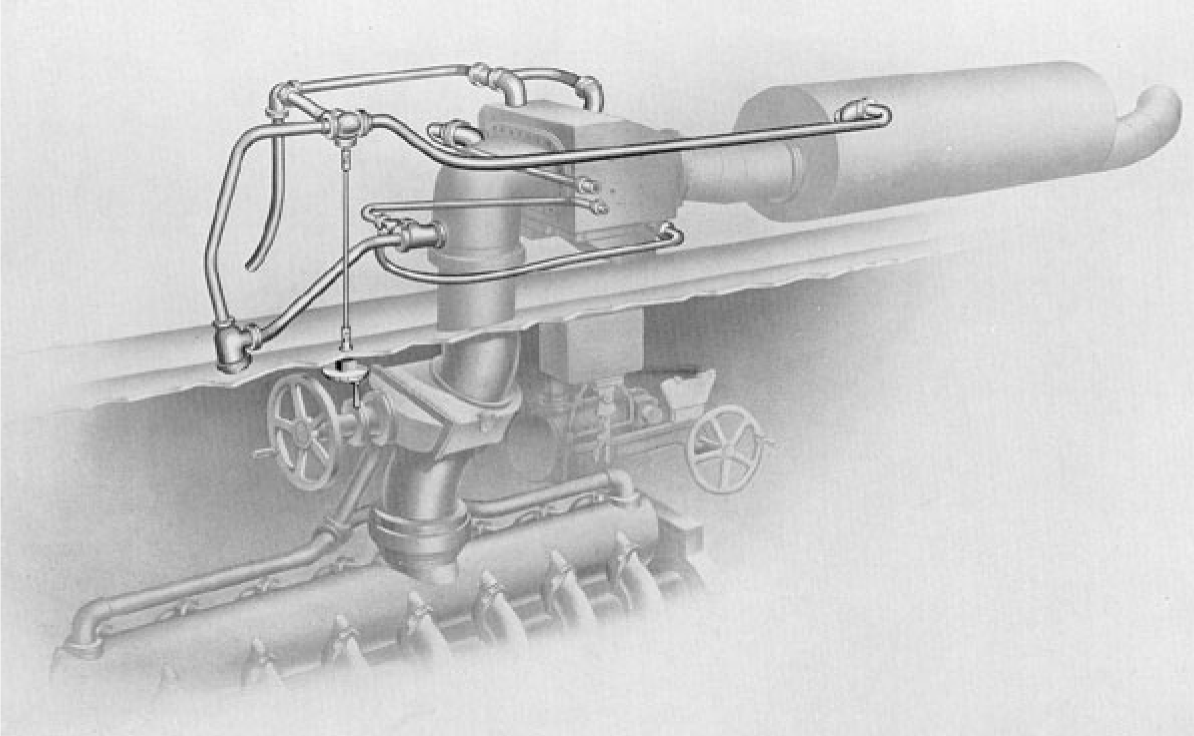


Figure 8-2. Salt water cooling system in superstructure.

8A7. Fresh water coolers. The engine water is cooled in a Harrison type heat exchanger or cooler, similar to the cooler used in the lubricating oil system. Although coolers used on various installations may differ in appearance and possibly to some extent in interior design, their operating principle is identical.

The cooler consists of a tube nest containing a number of oblong tubes fastened to a header plate at each end to form a core assembly. This assembly is attached to the cooler casing. The oblong tubes are baffled to form a winding passage for the liquid to be cooled. The liquid is cooled as it passes through the tubes, by the cold salt water (fresh water in lubricating oil coolers) which enters the casing, flows between the tubes and is discharged through the salt water outlet. The cooler is cooled enters the cooler at a higher pressure than the cooling agent. Thus, in a fresh water cooler the pressure of fresh water should, if possible, be greater than the pressure of salt water, so that in case of leaks, the fresh water will leak into the salt water, a more desirable condition than leakage of the salt water into the fresh water system. This is also true in a lubricating oil cooler wherein the pressure of the lubricating oil is found to be greater than that of the fresh water. This prevents water from getting into the engine lubricating oil if cooler leaks develop.

8A8. Temperature regulator. The fresh water in the engine is maintained at a uniform temperature by the temperature regulator which controls the amount of fresh water flowing through the fresh water cooler and by bypassing the remainder of the water around the cooler.

equipped with zinc plates in the sea water inlet and outlet passages and at the bottom of the cooler. These zincs centralize the electrolysis present in all submarine salt water systems. Their presence causes electrolytic action to eat away and disintegrate the zincs rather than the material of which the cooler tubes are made. This reduces to a minimum the number of cooler leaks to be expected in submarine installations. Zincs should be examined every 30 days or oftener where experience indicates the necessity. At each inspection they must be scraped clean. If this is not done, the efficiency of the zincs may become negligible and the electrolytic action will work on the tubes. When more than 50 percent of the zinc has been eaten away, the zinc should be renewed.

Coolers should be cleaned as frequently as found necessary to provide an unrestricted flow of water. In certain types of climate and service, deposits form more rapidly than in others. Heavy deposits cause an objectionable increase in pressure drop through the cooler and a consequent decrease in the cooling effect. Chemical cleanings at regular intervals in accordance with approved instructions will insure maximum operating efficiency at all times. Wires or prods which would damage the internal structure of the tubes must not be used in the cleaning operation. It is a universal rule that where the installation permits, the liquid to be

When the fresh water temperature is higher than the temperature for which the regulator is adjusted, the regulator valve is actuated to increase the flow of fresh water through the fresh water cooler and decrease the flow through the bypass. When the engine water temperature is lower than the temperature for which the

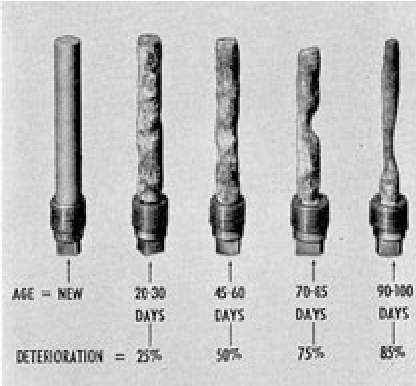


Figure 8-3. Salt water corrosion of zincs.

regulator is adjusted, the valve decreases the flow of fresh water through the fresh water cooler and increases the flow through the bypass.

The temperature regulator consists of a valve and a thermostatic control unit which is mounted on the valve. The thermostatic control, unit consists of two parts, the temperature control element and the control assembly.

The temperature control element consists of a bellows connected by a flexible armored tube to a bulb mounted in the engine cooling water discharge line. The temperature control element is essentially two sealed chambers. One is formed by the bellows and cap which are sealed together at the bottom. The other chamber is in the bulb. The entire system (except for a small space at the top of the bulb) is

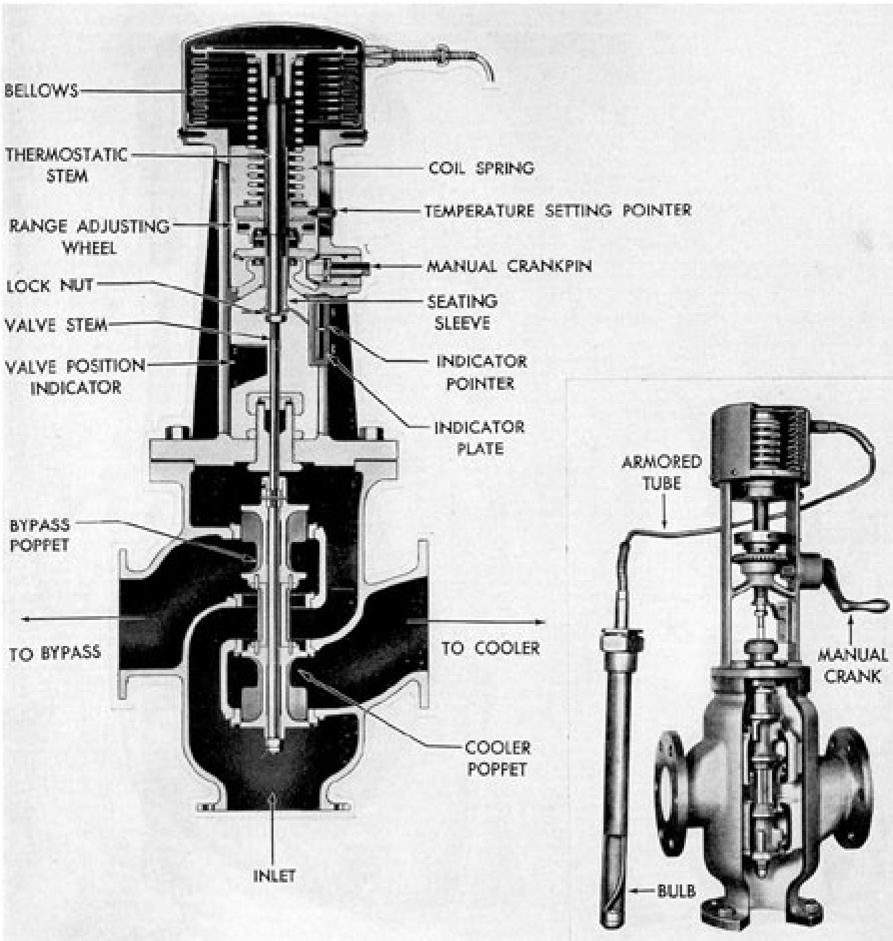


Figure 8-4. Fulton-Sylphon temperature regulator.

filled with a mixture of ether and alcohol which vaporizes at a low temperature. When the bulb is heated, the liquid vaporizes and increases the pressure within the adjusting wheel located under the spring seat. A pointer attached to the spring seat indicates the temperature setting on a scale attached to the regulator frame.

bulb. This forces the liquid out of the bulb and through the tube to move the bellows down and operate the valve.

The control assembly consists of a spring-loaded mechanical linkage which connects the temperature control element to the valve stem. The coil spring in the control assembly provides the force necessary to balance the force of the vapor pressure in the temperature control element.

Thus, the downward force of the temperature control element is balanced at any point by the upward force of the spring. This permits setting the valve to hold the temperature of. the engine cooling water within the allowed limits.

The regulator operates only within the temperature range marked on the name plate, and may be adjusted for any temperature within this range. The setting is controlled by the range

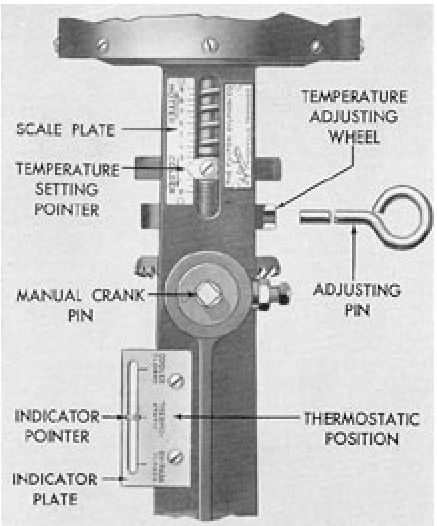


Figure 8-5. Thermostatic control unit.

The scale is graduated from 0 to 9, representing the total operating range of the regulator.

The temperature regulator can be controlled manually by turning the manual crankpin projecting from the side of the frame. This operates the regulating valve spring through a pair of beveled gears and a threaded sleeve. A pointer, attached to the threaded sleeve, indicates the valve position. When the pointer is in the BYPASS CLOSED position, the valve is set to allow all of the fresh water to be pumped directly through the water cooler. When the pointer is in the THERMOSTATIC position, it indicates that the unit is controlled completely by the automatic system as described. When the pointer is in the COOLER CLOSED position, it indicates that all of the fresh water is being bypassed around the water cooler. For automatic operation the pointer must be set at the THERMOSTATIC position.

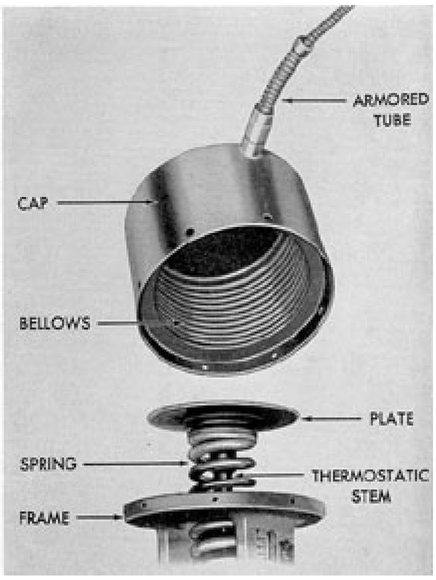


Figure 8-6. Temperature control element.

Figure 8-7 METHOD OF ADJUSTING AUTOMATIC TEMPERATURE REGULATORS.

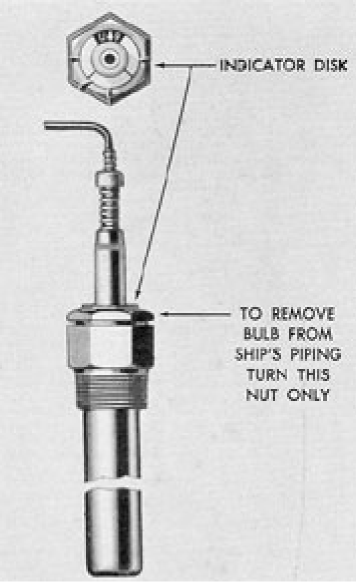


Figure 8-8. Temperature regulator bulb.

8A9. Fresh water treatment. A treating compound may be added to fresh water in a closed cooling system for the prevention of scale formation and corrosion. This compound, when added, must be correctly measured in relation to the amount of water in the system. Too little will have no effect on the prevention of scale formation, whereas too much will increase the corrosion tendencies of the cooling water. The compound consists of a mixture of six parts (by weight) of trisodium phosphate and one part of cornstarch. The mixture must be completely dissolved in warm water, then added to the circulating pump suction.

To determine whether the water contains a sufficient amount of treating compound, a sample of water is drawn from the system. After cooling the sample to 85 degrees F or lower, about 10 milliliters of the cooled sample is transferred to a test tube and one drop of indicator solution, known as corrosion control indicator, is added. The addition of the one drop of indicator solution will change the color of the sample water. If the resulting color is yellow, insufficient treatment is indicated. If the color is red, satisfactory treatment is indicated, and if the resulting color is purple, it denotes that an excessive amount of treating compound has been added.

It should be noted that the addition of the treating compound is a preventive treatment only. It will not remove scale deposits already in the cooling system. If the system is clean and filled with fresh water only, a test of the water as outlined above should result in a yellow color. This indicates that the fresh water is suitable for use and that it will require the addition of at least one standard dose of treating compound, consisting of an ounce of treatment per 100 gallons of water, to bring it into the satisfactory (red) range of the test. If the color of the same test remains yellow after the addition of one standard dose, another dose should be added and this

process repeated until a red color is obtained.

Should the test sample result in a purple color, about one-fourth of the cooling water should be drained from the system and replaced with fresh water. If on retest the purple color

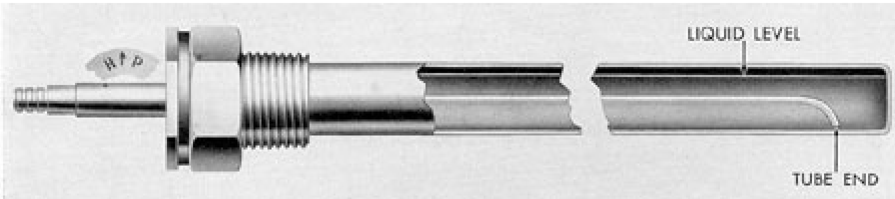


Figure 8-9. Cutaway of thermal bulb.

persists, additional water must be drained and replaced with fresh water. The color of the test sample must be red. It should never be permitted to enter the purple range.

Anti-freeze solutions. Approved anti-freeze solutions may be used to obviate the necessity of draining fresh water systems during freezing temperatures. The liquid usually used is ethylene glycol (Prestone). During freezing weather, all water jackets, cooling chambers, etc., not filled with anti-freeze solution must be thoroughly drained and blown out one at a time, using low-pressure air. Proper blowing out of the water can be accomplished only by closing off all cooling spaces and emptying them separately.

B. FAIRBANKS-MORSE COOLING SYSTEM

8B1. System piping. The F-M engine is cooled by circulating fresh water through its water passages. This water circulates in a closed system.

The external part of the system consists of the expansion tank, electrical resistance thermometer, high-temperature alarm contact maker, fixed orifice, temperature regulator, fresh water and lubricating oil coolers, and connecting piping pressure gages. After performing its engine cooling functions, the water leaves the engine and is piped to the fresh water cooler. There the fresh water is cooled by being passed through a large number of tubes around which cool sea water flows. After leaving this cooler, the fresh water is used as the coolant for the lubricating oil coolers. It is then piped back to the suction inlet, to repeat its passage through the engine.

with mercury bulb thermometers and

A cooler bypass pipe connects the outlet

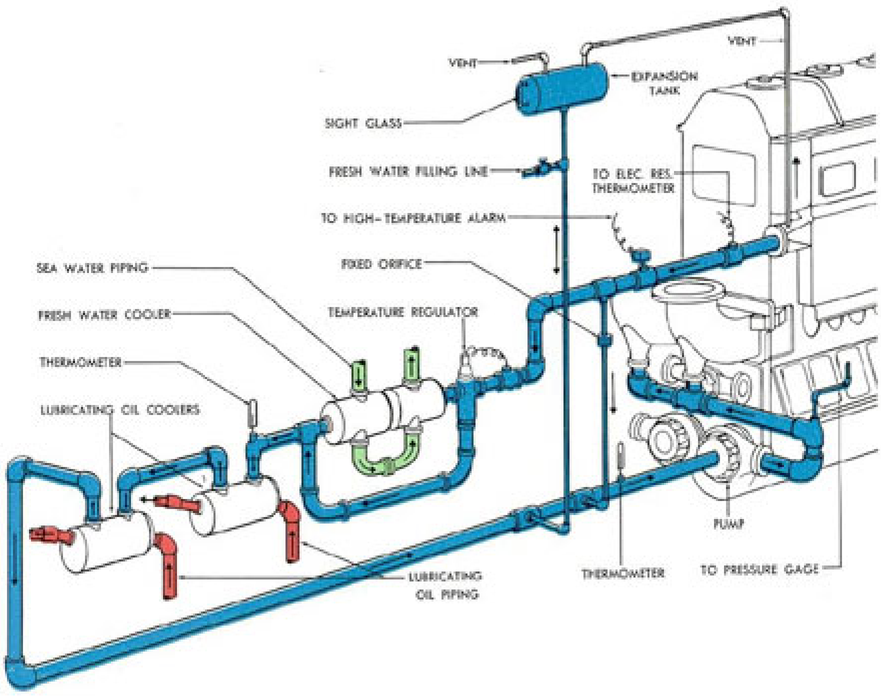


Figure 8-10. Fresh wafer system, F-M.

line from the engine and the suction line to the pump. An orifice in this pipe permits passing a predetermined portion of the fresh water directly back to the pump, rather than through the coolers. This permits cooling of that portion of water going through the complete part of the system sufficiently so that it, in turn, can cool the lubricating oil adequately. From the oil cooler the water mixes with the uncooled fresh water, and enters the engine at the desired temperature.

A bypass pipe is installed across the fresh water cooler inlet and outlet. Flow of water through this cooler bypass is controlled by the automatic temperature regulator. By adjustment of this regulator, the temperature of the water can be controlled at the desired point in the engine

A small pipe leads from the engine water header to the expansion tank. Another small pipe leads from the pump suction line to the expansion tank. This arrangement enables the closed system to accommodate variations in water volume which result from the expansion and contraction of heating and cooling. Water is added to the system through the fresh water filling line from the ship's fresh water system. Excess water is discharged through the expansion tank vent. The bulb of a continuous reading electrical resistance thermometer is in the line from the engine to the cooler. The indicator for the thermometer is mounted on the engine gage board. Between the fresh water pump and the engine inlet a small tube leads to the fresh water pressure gage on the engine gage board. A mercury bulb thermometer is installed near the

under varying operating conditions. Also, when starting the engine, cold water is quickly brought up to good operating temperature range. If the fresh water temperature exceeds a certain set limit, a high-temperature alarm contact maker, mounted in the line between the engine outlet and the cooler, closes the alarm circuit to ring a warning gong.

inlet to the lubricating oil cooler, and another on the line from the coolers to the fresh water pump.

The other pump mounted opposite the fresh water pump on the engine circulates the salt water. This pump draws salt water from the ship's sea chest and forces it through the

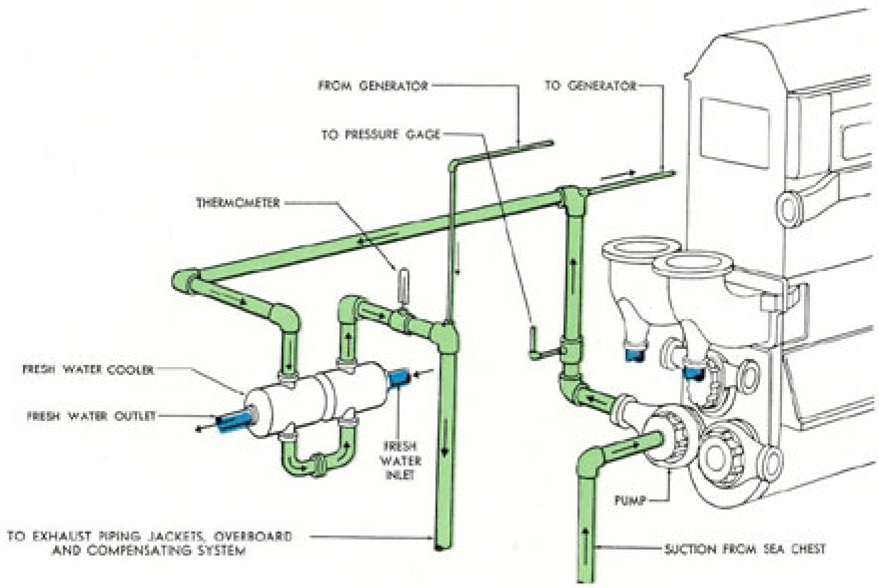


Figure 8-11. Salt water system, F-M.

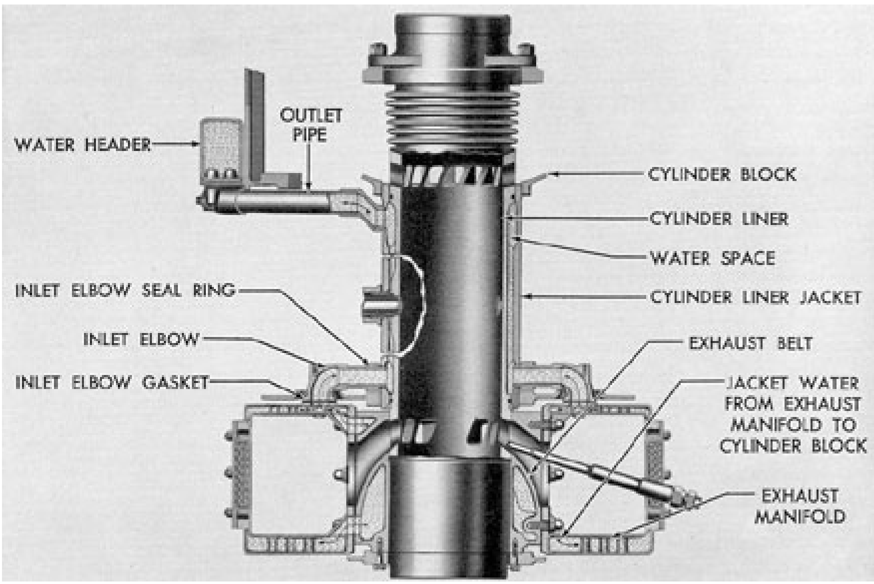


Figure 8-12. Fresh water passage through F-M cylinder.

fresh water cooler and the generator air cooler. Leaving the coolers, the salt water flows

side of each cylinder. These elbows carry the water to the spaces between the cylinder liner and its

through the exhaust piping jackets and overboard. A mercury bulb thermometer indicates the temperature of the salt water, flowing from the fresh water cooler. From the pump discharge pipe, a small tube leads to the salt water pressure gage on the engine gage board.

8B2. F-M engine cooling passages. Entering the engine through an inlet in each exhaust nozzle, the fresh water moves through passages which surround the exhaust nozzles, and on into the exhaust manifold water passages extending the full length of the engine. The exhaust passages from the cylinder liners and the lower part of the liner are also cooled by the fresh water circulation around the exhaust belts. The water enters from the exhaust manifolds at openings at the lower side of the belts and returns to the manifolds at openings at the top. The water then rises from the exhaust manifolds to pass through an inlet elbow on either jacket. Cast-in ribs on the cylinder liner direct the water upward, to cool the liner thoroughly from the bottom. Water passages also lead to the water jackets on the injection nozzle, cylinder relief valve, and air start check valve adapters, to cool these units. Upon reaching the top of the cylinder liner jacket, the water passes out of the cylinder water space through an outlet pipe which leads to the water header. This header, rectangular in cross section, extends along the opposite side of the cylinder block from the control quadrant, just below the air receiver. Its outlet flange is at the control end of the engine where it joins the external part of the system piping.

8B3. Fairbanks-Morse water pumps. The fresh water and salt water circulating pumps in the F-M installations are identical centrifugal pumps. The pumps, mounted on opposite sides at the control end of the engine, are driven by

the lower crankshaft through the flexible drive which also drives the fuel and lubricating oil pump and the governor.

The internal construction of the pump at the impeller end is similar to that of the GM pump. The pump shaft, however, is supported on two bearings, a guide bearing near the impeller end and a thrust bearing at the drive end. Lubricating oil reaches the bearings from the control end compartment of the engine through openings in the pump frame. The oil is distributed by the bearing spacer on the pump shaft. Leakage of oil to the outside of the engine is prevented by an oil seal ring and retainer.

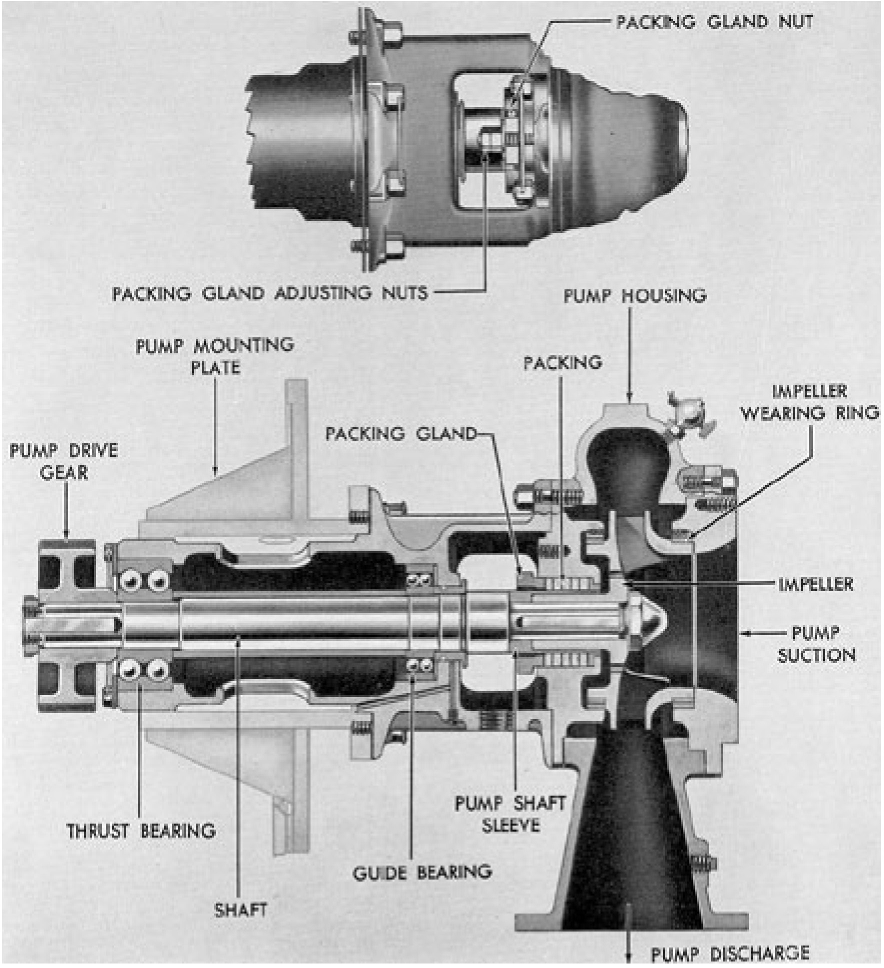


Figure 8-13. Cross section of F-M circulating water pump.

C. GENERAL MOTORS COOLING SYSTEM

8C1. System piping. With the exception of minor differences in the piping arrangement, the cooling system for GM engines is similar to that used in F-M engines. The external part of the closed system is composed of the expansion tank located at the highest point in the system, the fresh water and lubricating oil coolers, the automatic temperature regulator, electrical resistance and mercury bulb thermometers, a pressure gage at the fresh water pump discharge, and the necessary piping. After circulating through the engine, the fresh water passes through a temperature

The salt water overboard discharge is split into several parts. Some of the water goes to the outboard exhaust lines where it circulates through the exhaust line jacket. This water then goes through the outboard exhaust valve for cooling purposes and into the exhaust muffler. Part of this water is sprayed into the muffler to act as a spark arrester, and the rest is piped over the side.

Another line from the salt water system connects into the fuel compensating water line and to the header box. Most of the water going into this line is discharged over the side through the header box, but any water needed to keep

regulator before reaching the fresh water cooler. Water passing through the fresh water cooler is cooled by salt water. Part of the water is bypassed around the cooler and part of it flows through it, depending on the setting of the temperature regulator. The water then goes through the lubricating oil cooler where it acts as the cooling agent. From the lubricating oil cooler, the fresh water returns to the suction side of the fresh water pump for recirculation through the engine. Variations in water volume resulting from expansion and contraction caused by heating and cooling are controlled by two pipe lines, one extending from the expansion tank to the suction side of the pump, the other extending from the expansion tank to the engine fresh water manifold. A vent line at the expansion tank keeps the system free of air, thereby preventing the fresh water pump from becoming air bound, a condition that would result in excessive water temperature.

The salt water pump draws salt water from the sea chest through a strainer and forces it through the fresh water cooler and out through the overboard discharge. A branch line leaving the main line ahead of the fresh water cooler supplies salt water to the generator air cooler. The discharge from the generator cooler joins the outlet pipe extending from the fresh water cooler for overboard discharge. The pressure of the salt water before entering the fresh water cooler is indicated by a pressure gage and is controlled by a the fuel oil and compensating water systems filled flows by gravity to the desired tank through the fuel compensating water line.

8C2. General Motors engine cooling passages. The attached fresh water pump forces fresh water to a manifold located in the scavenging air chamber in each cylinder bank. From the manifolds, the water passes into the cooling spaces of the cylinder liners by way of a water

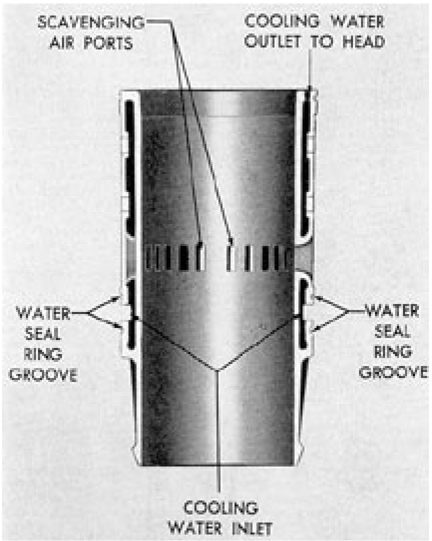


Figure 8-14. Cross section of GM cylinder liner showing cooling passages.

throttling valve located between the salt water pump discharge and the fresh water cooler. A similar valve is used to control the flow of water to the generator cooler.

Figure 8-15. FRESH WATER SYSTEM, GM 16-278A.

Figure 8-16. SALT WATER SYSTEM, GM 16-278A.

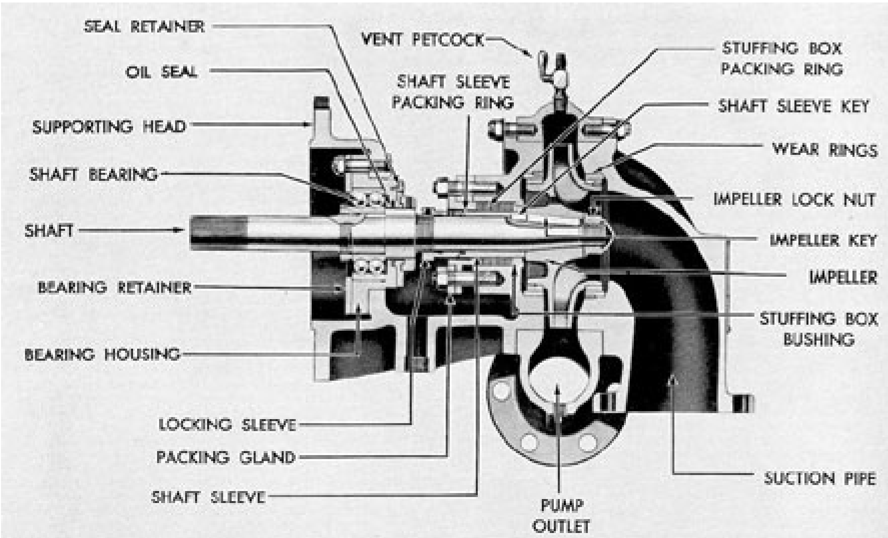


Figure 8-17. Cross section of circulating wafer pump, GM.

connection at the lower deckplate in the engine cylinder block. The water is then forced upward into the cylinder heads through ferrules in the top of the liner, into the water jacket around the exhaust elbows, and finally into the water jacket surrounding the exhaust manifold. From the exhaust manifold, the water enters the external piping leading to the temperature regulator.

8C3. General Motors water pumps. The salt water and fresh water pumps used in GM. cooling systems are of the centrifugal type. The pumps are mounted on opposite sides of the blower housing of the engine and are driven by the crankshaft through the pump outlet. The impeller rotates in the housing on two pairs of replaceable wear rings. A valve sleeve that prevents shaft wear is keyed to the pump shaft and butts against the impeller. A small packing ring fitted in a recess in the end of the valve sleeve provides a watertight seal. The packing is compressed between the sleeve and the shaft by a locking sleeve held in place by a setscrew. When tightening the packing it is first necessary to remove the packing gland which provides access to the setscrew. After loosening the setscrew, the locking sleeve can be rotated with a spanner wrench.

The stuffing box packing that surrounds the shaft sleeve is made

through the accessory drive gear train. The principal differences between the two pumps are in size and capacity. The salt water pump has a capacity of 560 gallons per minute, the fresh water pump, 350 gallons per minute. The following description applies to both pumps.

The principal parts of the pump are the housing, impeller, drive shaft, and pump supporting head. The impeller is keyed to the tapered end of the driving shaft and consists of a number of vanes which throw the water entering at the center of the impeller, outward up of five rings composed of a plastic binder impregnated with lead and graphite. Each ring is about 5 1/16 inch thick.

The pump drive shaft rotates in a ball bearing that is pressed on the shaft and is supported in a bearing housing inside the supporting head of the pump. The bearing is splash lubricated from the accessory drive gear train. A felt seal prevents oil from leaking out of the housing. Water that may work its way along the shaft is prevented from reaching the bearing by a finger locked to the shaft with a setscrew.

SAN FRANCISCO MARITIME NATIONAL PARK ASSOCIATION

Figure 8-7. METHOD OF ADJUSTING AUTOMATIC TEMPERATURE REGULATORS.

Sub Diesel Home Page

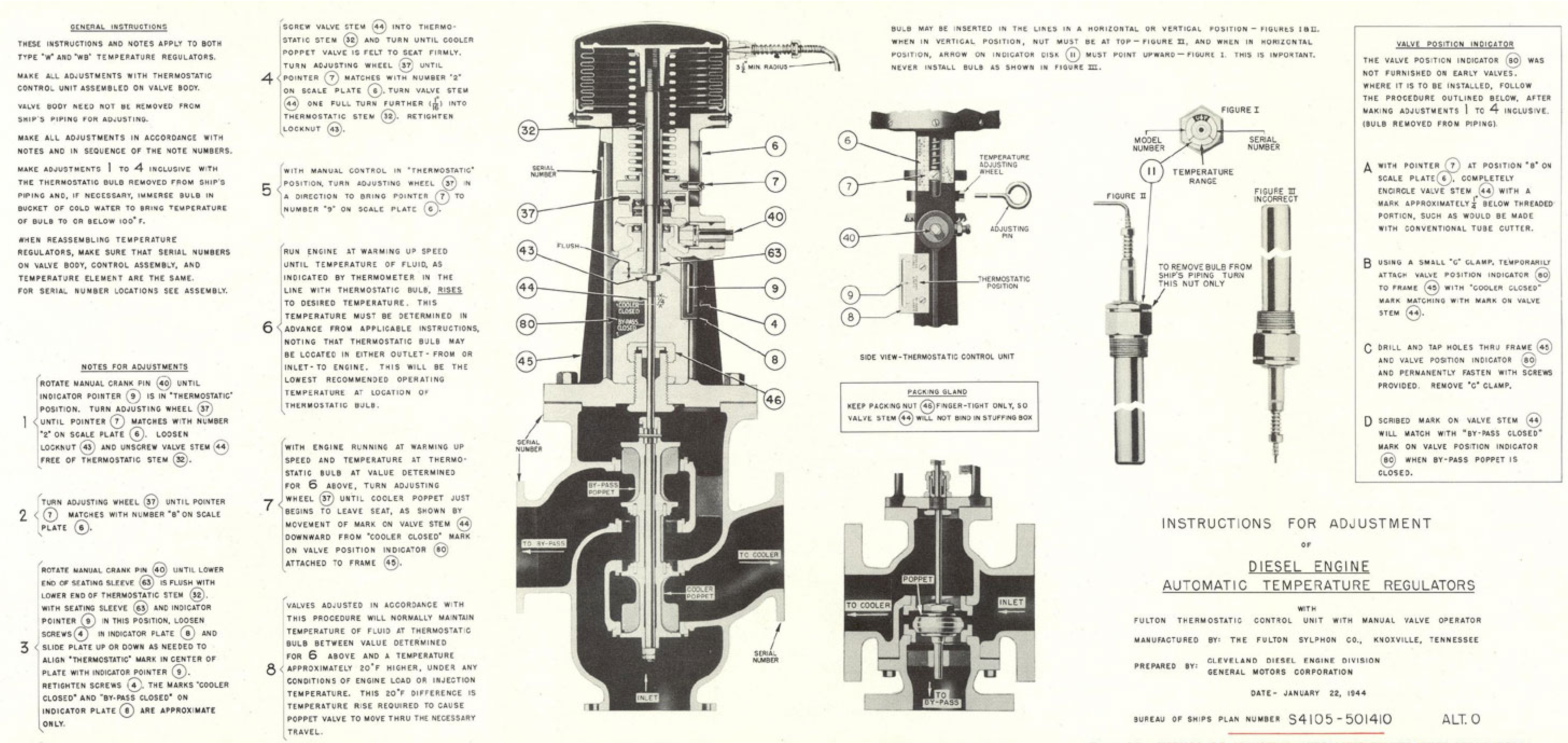
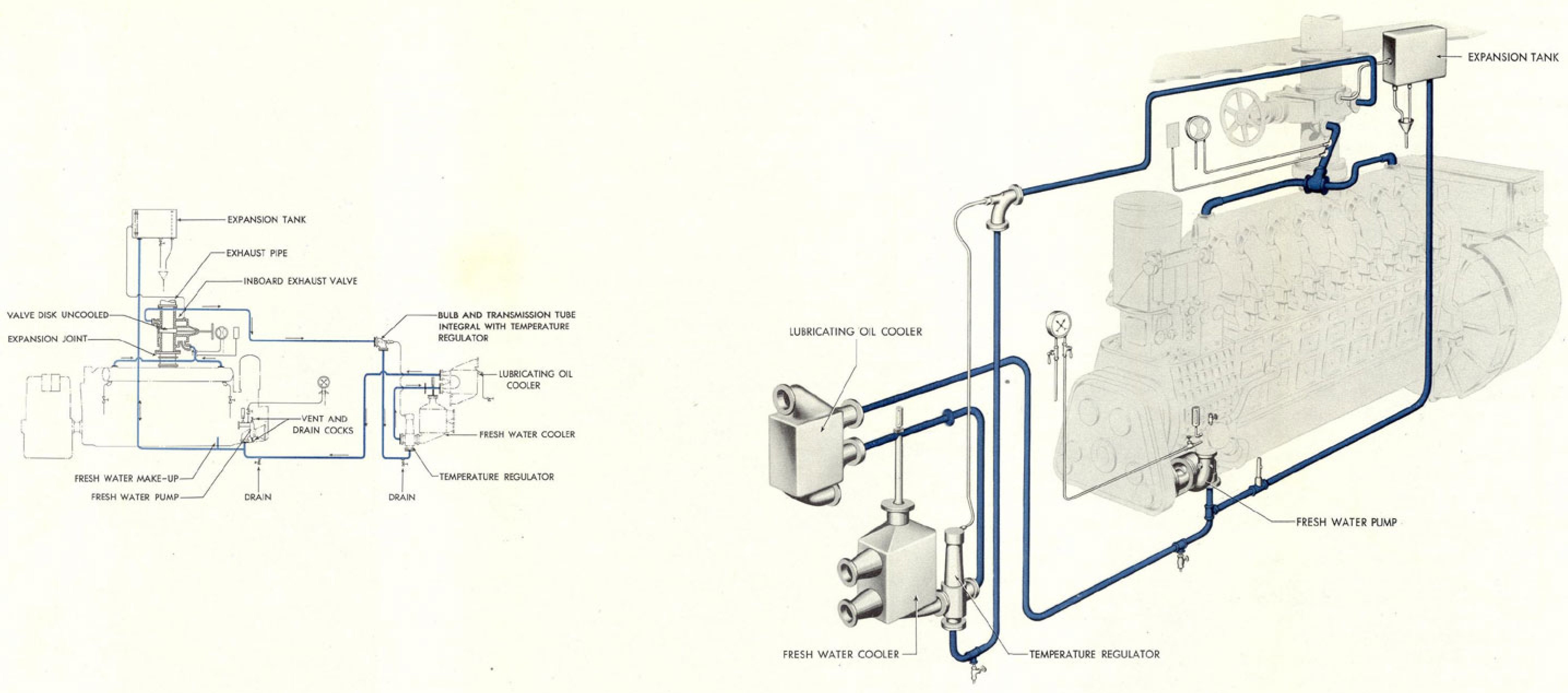


Figure 8-9. METHOD OF ADJUSTING AUTOMATIC TEMPERATURE REGULATORS.

Figure 8-15. FRESH WATER SYSTEM, GM 16-278A.

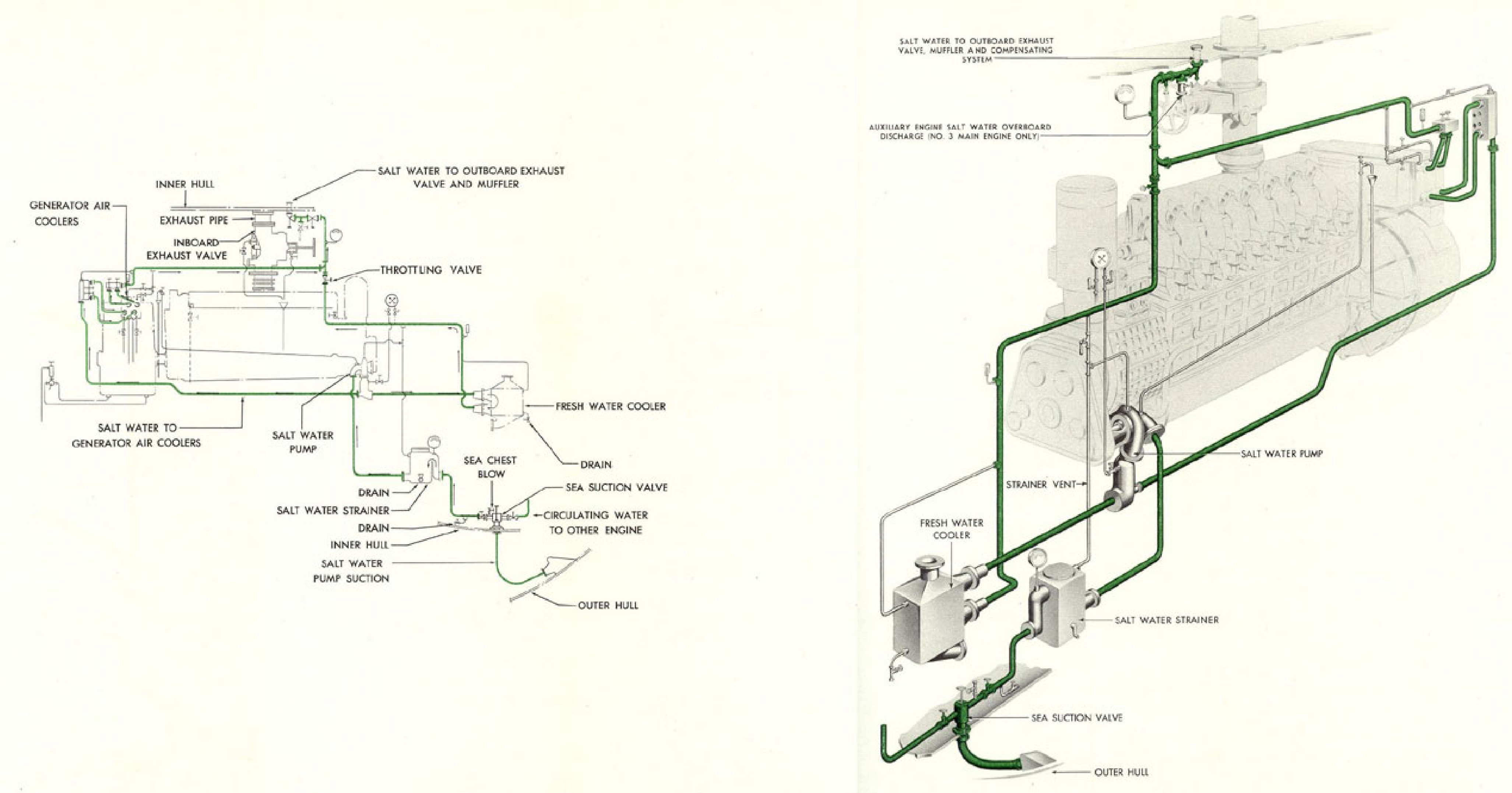
Sub Diesel Home Page



Version 1.10, 22 Oct 04

Figure 8-16. SALT WATER SYSTEM, GM 16-278A.

Sub Diesel Home Page



Version 1.10, 22 Oct 04

9 ENGINE PERFORMANCE AND OPERATION

A. COMBUSTION, AND EFFICIENCY

9A1. Combustion. Engine efficiency is a comparison of the amount of power developed by an engine to the energy input as measured by the heating value of the fuel consumed. In order to understand the various factors responsible for differences in engine efficiency, it is necessary to have some knowledge of the combustion process which takes place in the engine.

In the diesel engine, ignition of the fuel is accomplished by the heat of compression alone. To support combustion, air is required. Approximately 14 pounds of air are required for the combustion of 1 pound of fuel oil. However, to insure complete combustion of the fuel, an excess amount of air is always supplied to the cylinders. The ratio of the amount of air supplied to the quantity of fuel injected during each power stroke is called the air-fuel ratio and is an important factor in the operation of any internal-combustion engine. When the engine is operating at light loads there is a, large excess of air present, and even when the engine is overloaded, there is an excess of air over the minimum required for complete combustion.

1. The fuel must enter the cylinder at the, proper time. That is, the fuel injection valve must open and close in correct relation to the position of the piston.

2. The fuel must enter the cylinder in a fine mist or fog.

3. The fuel must mix thoroughly with the air that supports its combustion.

4. Sufficient air must be present to assure complete combustion.

5. The temperature of compression must be sufficient to ignite the fuel.

Figure 9-1 is a reproduction of a pressure-time diagram of a mechanical injection engine. The lower curvy part of which is a dotted line, is the curve of compression and expansion when no fuel is injected. At A the injection valve opens, fuel enters the combustion chamber and ignition occurs at B. The pressure from A to B should fall slightly below the compression curve without fuel due to absorption of heat by the fuel from the air. The period from A to B is the ignition delay. From B the pressure rises rapidly until it reaches a maximum at C. This maximum, in some instances, may occur at top dead center. At D the injection valve

The injected fuel must be divided into small particles, usually by mechanical atomization, as it is sprayed or injected into the combustion chamber. It is imperative that each of the small particles be completely surrounded by sufficient air to effect complete combustion of the fuel. To accomplish this, the air in the cylinder must be in motion with good fuel atomization, combined with penetration and distribution. In mechanical injection engines this is accomplished by forcing scavenging air into the cylinder with a whirling motion to create the necessary turbulence. This is usually done, in the 2-cycle engine, by shaping the intake air ports, or by casting them so that their centers are slightly tangential to the axis of the cylinder bore.

Before proceeding with the study of the combustion process, the conditions considered essential to good combustion should be reviewed:

closes, the fuel is cut off, but burning of the fuel continues to some undetermined point along the expansion stroke.

The height of the diagram from B to C is called the firing pressure rise and the slope of the curve between these two points is the rate at which the fuel is burned.

Poor combustion of the fuel is usually indicated by a smoky exhaust, but some smoke may be the result of burning lubricating oil that has passed the rings into the combustion chamber. Incomplete combustion is indicated by black smoke, or if the fuel is not igniting, it may appear as blue smoke. Immediately after starting an engine, when running at light loads or at overloads, or when changing from one load to another, smoke is likely to appear.

A smoky exhaust from the engine does not indicate whether one or all the cylinders are

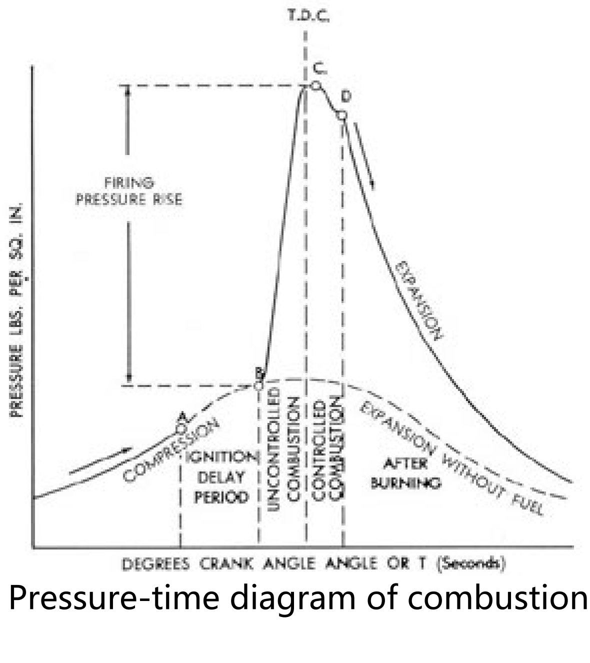


Figure 9-1. Pressure-time diagram of combustion process.

causing it A black-smoking cylinder usually shows a higher exhaust temperature which can be observed from pyrometers installed in the individual exhaust lines from the cylinders. Opening the indicator cock on each cylinder to observe the color of the exhaust is another check. Still another method is cutting off the fuel supply to one cylinder at a time to see what effect it has on the engine exhaust. This latter should never be done when the engine is operating at full load as overloading of the other cylinders will result if the engine is governor controlled.

9A2. Engine losses. It is obvious that not all of the heat content of a fuel can be transferred into useful work during the combustion process. The many different losses that take place in the transformation of heat energy into work may be divided into two classes, thermodynamic and mechanical. The net useful work delivered by an engine is the result obtained by deducting the total losses from the heat energy input.

Thermodynamic losses are caused by:

1. Loss to the cooling system and losses by radiation and convection to the surrounding air.

2. Heat rejected and lost to the atmosphere in the exhaust.

3. Inefficient combustion or lack of perfect combustion.

A loss due to imperfect or incomplete combustion is an important item, because such losses have a serious effect on the power that can be developed in the cylinder as shown by the pressure-volume diagram or indicator card. Complete combustion is not possible in the short time permitted in modern engine design. However, these losses may be kept to a minimum if the engine is kept adjusted to the proper operating condition. Incomplete combustion can frequently be detected by watching exhaust temperatures, noting the exhaust color, and being alert for unusual noises in the engine.

Heat energy losses from both the cooling water systems and lubricating oil system are always present. Some heat is conducted through the engine parts and radiated to the atmosphere or picked up by the surrounding air by convection. The effect of these losses varies according to the part of the cycle in which they occur. The

heat appearing in the jacket cooling water is not a true measure of cooling loss because this heat includes:

1. Heat losses to jackets during compression, combustion, and expansion phases of the working cycle.

2. Heat losses during the exhaust stroke.

3. Heat losses absorbed by the walls of the exhaust passages.

4. Heat generated by piston friction on cylinder walls.

Heat losses to the atmosphere through the exhaust are inevitable because the engine cylinder must be cleared of the still hot exhaust gases before another fresh air charge can be introduced and another power stroke begun. The heat lost to the exhaust is determined by the temperature within the cylinder when exhaust begins. It depends upon the amount of fuel injected and the weight of air compressed within the cylinder. Improper timing of the exhaust valves, whether early or late, will result in increased heat losses. If early, the valve releases the pressure in the cylinder before all the available work is obtained; if late, the necessary amount of air for complete combustion of the next charge cannot be realized, although a small amount of additional work may be obtained. The timing of the exhaust valve is a compromise, the best possible position of opening and closing being determined by the engine designer. It is essential that the valve be tight and properly timed in order to maintain the loss to the exhaust at a minimum. This is also true for air

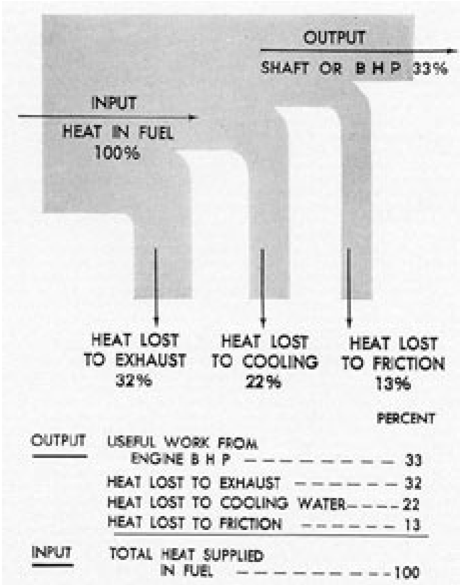


Figure 9-2. Heat balance for a diesel engine.

pumping losses caused by operation of water pumps, lubricating oil pumps, and scavenging air blowers, power required to operate valves, and so forth. Friction losses cannot be eliminated, but they can be kept at a minimum by maintaining the engine in its best mechanical condition. Bearings, pistons, and piston rings should be properly installed and fitted, shafts must be in alignment, and lubricating and cooling systems should be at their highest operating efficiency.

9A3. Compression ratio and efficiencies. a. Compression ratio. The term compression ratio is used quite extensively in connection with engine performance and various types of efficiencies. It may be defined as the ratio of the total volume of a cylinder to the clearance volume of the cylinder. It may be best explained by reference to the pressure-volume indicator card of a diesel cylinder. In Figure 9-3, the volume is reduced from square root(C) + square root(D) to square root(C) during compression. The compression ratio is then equal to

inlet valve setting on 4-cycle type engines.

If an indicator card is taken of a diesel engine cylinder, it is possible to calculate the horsepower developed within the cylinder. This calculation does not take into account the power loss resulting from mechanical or friction losses, as will be discussed later, but it reflects the actual work produced within the cylinder.

Mechanical losses are of several kinds, not all of them present in every engine. The sum total of these mechanical losses deducted from the indicated horsepower developed in the cylinders will give the brake horsepower finally delivered as useful work by the engine. These mechanical or friction losses include bearing friction, piston and piston ring friction, and

(square root(C) + square root(D))/square root(C)

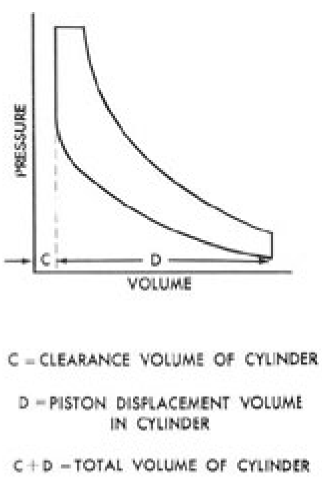


Figure 9-3. Compression ratio.

Compression ratio influences the thermal efficiency of an engine. Theoretically the thermal efficiency increases as the compression ratio is increased. The minimum value of a diesel

the fuel would fire or detonate before the piston could reach the correct firing position.

The temperature-entropy (T-S) diagram of any particular cycle indicates the amount of heat input and the amount of heat rejected. For example, in Figure 9-4, the T-S diagram of a modified diesel cycle, the heat input is represented by the area FBDG and the heat rejected to the exhaust by the area FAEG. The heat represented in doing useful work is represented by the difference between these two, or area ABDE. The efficiency of the cycle can then be expressed as

engine compression ratio is determined by the compression required for starting, which, to large extent is dependent on the type of fuel used. The maximum value of the compression ratio is not limited by the fuel used but is limited by the strength of the parts of the engine and the allowable engine wgt/bhp output.

b. Cycle efficiency. The efficiency of any cycle is equal to the output divided by the input. The diesel cycle shows one of the highest efficiencies of any engine yet built because of the higher compression ratio carried and because of the fact that combustion starts at a higher temperature. In other words, the heat input is at a higher average temperature. Theoretically, the gasoline engine using the Otto or constant volume cycle would be more efficient than the diesel if it could use compression ratios as high as the latter.

The gasoline engine operating on the Otto cycle cannot use a compression ratio comparable to the diesel engine due to the fact that the fuel and air are drawn in together and compressed. If high compression ratios were used, $(H_1-H_2)/H_1$ where H_1 is the heat input along lines BC and CD (the lines representing the constant volume and constant pressure combustion), and H_2 is the heat rejected along line EA (the line representing the constant volume exhaust). Since heat and temperature are proportional to each other, the cycle efficiency is actually computed from measurements made of the temperature. The specific heat of the mixture in the cylinder is either known or assumed, and when combined with the temperature, the heat content can be calculated at any instant. Thus, it is seen that temperature is a measure of heat, and that the heat is proportional to the temperature of the gas.

c. Volumetric efficiency. The volumetric efficiency of an engine is the ratio of the volume that would be occupied by the air charge at atmospheric temperature and pressure to the cylinder displacement (the product of the

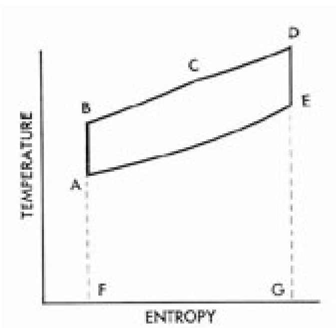


Figure 9-4. Temperature-entropy diagram of modified diesel cycle.

area of the bore times the stroke of the piston). The volumetric efficiency determines the amount of air available for combustion of the fuel, and calculated as previously explained, the indicated thermal efficiency can be computed.

hence influences the maximum power output of the engine.

Volumetric efficiency is actually the completeness of filling of the cylinder with fresh air at atmospheric pressure. The volumetric efficiency of an engine may be increased by enlarging the areas of intake and exhaust valves or ports, and by having all valves properly timed so that as much air as possible will enter the cylinders. Since any burned gases will reduce the charge of fresh air, the supercharging effect gained by early closing of the exhaust valves or ports will reduce the volumetric efficiency. In some engines, the volumetric efficiency is also increased by using special apparatus to utilize air at 2 to 3 psi over the atmospheric pressure. This procedure is commonly called supercharging.

d. Thermal efficiency. Thermal efficiency may be regarded as a measure of the efficiency and completeness of combustion of the injected fuel. Thermal efficiencies are generally considered as being of two kinds, indicated thermal efficiency and over-all thermal efficiency.

If all the potential heat in the fuel were delivered as work, the thermal efficiency would be 100 percent. This is not possible in practice, of course. To determine the values of the above efficiencies the amount of fuel injected is known, and from its heating value, or Btu per pound, the total heat content of the injected fuel can be found. From the mechanical equivalent of

Indicated thermal efficiency = (Indicated hp X 42.42 Btu per minute per hp) / (Rate of heat input of fuel in Btu per minute) X 100 percent

In like manner the over-all thermal efficiency can be found from the brake horsepower or the actual power available at the engine shaft.*

Over-all thermal efficiency =
Brake horsepower / Heat input of fuel X 100 percent

e. Mechanical efficiency. The mechanical losses in an engine decrease the efficiency of the engine and represent the skill with which the engine parts were designed as well as the skill with which the operator maintains the engine. As previously stated, the brake horsepower is equal to the indicated horsepower minus the mechanical losses. The ratio of brake horsepower to indicated horsepower, then, is the mechanical efficiency of the engine which increases as the mechanical losses decrease.

Mechanical efficiency =
Brake horsepower / Indicated horsepower X 100 percent

* This power referred to as shaft horsepower, is the amount available for useful work. It is the power available at the propeller. There is a further loss of power between the main propulsion engine (measured as brake horsepower) and shaft horsepower due to the friction in the reduction gears, hydraulic or electric type couplings, line shaft bearings, stuffing boxes, stern tube

heat (778 foot-pounds are equal to 1 Btu), the number of foot-pounds of work contained in the fuel can be computed. If the amount of fuel injected is measured over a period of time, the rate at which the heat is put into the engine can be converted into potential power. Then, if the indicated horsepower developed by the engine is bearings, and strut bearings. These losses in some cases are considerable and the total loss may be as high as 7 or 8 percent. Therefore, they should not be neglected in making computations.

B. ENGINE PERFORMANCE

9B1. Engine performance. a. General. Many factors affect the engine performance of an engine. Some of these factors are inherent in the engine design; others can be controlled by the operator. The following list of variable conditions affecting the performance of a diesel engine is not complete, but contains all the important factors that should be familiar to operating personnel.

b. Fuel characteristics. The cetane number of the fuel has an important effect on engine performance. Fuels with low cetane rating have high ignition lag. A considerable amount of fuel collects in the combustion space before ignition occurs, with the result that high maximum pressures are reached, and there is a tendency toward knocking. This tends to increase wear of the engine and reduce its efficiency. Fuels with high cetane ratings have low auto-ignition temperatures and hence are easier starting than fuels with low cetane ratings. Therefore, which the engine will operate with a smoky exhaust.

f. Injection rate. The rate of injection is important because it determines the rate of combustion and influences engine efficiency. Injection should start slowly so that a limited amount of fuel will accumulate in the cylinder during the initial ignition lag before combustion begins. It should proceed at such a rate that the maximum rise in cylinder pressure is moderate, but it must introduce the fuel as rapidly as permissible in order to obtain complete combustion and maximum expansion of the combustion products.

g. Atomization of fuel. The average size of the fuel particles affects the ignition lag and influences the completeness of combustion. Small-sized particles are desirable because-they burn more rapidly. Opposed to this requirement is the fact that small particles have a low penetration, and there is therefore a tendency toward incomplete mixing of the fuel and the

diesel engine performance is improved by the use of high cetane number fuel oils.

c. Air temperature. The temperature of the air in the cylinder directly affects the final compression temperature. A high intake temperature results in decreased ignition lag and facilitates easy starting, but is generally undesirable because it decreases the volumetric efficiency of the engine.

d. Quantity of fuel injected per stroke. The quantity of fuel injected determines the amount of energy available to the engine, and also (for a given volumetric efficiency) the air-fuel ratio.

e. Injection timing. The injection timing has a pronounced effect on engine performance. For many engines, the optimum is between 5 degrees to 10 degrees before top dead center, but it varies with engine design. Early injection tends toward the development of high cylinder pressures, because the fuel is injected during a part of the cycle when the piston is moving slowly and combustion is therefore at nearly constant volume. Extreme injection advance will cause knocking. Late injection tends "to decrease the mean indicated pressure (mip) of the engine and to lower the power output. Extremely late injection tends toward incomplete combustion, as a result of combustion air, which leads to incomplete combustion.

h. Combustion chamber design. The amount of turbulence present in the combustion chamber of an engine affects the mixing of the fuel and the combustion air. High turbulence is an aid to complete combustion.

9B2. Power. Engine performance of an internal-combustion engine may be measured in terms of torque, or power developed by the engine. The power that any internal-combustion engine is capable of developing is limited by mean effective pressure, length of stroke, cylinder bore, and the speed of the engine in revolutions per minute (rpm).

a. Mean indicated pressure. The average or mean pressure exerted on the piston during each expansion or power stroke is known as the mean indicated pressure. Mean indicated pressure is of great importance in engine design. It can be obtained from indicator cards mathematically or directly from the planimeter. Excessive mean pressures result in overloading the engine and consequent high temperatures. Temperatures greater than those contemplated in the engine design may cause cracked cylinder heads, liners, and warped valves. There are two kinds of mean effective pressures. One, mip, or mean

indicated pressure is that developed in the cylinder and can be measured. The other is bmep or brake mean effective pressure and is computed from the bhp delivered by the engine.

NOTE. Maximum pressure developed has no bearing on mep.

b. Length of stroke. The distance the piston travels from one dead center to its opposite dead center is known as the length of stroke. This distance is one of the factors that determines the piston speed which is limited by the frictional heat generated and the inertia of the moving parts. In modern engines, piston speed reaches approximately 1600 feet per minute. If the length of stroke is too short, excessive side thrust will be exerted on a trunk type piston. The length of stroke, however, cannot be too great because of the lack of overhead space available on submarine type engines.

c. Cylinder bore. The cylinder bore is its diameter, and from this the cross-sectional area of the piston is determined. It is upon this area that the gas pressure acts to create the driving force. This pressure is the mean indicated pressure referred to above, expressed and calculated for an area of 1 square inch. The ratio of length of stroke to cylinder bore is somewhat fixed in engine design. There are a few instances in which the stroke has been less than the bore, but in almost every case the stroke is longer than the bore. This ratio in a modern trunk-piston type engine is single-acting, 2-stroke cycle engine, there is a power stroke for each revolution.

Having defined the factors influencing the power capable of being developed, the general formula for calculating horsepower is as follows:

IHP = (P X L X A X N) / 33,000

P = Mean indicated pressure, in psi
L = Length of stroke, in feet
A = Effective area of the piston in square inches
N = Number of power strokes per minute

The horsepower developed within the cylinder as a result of combustion of the fuel can be calculated by measuring the mean indicated pressure and engine speed. Then with the bore and stroke known, the horsepower can be computed for the type of engine being used. This power is called indicated horsepower because it is obtained from the pressure measured from an engine indicator card. It does not take into account the power loss due to friction, as will be discussed later. Example:

Given a 12-cylinder, 2-cycle, single-acting engine having a bore of 8 inches and a stroke of 10 inches. Its rated speed is 720 rpm. When running at full load and speed, the mean indicated pressure is measured and is found to be 105 psi. What is the indicated horsepower developed by the engine?

Solution:

From the formula

about 1.25, while in a crosshead type engine in use today it is about 1.50.

d. Revolutions per minute. This is the speed at which the crankshaft rotates, and since the piston is connected to the shaft, it determines, with the length of stroke, the piston speed. Since the piston moves up and down each revolution, the piston speed is equal to twice the stroke times the revolutions per minute (rpm), and is usually expressed in feet per minute. If the stroke is 10 inches, and the speed of rotation is 750 rpm, the piston speed is

750 X 2 X (12/10) = 1,250 feet per minute.

The power developed by the engine depends upon the engine's speed and the type of engine. If it is a single-acting, 4-stroke cycle engine there will be one power stroke for every two revolutions of the crankshaft. If it is a

IHP = (P X L X A X N) / 33,000

P = 105
L = 10 / 12
$A = 3.1416\ (8/2)^2$
N = 720

$IHP = (105 \times (10/12) \times 3.1416\ (8/2)^2) \times 720$

IHP = 96.96

Since this is just the horsepower developed in one cylinder, if the load is perfectly balanced among all cylinders, the total indicated horsepower of the engine is

IHP = 12 X 96.96 = 1163.5

e. Brake horsepower. As stated above, brake horsepower is the power delivered by the engine in doing useful work. Numerically, it is equal to the indicated horsepower minus the mechanical losses.

BHP = IHP minus the mechanical losses.

From the example above, the IHP was found to be 1163.5. If the brake horsepower of this engine was 900 as determined in a test laboratory, then the mechanical losses would be

be determined from the indicated horsepower under varying conditions of operation. It should be noted that as a rule, indicator cards taken on engines having a speed over 450 rpm are not reliable and therefore no indicator motions are provided.

9B3. Engine performance limitations. The power that can be developed by a given size cylinder whose piston stroke is fixed is limited only by the piston speed and the mean effective pressure. The piston speed is limited by the inertia forces set up by the moving

1163.5 - 900 = 263.5 horsepower

or

(263.5 / 1163.5) X 100 = 22.6 percent of the indicated horsepower developed in the cylinders

or 90 / 1163.5 = 77.4 percent mechanical efficiency.

Engine power is frequently limited by the maximum mean pressure allowed. To find the bmep of the above engine, first obtain the power developed in one cylinder. Thus,

900 / 12 = 75.0 bhp

From the general formula for horsepower,

HP = (P X L X A X N) / 33,000

75 = P X (10/12) X 3.1416 X $(8/2)^2$ 720 /33,000

P = (75 X 33,000) / (10/12 X 3.1416 X $(8/2)^2$ X 720)

P = 82.1 psi

Hence, for the above engine under the conditions stated the bmep is 82.1 while the mip is 105 psi.

The brake horsepower is the power available at the engine shaft for useful work. Brake horsepower cannot usually be measured after an engine is installed in service, unless the engine drives an electric generator. The brake horsepower is determined by actual tests in the shops of the manufacturer before delivery of the engine. Frictional losses are quite independent of the load on the parts and the problem of lubrication due to frictional heat.

The mean indicated pressure is limited by:

1. Heat losses and efficiency of combustion.

2. Volumetric efficiency or the amount of air charged into the cylinder and the degree of scavenging.

3. Complete mixing of the fuel and air which requires fine atomization, sufficient penetration, and a properly designed combustion chamber.

The limiting mean effective pressures, both brake and indicated, are prescribed by the manufacturer or the Bureau of Ships and should never be exceeded. In a direct-drive ship, the mean effective pressures developed are determined by the rpm of the shaft. In electric-drive ships, the horsepower and mep can be determined readily from the electrical readings, taking into account generator efficiency.

The diesel operator should remember that the term overloading means exceeding the limiting mean effective pressure.

9B4. Operation. All submarine type diesel engines are rated at a given horsepower and a given speed by the manufacturer. These factors should ordinarily never be exceeded in the operation of the engine. Using the rated speed and bhp, it is possible to determine a rated bmep which each individual cylinder should never exceed, otherwise that cylinder will become overloaded. The rated

engine. Hence, unless the brake horsepower has been measured at various loads and speeds, the mechanical losses cannot

bmep holds only for rated speed. If the speed of the engine drops down below rated speed, then the cylinder bmep which should not be exceeded generally drops down to a lower value due to propeller characteristics. The bmep should never exceed the normal mep at lower engine speed. Usually it

should be somewhat lower if the engine speed is decreased.

Navy type engines are generally rated higher for emergency use than would normally be the case with commercial engines. The economical speed for most Navy type diesel engines is found to be about 90 percent of rated speed. For this speed the optimum load conditions have been found to be from 70 percent to 80 percent of the rated load or output. Thus, we speak of running the engines at an 80-90 combination which will give the engine parts a longer life and will keep the engine itself much cleaner and in better operating condition. The 80-90 means that we are running the engine with 80 percent of rated load at 90 percent of rated speed.

Diesel engines do not operate well at exceedingly low bmep such as that occurring at idling speed. This type of engine running tends to gum up pistons, rings, valves, and exhaust ports. If an engine is run at idling speed for long periods of time, it will require cleaning and overhaul much sooner than if it had been run at 50 percent to 100 percent of load.

Some engine manufacturers design their engine fuel systems so that it is impassible to exceed the rated bmep to any great extent. This is done by limiting the maximum throttle or fuel control setting by means of a positive stop. This regulates the maximum amount of fuel that can enter the cylinder and therefore the maximum load of the cylinder.

C. LOAD BALANCE

9C1. Indications. Load balance means the adjustment of the engine so that the load will be evenly distributed among all the cylinders of the engine. Each cylinder must produce its share of the total work done by the engine in order to have a

from individual cylinders indicate an overloaded condition of these cylinders. A high common exhaust temperature in the exhaust header indicates a probable overloading of the whole engine. These conditions are indicated by pyrometers installed in all modern

balanced load. If the engine is developing its rated full load, or nearly so, and one cylinder or more is producing less than its share of the load, the remainder of the cylinders obviously must be doing more than their share of the total work and hence are overloaded.

An overloaded condition of an engine, or of one or more of its cylinders, may be indicated by:

1. Black smoke in the exhaust.

2. High exhaust temperature.

3. High lubricating oil and cooling water temperature.

4. Hot bearings and high temperatures of other engine parts (in general, a hot running engine).

5. Excessive vibration of the engine.

6. Unusual sound of the engine.

When black smoke is observed in the exhaust from the mufflers, it is not possible to determine immediately whether the entire engine or just one of the cylinders is overloaded. However, by opening the indicator cocks on the individual cylinders, the color of their exhausts can be determined.

High temperatures of the exhaust gases

engines. A constant check on the pyrometer readings will indicate accurately when any cylinder is firing properly and carrying its correct share of the load. Any sudden change in the reading of the exhaust temperature of any cylinder should be investigated immediately. The difference in exhaust temperatures between any two cylinders should not exceed 25 degrees F for a well-balanced engine. However a certain tolerance is allowed; usually 50 degrees to 75 degrees is permissible.

Thermometers are provided in the lubricating oil and cooling water systems. Modern diesel engines have thermometers installed in the cooling systems of individual cylinders. An abnormal rise in any of these temperatures may indicate an overloaded condition and should be investigated as quickly as possible.

In general, excessive heat in any part of the engine may indicate overloading. An overheated bearing may be the result of overloading a cylinder. An abnormally hot crankcase could result from overloading the engine as a whole. Excessive temperatures of some engine parts can be checked by touch.

If all cylinders are not doing an equal

amount of work, the force exerted by individual pistons will be unequal. In this event, the unequal forces may cause an

out should be within 10 to 20 psi of each other in all cylinders of a properly adjusted engine.

uneven turning moment to be exerted on the crankshaft and vibrations will be set up. The skilled operator can tell by the feel and the sound of an engine when a poor distribution of load exists. This, of course, comes from long experience, but it is important that the beginner avail himself of every opportunity to observe engines running under all conditions of loading and performance.

9C2. Causes of unbalance. In the preceding section some of the general causes of unequal load distribution were discussed. To prevent unbalance in an engine, the foremost consideration is that the engine must be in excellent mechanical condition. A leaky valve or fuel injector, leaky compression rings, or any other such mechanical difficulties will make it impossible for the operator to balance the load unless he secures the engine and dismantles at least a part of it. Therefore, the engine must be placed in proper mechanical condition before the load can be balanced.

Since the heat of compression is relied upon to ignite the fuel injected in the diesel engine, the amount of this compression must be maintained within fixed limits. In order to have the same type of combustion in each cylinder, the degree of compression in all cylinders should be approximately the same. For example, low compression pressure in one cylinder may prevent all the fuel from burning, or may even prevent ignition of the fuel in

In order to have the load equally distributed, each cylinder must receive the same amount of fuel. It is here that the effect of an improperly adjusted fuel pump is evident. A cylinder receiving more fuel than necessary for a given load will develop more power than required.

Any adjustment of the fuel pump must be undertaken only by a person thoroughly familiar with the type of pump being used. He should first determine beyond all doubt that the engine is in proper mechanical condition. A great many factors may cause the cylinder to fire unevenly. Some of these causes are a clogged or improperly timed fuel injection valve, improperly timed air intake or exhaust valve, air or water in the fuel system, improper rocker arm valve clearance, dirt or other foreign matter in the fuel oil which may be plugging up the strainers and filters, and any other factor that contributes to poor combustion. If a cylinder is firing incorrectly, always check the above conditions before making any adjustments to the fuel pump.

Changing the amount of fuel being delivered by adjusting the pump should be done only when it is certain that the cause of the trouble is in the pump. This point cannot be emphasized too strongly. For instance, if the failure of a cylinder to fire correctly was due to a clogged fuel injection valve tip and the operator increased the fuel supply to the cylinder with the intention of increasing the power developed by that particular cylinder, the increase in fuel might wash the valve clean and cause the cylinder

that cylinder. This would result in a reduced amount of work or no work being done by this cylinder. The common causes of low compression are:

1. Sticking compression rings.

2. Excessive ring or cylinder wear.

3. Leaky cylinder head gasket or cracked cylinder head.

4. Leaky valve in cylinder head.

5. Cracked cylinder liner.

6. Excessive clearance volume.

In correcting these, it is generally necessary to replace the defective part. However, in some cases such as a sticking ring or valve, it is necessary only to clean the part and replace it. These cold compression pressures with fuel cut to become badly overloaded from the excess fuel supplied. The correct procedure would have been to replace the clogged injection valve with a spare and to clean the one that was removed. The decrease in power delivered by a cylinder may also be due to some foreign matter under a valve or piston ring, and once cleared, the cylinder would become overloaded if the fuel supply had, in the meantime, been increased.

The operator who always maintains his plant in good mechanical condition will be required to make few, if any, adjustments to the

fuel system while it is running. The fuel supply to an individual cylinder should not be adjusted until after an exhaustive search has revealed that every other condition is normal in all respects.

After an overhaul in which piston rings of cylinder liners have been renewed, considerable adjustment of the engine may be necessary. Lubricating oil will leak by the rings into the combustion space until after the rings have properly seated. The compression will also increase as the seal between the rings and the liner becomes more effective. The lubricating oil will burn in the cylinder, giving an incorrect developed within a cylinder is directly proportional to the power produced by that cylinder, any increase in one will cause a corresponding increase in the other. Hence, if the power is not evenly distributed throughout the cylinders, the mean indicated pressures in the individual cylinders will vary. Temperature varies directly as the pressure, so that a decrease in pressure will result in a corresponding decrease in temperature, The quality of combustion obtained depends upon the heat, and heat upon the temperature, so that with a decrease in pressure, combustion will not be so good as before. This poor combustion will lower the

indication of fuel oil combustion, and if the pump has been properly set when the engine was started, the engine will be overloaded, or at least unbalanced. As the compression rises to normal pressure, the power developed will increase as also will the conditions of pressure and temperature under which the combustion takes place. Hence, when an overhaul has been completed, the engine must be carefully watched until the rings are seated, and the compression set to the level specified in the instructions for that type of engine. This adjustment will be facilitated by the use of frequent compression tests. If the engine is not fitted so that the compression can be readily varied, the engine should be run under light load until it is certain that the rings have seated.

9C3. Effect of unbalance. In general, the effect of unbalance is an overheated engine. Clearances are established by the engine designer to allow for sufficient expansion of moving parts when operating at the designed temperatures. Consequently, an engine operating at temperatures in excess of those for which it was designed may suffer many casualties. Excessive expansion of the moving parts will cause seizures and a burning up of the engine. If the temperatures rise above the flash point of the lubricating oil vapors in the crankcase, an explosion may result. The high temperatures may destroy the lubricating oil film between adjacent surfaces of the moving parts and result in thermal efficiency, and the output of the engine will be reduced.

If an engine is developing 600 bhp, and its mechanical efficiency is 80 percent, the indicated horsepower being developed is 750. If the engine has 10 working cylinders, each cylinder should be producing 75 indicated horsepower. When this is not the case the engine is unbalanced. The effect here would be to increase the mean indicated pressure of those cylinders doing less than their share of the work, and to decrease that of those cylinders producing more than 75 indicated horsepower.

The turning moment acting on the crankshaft is proportional to the force acting on the piston. This force, in turn, is the result of the mean indicated pressure developed in the cylinder. If these forces from different cylinders are not equal, there is an uneven turning moment acting along the length of the crankshaft, and vibrations result. These vibrations, if sufficiently severe, may shake the engine loose in its foundation, crack the engine housing, framework, and bedplate, destroy the bearings, and even break the crankshaft. It is obvious that a badly vibrating engine can result in serious damage and should be stopped immediately.

To avoid all the harmful effects of overloading and unbalancing of load, the load on a diesel engine should be equally distributed among the working cylinders; and no cylinder, or the engine itself, should ever be overloaded. In conclusion, the correct procedure to follow in balancing an engine is:

further increased temperatures due to the increased friction. In fact, the effect is the same as for overheating from any cause.

Since the mean indicated pressure

1. Maintain the engine in proper mechanical condition.

2. Adjust the fuel system in accordance with the manufacturer's instructions.

3. Operate the engine within the temperature limits specified in the instructions.

4. Keep the cylinder temperatures and pressures as evenly distributed as possible.

5. Train yourself to detect a bad condition by the senses of touch and hearing.

D. ENGINE DYNAMICS AND VIBRATIONS

9D1. Balancing. It is not possible to balance out all the forces producing vibration in an engine. However, the primary or principal forces may be almost entirely balanced by the addition of weights to the crankshaft or connecting rods at the proper places. Balancing by the addition of weights so as to create forces equal and opposite to those of inertia is known as counterbalancing. Usually, after counterbalancing, there are still some small forces remaining that have not been completely balanced out. These remaining forces are produced by the reciprocating parts, since it is possible to completely counterbalance all primary rotating forces.

All rotating parts are subjected to two kinds of unbalance. They are called static unbalance and dynamic unbalance. The unbalanced condition in both cases can be readily determined excessive vibration in service. This is due to the low speeds used with the balancing machines. Diesel engines in the service must operate over a wide speed range usually, and for this reason they are not accepted until after they have been tried at all speeds at which they must operate when installed in service.

In any event, all rotating parts of the engine should be as accurately balanced as possible.

9D2. Flywheels. A flywheel stores up energy, the amount of which depends upon the rotating speed, the weight, and the diameter of the wheel. In most marine engines heavy flywheels are not necessary, as the other rotating masses on the shaft serve the same purpose. These masses are the clutch and generator, and with a large number of cylinders firing, the power stroke is smoother, and there is less need for a flywheel.

and corrected by counterbalancing.

A static balancing test is conducted by placing the two ends of the rotating part on perfectly smooth, horizontal, and parallel rails. If statically unbalanced, the part will roll on the rails until its center of gravity reaches its lowest position and then it will come to rest. If, however, its center of gravity lies along its axis it will remain at rest when placed in any position, and it is then in static balance.

It frequently occurs that the center of gravity of a body lies in its axis of rotation but that its irregular shape or composition generates a disturbing force when the body is rotated. In this case the body would be in static balance and in dynamic unbalance. In general, before balancing, most rotating parts are in both static and dynamic unbalance.

In all cases, complete balancing can be obtained by attaching weights to the rotating body, if the position and degree of unbalancing are known. For determining this unbalance all naval shipyards are equipped with balancing machines. Experience with large and high-speed machinery has shown that balancing machines show good results but do not insure against

The flywheel serves three purposes, namely:

1. To prevent the engine from stalling when running at idling speed.

2. To reduce the variations in speed at all loads.

3. To help carry the engine over centers when starting.

When the speed of the shaft tends to increase, the flywheel absorbs energy. When it tends to decrease, the flywheel gives up its energy to the shaft in an effort to keep it rotating at a uniform speed.

9D3. Torsional vibrations. The twisting and untwisting of the shaft system result in torsional vibrations. All shafts have some flexibility and with weights attached to them, such as pistons, gears and camshafts in diesel engines, they have what is known as a natural fixed frequency. When the frequency of the power stroke impulses coincides with the natural frequency of the entire shaft system, a torsional vibration is produced, and the shaft is then said to be

rotating at a critical speed. This critical speed is dependent on the dimensions of the crankshaft, the number of cylinders, all

bedplate, crankcase, or similar members, results in flexural vibrations. The cause of flexural vibration lies in the faulty balance

rotating masses of the engine, other shafting and masses including the propeller, the number of power strokes per minute, the arrangement of the cylinders (whether they are in line or in a V), and the cylinder firing order.

Without going into further detail, it is sufficient to say that torsional critical speeds depend upon the number of power impulses per revolution and the natural rate of vibration of the combined shaft system. Special instruments are available for determining the degree of torsional vibration and the natural frequency of any particular shaft system.

To change the range or point of maximum vibration of the critical speeds for a given installation it is necessary to make a change in the masses on the elastic shaft system. It is evident therefore that in engines operating at a constant speed, it is much simpler to change the natural frequency in order to avoid dangerous critical speeds than it is in a marine engine requiring a wide range of operating speeds.

Critical speeds and mode of vibrations are determined with the aid of an instrument recording torsional vibrations. The engine builders calculate the critical speeds and furnish a guarantee that, with the engine coupled to the load for which it is designed, no dangerous critical speeds will occur within the operating speeds.

Torsional vibrations need not necessarily shake the framing of of the rotating and reciprocating masses of the engine and the presence of the so-called free forces or rocking couples. It may be manifest in the horizontal or vertical planes and may in turn be the cause of vibration of surrounding structures, such as the ship's hull in marine installations. This type of vibration does not depend on the way the engine is coupled to its load, and if an engine does not vibrate on test, no vibrations will develop after it is placed in service.

9D5. Torsional vibration dampening. There are certain forces acting in resistance to torsional vibrations. These forces are due to the friction of the bearings that carry the shafting and the work absorbed in the metal of the shaft in resisting the twisting called hysteresis. Propellers in the water are the most influential factor. All of these forces may be said to be the result of natural causes, and they act to dampen out, or reduce the amplitude of the torsional vibrations. In addition to these natural forces, there are other methods employed to reduce or eliminate the severity of the vibrations. This may be accomplished by changing the firing order of the cylinders in the engine, or by changing the rotating weights, or the flexibility of the shaftings.

In addition to the above dampening factors and methods there are various types of commercial torsional vibration dampeners, such as that used on the F-M 38D 8 1/8 10-cylinder engine. Each such dampener must be designed for a specific shaft

the engine and may not even be noticeable to the operator. This fact has been borne out in several casualties in which the crankshaft broke without warning. Excessive wear of gears or of attached auxiliaries and repeated breakage of shafting or other parts attached to it can very well be caused by torsional vibrations. Most installations in naval vessels have been checked and tested to determine the exact location of torsional vibrations, their amplitudes, and frequencies.

9D4. Flexural vibrations. The bending of the parts of the engine framing such as the system operating with a particular type engine. Vibration dampeners are usually located at or near a point of maximum torsional vibration amplitude along the shaft, generally at the forward end of an engine.

There are several different types of dampeners. All, however, accomplish the same purpose. They tend to reduce the swinging motion of the shaft. This is accomplished by having a freely rotating disk or disks acting against a fixed disk which creates friction and thereby acts as a brake. This prevents the shaft, from twisting and untwisting while rotating on its axis.

E. ENGINE PRESSURE INDICATOR

9E1. General. Efficient uninterrupted performance of the engine depends upon the maintenance of equal correct compression without fuel, and firing pressures with fuel among the various cylinders. Poor engine compression causes loss of power, poor acceleration, smoky exhaust, and starting difficulties. An abnormally high firing pressure in one or more cylinders may cause engine wear, uneven running, and overheating. These compression pressures may be measured by instruments known as pressure indicators. Compression readings without fuel are taken after the engine is warmed up and the fuel cut off on that particular cylinder. Firing pressure readings are taken with the engine horizontal distance will represent piston movement. As an example, in a two-stroke cycle engine, one complete revolution or cycle would produce a diagram like the one shown in the illustration. This diagram is called an indicator card. If the indicator spring is calibrated so that the number of pounds of pressure required to raise the pencil 1 inch is known, then to read the pressure at any point on the card all that is necessary is to measure the distance in inches from the atmospheric line X-Y on the diagram to the point at which the amount of pressure is desired, and multiply this by the calibration number of the spring. The total length of the diagram represents the stroke of the piston. This horizontal scale then can be laid

warmed up and operating under a stated load at a stated speed.

9E2. Types of engine indicators. There are two general classes of engine pressure recording indicators. In the first, the instrument measures graphically the cylinder pressure and at the same time indicates the position of the piston at any point of its stroke or cycle. In other words, the indicator draws a diagram of the pressure in the cylinder with respect to the movement of the piston. Since the movement of the piston is a measure of the volume displaced, the diagram is drawn to the ordinates of pressure and volume. In the second general class, the indicator records the maximum pressures only.

Figure 9-5 shows the fundamental principle of the operation of an engine indicator in which the movement of the piston is recorded. The indicator equipment includes a small cylinder that can be attached to the main working cylinder of the engine, a piston and rod that work in this small cylinder, with a pencil on the end of the rod. The pencil point bears on the paper tacked to the drum which is moved by hook and string over a pulley. Any pressure in the working cylinder enters the indicator cylinder and forces the small indicator piston and pencil in a vertical direction at the same time the main piston moves the card in a horizontal direction by means of the string and pulley.

It is readily seen that any vertical distance on the diagram will represent pressure, and the off in inches, feet, piston stroke, or volume of piston displacement.

This type of indicator is little used by operating personnel on fleet type submarines today, mainly because there is no provision made on modern engines for the attachment of the equipment necessary to take the indicator card, and also because there are no means of compression adjustment other than complete overhaul of the engine.

The other type of indicator (indicating maximum pressures only) is used to some extent for taking maximum cylinder pressures, to check against manufacturer's test data and previous shipboard pressure tests. The two most commonly used indicators of this type are the Premax indicator and the Kiene indicator.

The method normally used to check the equal distribution of power among the various cylinders is to compare the exhaust gas temperatures of the cylinders by means of thermocouples placed in the exhaust elbows of each cylinder. Pyrometer readings have proved to be a good check on the general running conditions of an engine, and the records of exhaust gas temperatures are of great value in conjunction with indicator readings as aids in getting the best results from a diesel engine. However, even though the exhaust temperatures are normal, the engine at times may not develop its rated horsepower.

9E3. Premax indicator. The Premax indicator is an instrument for determining cylinder

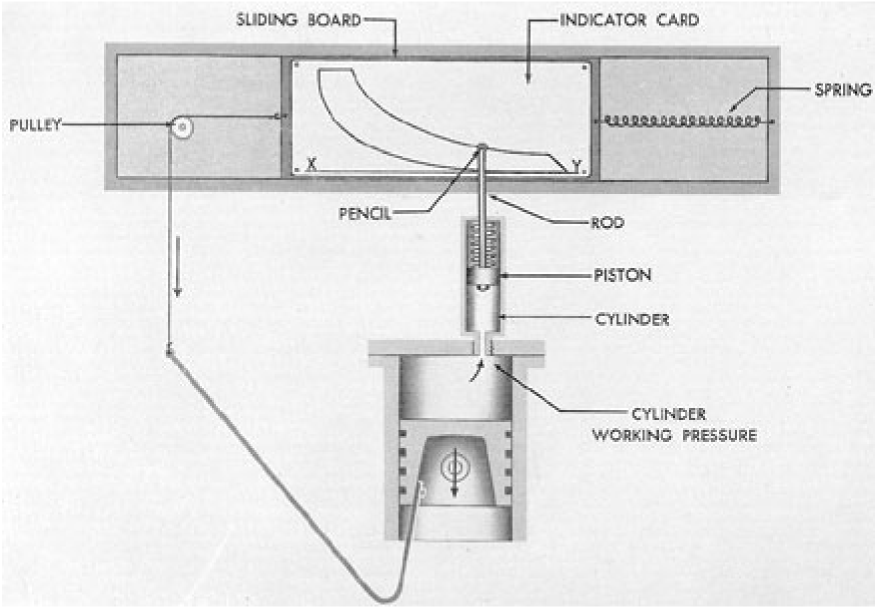


Figure 9-5. Principle of engine indicator.

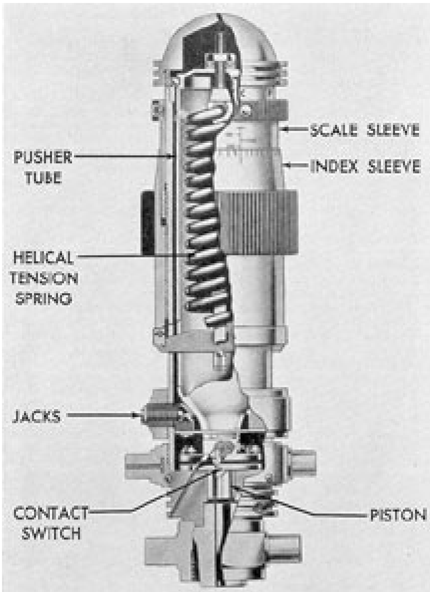


Figure 9-6. Premax pressure indicator.

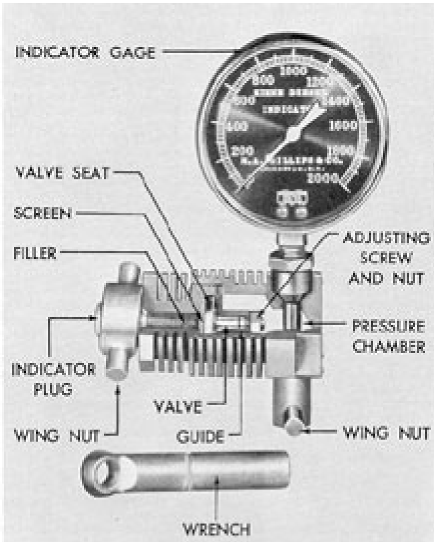


Figure 9-7. Kiene pressure indicator.

compression and firing pressures. The indicator consists essentially of a piston subject to cylinder pressure, a spring against which the piston acts and the tension of which is adjustable by means of an index sleeve, a control switch, and a neon light circuit that shows if the piston is moving. It is attached to the cylinder indicator cock in the same way as any

9E4. Kiene indicator. The Kiene diesel indicator is a pressure indicator gage for measuring the compression and firing pressure of an engine while it is running. The complete unit consists of a pressure gage and an air-cooled pressure chamber which is attached to the cylinder indicator cock.

other indicator. The pressure acting on one side of the piston in the indicator is gradually increased by increasing the spring tension with the index sleeve until this spring pressure is equal to the maximum cylinder pressure which acts on the opposite side of the piston. When the two pressures are equal, the piston stops moving, as shown by stopping of the neon light flashes. The pressure reading is then read on the scale sleeve.

The cylinder discharge passes through the indicator plug up through the filler, screen, and seat piece. This raises the valve, allowing the gas to pass through the drilled holes in the guide piece into the pressure chamber and on to the gage. The action of the gas in the curved tube of the gage tends to straighten the tube, thereby moving the gage needle and recording the pressure on a calibrated scale.

10
GOVERNORS AND ENGINE CONTROLS

A. GENERAL

10A1. Function and types of governors. The purpose of a governor is to control the speed of an engine. If an engine is loaded beyond its rated capacity, it will slow down or may even stop. Governors act through the fuel injection system to control the amount of fuel delivered to the cylinders. The quantity of fuel delivered, in turn, governs the power developed.

The two types of governors, each of which serves a distinctly different purpose, are the overspeed governor and the regulating governor. The overspeed type is used on most marine engines where the speed of the engine is variable. By necessity, the marine engine requires a flexibility in speed due to the maneuvering of the ship. This type of governor is installed as a safety measure and comes into action when the engine approaches dangerous overspeed. This condition could occur before the operator had time to bring the engine under control by other means. The overspeed trip functions only if the regulating governor fails. This governor controls all abnormal speed surges.

Overspeed governors are of the centrifugal type; that is, the force similar to the overspeed type, while the hydraulic type employs a centrifugally actuated pilot valve to regulate the flow of a hydraulic medium under pressure. The mechanical governor is more applicable to the small engine field not requiring extremely close regulation while the hydraulic type finds favor with the larger installations demanding very close regulation. The regulating governor is much more sensitive to slight speed fluctuations than is the overspeed governor. Its duty is to control the speed within very narrow limits when an engine is operating under varying loads. It takes the place of the operator's manual control of the throttle. When the load on the engine increases, and before the engine's speed has appreciably dropped, it permits an increase of fuel to the cylinders, thus maintaining the engine speed at the set rate. To perform this function, the governor must be sensitive to the slightest variation in speed. The Woodward hydraulic governor of the regulating type is widely used in the United States Navy and will be described in detail.

10A2. Submarine shipboard control installations. Each main and auxiliary submarine engine installation includes a regulating and an overspeed governor. Both

action of the governor depends upon the centrifugal force created as the governor weights revolve. Centrifugal force is the force that tends to move a body away from the axis about which it is revolved. This force is transmitted to the fuel injection system by means of levers connected to the governor collar and a linkage system. In some types of overspeed governors the action merely cuts off the fuel until the engine has slowed to a point of safety and then allows the resumption of normal operation. The other type trips a fuel cutout mechanism and effects a complete stopping of the engine. The F-M engines employ an F-M design overspeed governor and the GM engines use Woodward overspeed governors.

For this discussion governors will be classified as either hydraulic or mechanical. The mechanical type embodies the principle of centrifugal

of these governors perform their function by actuating the fuel injection pump controls in some manner. The engines may be stopped at the throttleman's station at the engine or pneumatically by remote control from the control cubicle.

Engine speeds are held uniform by regulating governors whose power mechanism transmits movement to the fuel control shift on the engine. These main engine governors may be controlled at the engine or in the control cubicle. A control cabinet mounted on the main control cubicle instrument panel in the maneuvering room permits remote control of the governors through a Selsyn installation.

The engines are prohibited from exceeding a given maximum allowable speed by the

overspeed governors which are either of hydraulic or centrifugal type and driven off one of the

engine camshafts. The means by which these overspeed trips fulfill their purpose are different for the GM and F-M engines and will be explained later in this chapter.

B. REGULATING GOVERNORS

10B1. Description and operation. The type of regulating governor used on all submarine main engines is the Woodward SI hydraulic type governor. On F-M engines, it is driven from the lower crankshaft, and on GM engines, from one of

setting of the governor; the power element, consisting of the power spring, power piston, and power cylinder; and the compensating assembly which consists of the actuating compensating plunger, the receiving compensating plunger, the compensating spring,

the camshafts. The purpose of the governor is to regulate the amount of fuel supplied to the cylinders so that a predetermined engine speed will be maintained despite variations in load. Figure 10-2 is a schematic diagram of the governor. The principal parts of the governor are a gear pump and accumulators which serve to keep a constant oil pressure on the system at all times; a pilot valve plunger, pilot valve bushing, and flyweights which control the amount of oil going to the power assembly; a speed adjusting spring whose tension governs the speed

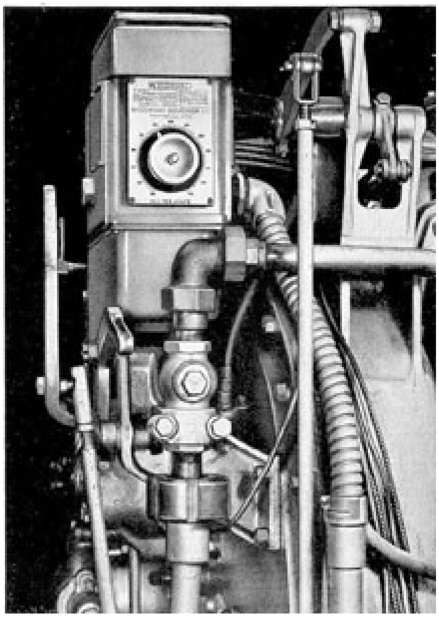


Figure 10-1. Woodward regulating governor installed.

and two compensation needle valves. The pilot valve plunger is constructed with a land which serves to open or close the port in the pilot valve bushing leading to the power cylinder.

In this governor the flyweights are linked hydraulically to the fuel control cylinder. The downward pressure of the power spring is balanced by the hydraulic lock on the lower side of the power piston. The amount of oil below the power piston is regulated by the pilot valve plunger controlled by the flyweights.

When the engine is running at the speed set on the governor, the land on the pilot valve plunger covers the regulating port in the bushing. The plunger is held in this position by the flyweights. However, if the engine load decreases, the engine speeds up and the additional centrifugal force moves the flyweights outward, raising the pilot valve plunger. This opens the regulating port of the bushing, and trapped oil from the power cylinder is then allowed to flow through the pilot valve cylinder into a drainage passage to the oil sump. As the trapped oil drains to the oil sump, the power spring forces the piston down, actuating the linkage to the fuel system controls, and the supply of fuel to the engine is diminished. As the engine speed returns to the set rate, the flyweights resume their original position and the, pilot valve plunger again covers the regulating port.

If the load increases, the engine slows down, and the flyweights move inward. This lowers the pilot

valve plunger, allowing pressure oil to flow through the pilot valve chamber to the power cylinder. This oil supplied by a pump is under a pressure sufficient to overcome the pressure of the power spring. The power piston

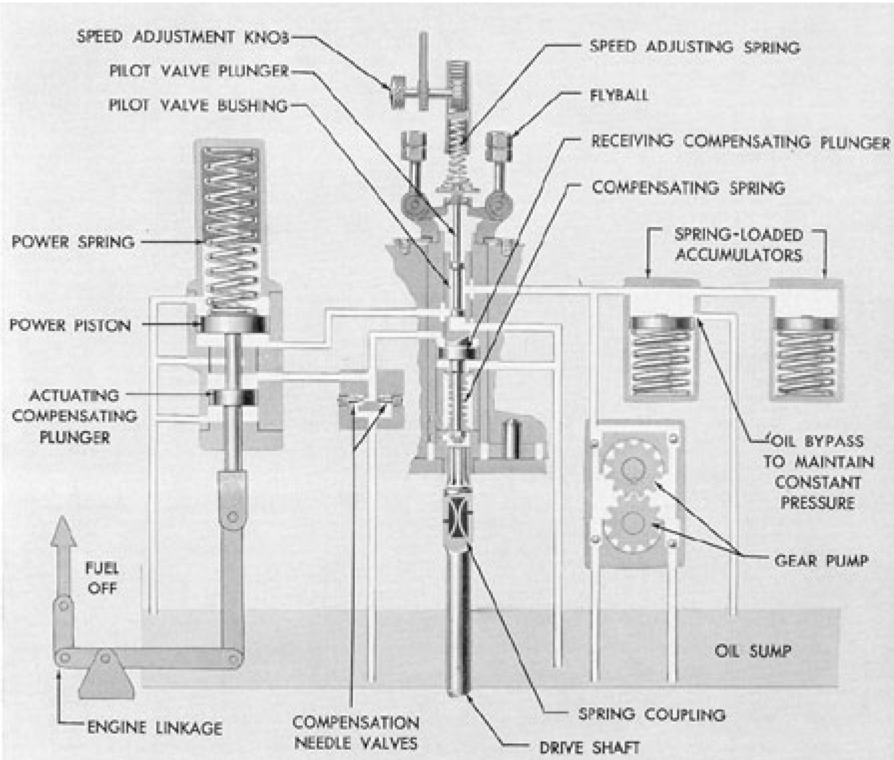


Figure 10-2. Schematic diagram of Woodward regulating governor.

moves upward, actuating the linkage to increase the amount of fuel injected into the engine cylinders. Once again, as the speed returns to the set rate, the flyweights resume their central position. The gear pump that supplies the high-pressure oil is driven from the governor drive shaft and takes suction from the governor oil sump. A spring-loaded accumulator maintains a constant pressure of oil and allows excess oil to return to the sump.

To prevent overcorrection in the regulating governor a compensating mechanism is used. This acts on the pilot valve flyballs would normally direct the pilot valve to cover the port. A compensating plunger on the power piston shaft moves in a cylinder that is also filled with oil. When the engine speed increases and the power piston moves downward, the actuating compensating plunger is also carried down, drawing oil into its cylinder. This creates a suction above the receiving compensating plunger which is part of the pilot valve bushing. The bushing moves upward, closing the port to the power piston. Thus the power piston is stopped, allowing no time for overcorrection. As the flyweights and pilot valve return to

bushing so as to anticipate the pilot valve movement and close the regulating port slightly before the centrifugal their central position, oil flowing through a needle valve allows the compensating

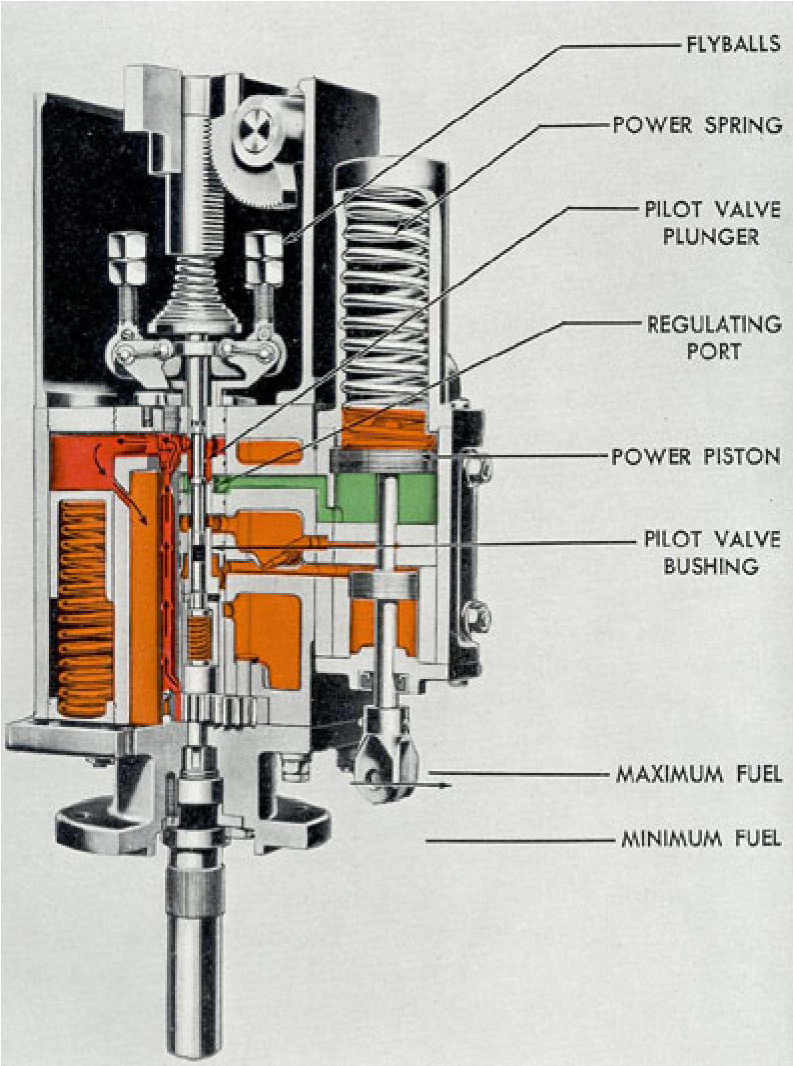


Figure 10-3. Governor cross section-normal speed, steady load.

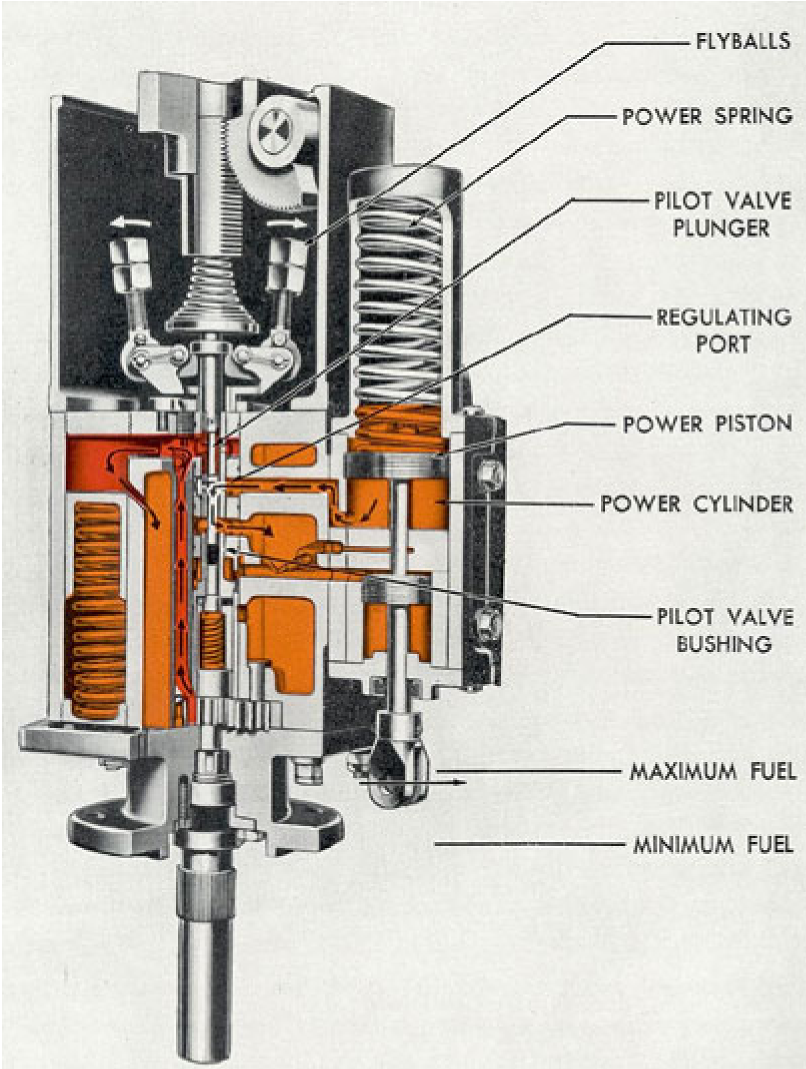


Figure 10-4. Governor cross section-increased speed, decreased load.

spring to return to its central position. To keep the port closed, the bushing and plunger must return to normal position at exactly the same speed. Therefore, the needle valve must be adjusted so that the oil passes through at the required rate for the particular engine.

When the engine speed drops below the set rate, the actuating compensating plunger moves upward with the power piston. This increases the pressure above the actuating compensating plunger and consequently above the receiving compensating piston which therefore moves down, port to the sump. The power spring is thus allowed to move the power piston downward and consequently reduce the fuel supply to the engine, thereby decreasing the engine speed.

The downward motion of the power piston reduces the fuel supply and thereby reduces the engine speed as described above. However, to prevent this reduction from being carried too far, the actuating compensating piston moves down with the power piston as shown in Figure 10-5. This creates an oil suction on the receiving compensating piston which draws up the pilot valve bushing, compressing the

carrying with it the pilot valve bushing. As before, the lower bushing port is closed. The excess oil in the compensating system is now forced out through the needle valve as the compensating spring returns the bushing to its central position.

The governing speed of the engine is set by changing the tension of the speed adjusting spring. The pressure of this spring determines the engine speed necessary for the flyweights to maintain their central position. Oil allowed to leak past the various plungers for lubricating purposes is drained into the governing oil sump.

In actual operation, the events described above occur almost simultaneously.

Figures 10-3 through 10-9 show actual cross sections of the governor for various engine loads and engine speeds. Figures 10-3 through 10-6 illustrate the actual governor operation cycle for a decrease in the engine load. Figure 10-3 shows the governor operating with the engine at normal speed under a steady load. The flyballs, pilot valve plunger, and pilot valve bushing are in normal positions. The regulating port in the bushing is covered by the land on the plunger. Thus the power piston is held stationary by the trapped oil.

Figure 10-4 shows the governor acting in response to a load decrease and a consequent increase in speed. As the speed increases, the fly balls move outward, raising the pilot valve compensating spring. Movement of the power piston and pilot valve bushing continues until the lower or regulating port in the bushing is covered by the land on the pilot valve plunger. As soon as the regulating port is covered, the power piston is stopped at a position corresponding to the decreased fuel needed to run the engine at the reduced load.

As the speed decreases to normal, the flyballs return to their normal position, thus lowering the pilot valve plunger to its normal position as shown in Figure 10-6. To keep the regulating port closed while the plunger is being returned to normal position, the bushing must move downward at the same rate as the plunger. This is done by the compensating spring. The flow of oil through the needle valve determines the rate at which the compensating spring is able to move the bushing. Thus, it can be seen that accurate governing is dependent on a proper adjustment of the needle valve since any opening in the regulating port during this phase of the cycle would permit the power piston to move, thereby causing an undesirable change in the fuel supply.

At the completion of the cycle, the flyballs, pilot valve plunger, and pilot valve bushing have returned to normal position. The power piston is stationary, held by trapped oil, in a position corresponding to the decreased fuel needed to run the engine at normal speed under a decreased load.

Figure 10-7 shows the governor acting in response to an increase

plunger so that its land uncovers the lower or regulating port in the pilot valve bushing. This releases the trapped oil from the power cylinder and permits it to flow through the regulating in load with a resulting decrease in engine speed. As the speed decreases,

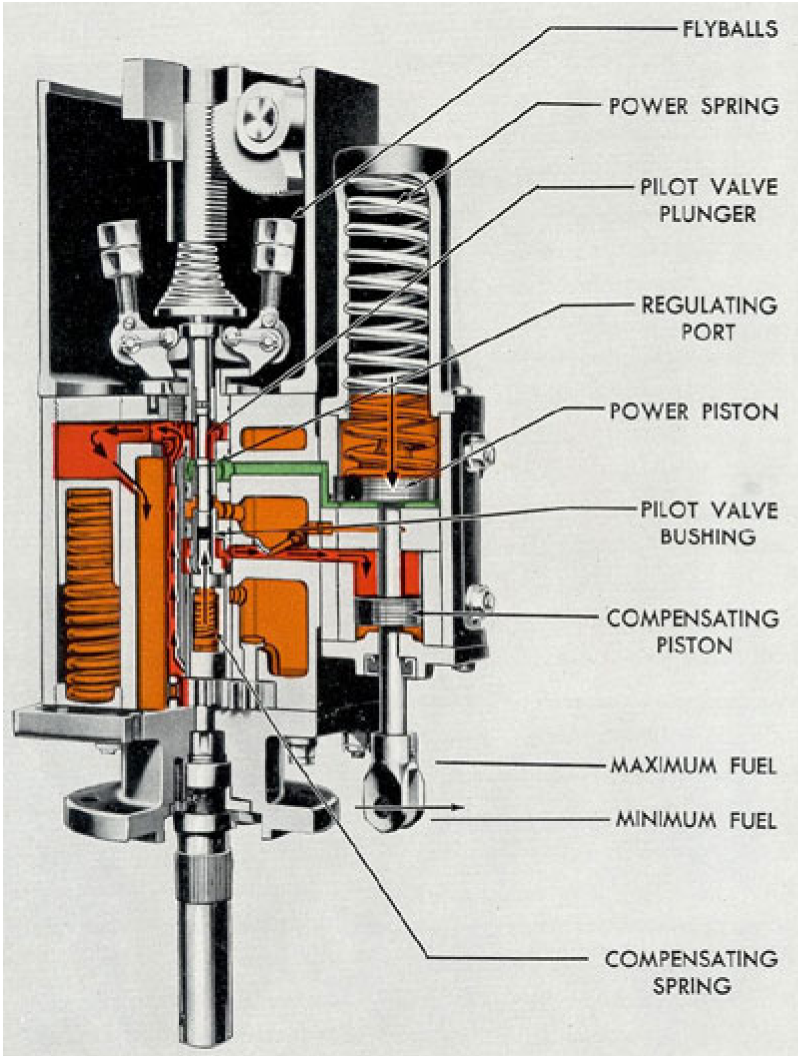


Figure 10-5. Governor cross section-normal speed, decreased load.

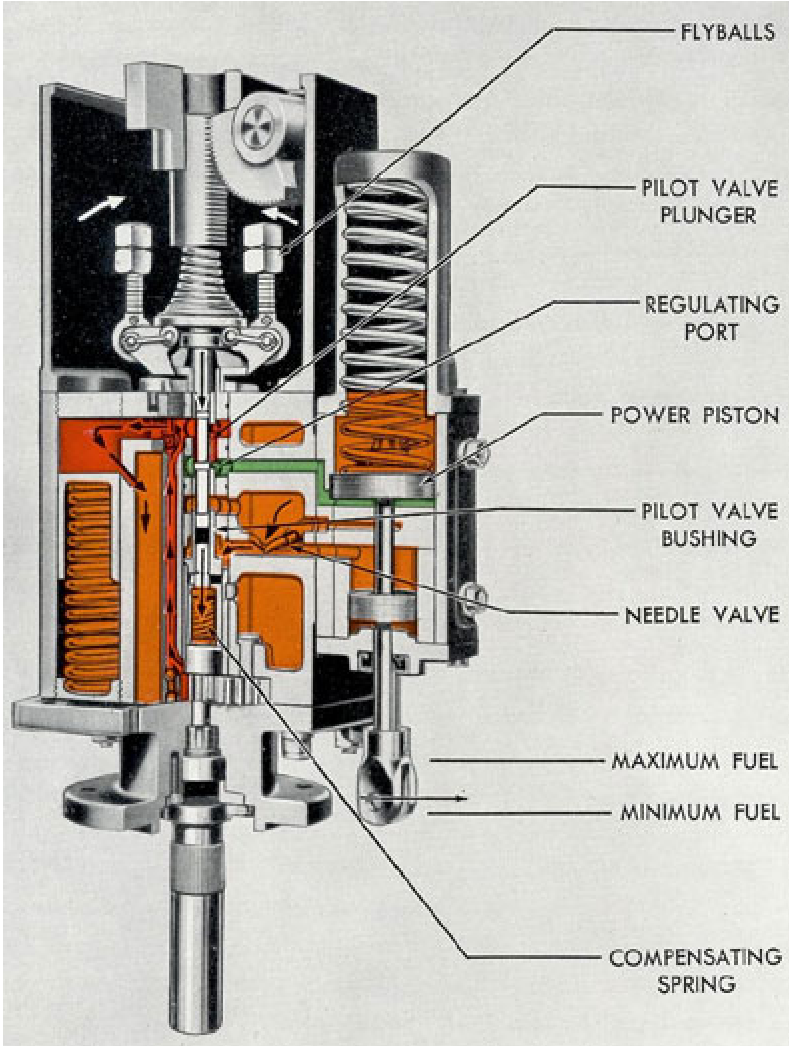


Figure 10-6. Governor cross section-normal speed, new load.

the flyballs move inward, lowering the pilot valve plunger and uncovering the regulating port in the pilot valve bushing. Thus, pressure oil from the pump and the accumulators is admitted to the power cylinder, causing the power piston to move up and increase the flow of fuel.

As the power piston moves up (Figure 10-8), the actuating compensating piston also moves up, causing oil pressure on the receiving compensating piston and thereby forcing the pilot valve bushing down, compressing the compensating spring. Movement of the power piston and the pilot valve

The drive shaft assembly is flexible in order to keep from the governor, as far as possible, the inherent vibrations of the camshaft from which the governor is driven. This shaft is so constructed that the power required to drive the governor is transmitted from the serrated drive sleeve through the drive pin to the lowest section of the plug, and from the lower section through leaf springs to the upper section of the drive shaft. The governor drive is made positive, even if the springs should break, by the construction of the two sections of the shaft. Each section is cut with a projection on the end. In the event of leaf spring failure, these projections will make

bushing continues until the regulating port in the bushing is covered by the land on the pilot valve plunger. As soon as the regulating port is covered, the power piston is stopped (oil being trapped under the piston) at a position corresponding to the increased fuel needed to run the engine at normal speed under an increased load.

As the speed increases to normal, the flyballs return to their normal position, raising the pilot valve plunger back to its normal position (Figure 10-9). The pilot valve bushing is returned to its normal position by the compensating spring at the same time and rate as the pilot valve plunger. This keeps the regulating port covered by the land on the plunger, thus keeping the power piston stationary. The flow of oil through the needle valve determines the rate at which the bushing is returned to normal. At the completion of the cycle, the flyballs, pilot valve plunger, and pilot valve bushing are in their normal position. The power piston is stationary at a position corresponding to the increased fuel needed to run the engine at normal speed under the increased load.

10B2. Regulating governor sub-assemblies. The governor consists of five principal subassemblies as follows:

a. Drive adapter. The drive adapter assembly serves as a mounting base for the governor. The upper flange of the casting is bored out at the center to form a bearing surface for the contact and continue to drive the governor.

b. Power case assembly. This assembly includes the governor oil pump, oil pump check valves, oil pressure accumulators, and compensating needle valves.

The oil pump drive gear turns the rotating sleeve to which it is attached. The pump idler gear is carried on a stud and rotates in a bored recess in the power case. These two gears and their housing constitute the governor oil pump. On opposite sides of the central bore in the power case, and parallel to it, are two long oil passages leading from the bottom of the power case to the top of the accumulator bores. Check valve seats are arranged at the top and bottom of each chamber. Both check valves have openings leading from the space between the valves to the oil pump. In this way the pump is arranged for rotation in either direction, pulling oil through the lower check valve on one side and forcing it through the upper check valve on the opposite side.

There are two oil pressure accumulators. Their function is to regulate the operating oil pressure and insure a continuous supply of oil in the event that the requirements of the power cylinder should temporarily exceed the capacity of the oil pump. There is no adjustment for oil pressure, as this pressure is determined by the size of the springs in the accumulators.

The two compensating needle valves control the size of the openings in the two small

hub of the pump drive gear and for the upper end of the drive shaft.

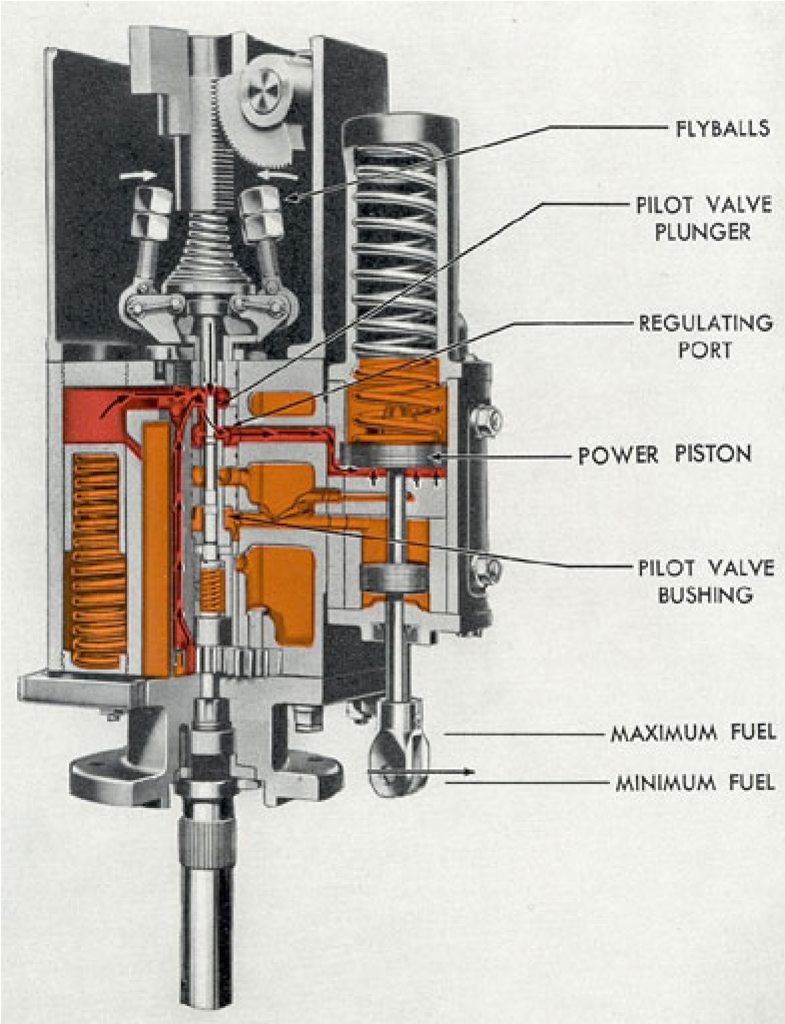


Figure 10-7. Governor cross section-decreased speed, increased load.

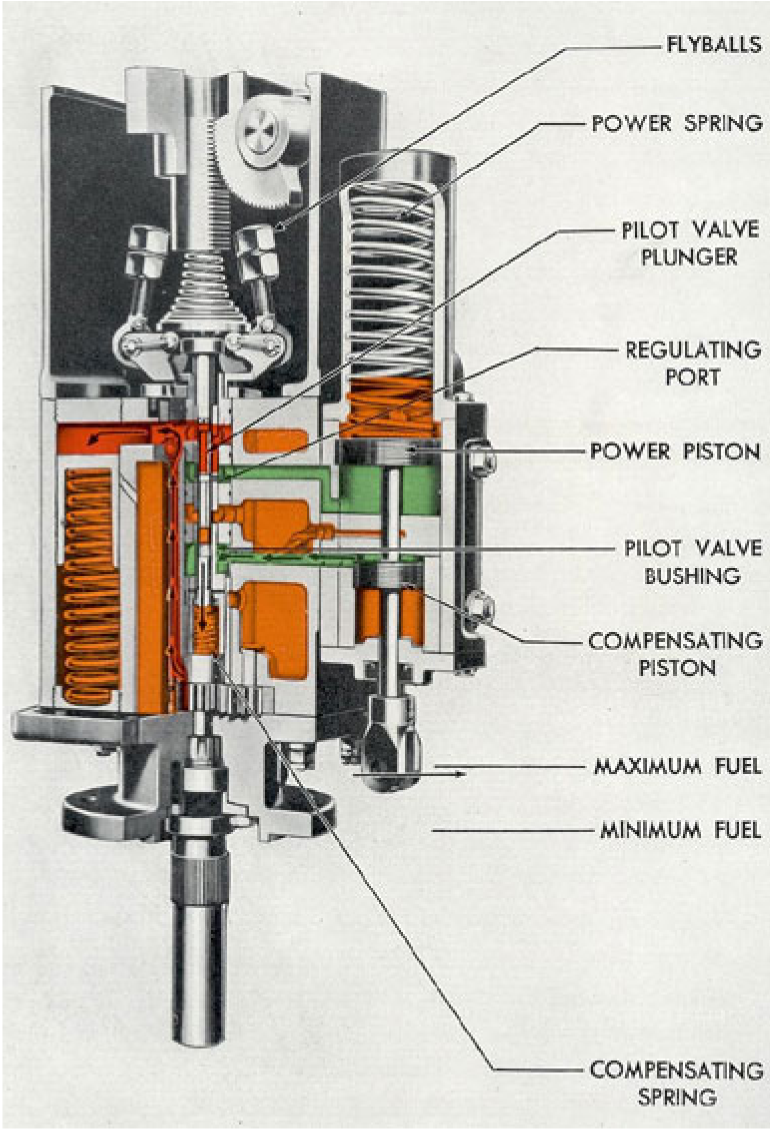


Figure 10-8. Governor cross section-normal speed, increased load.

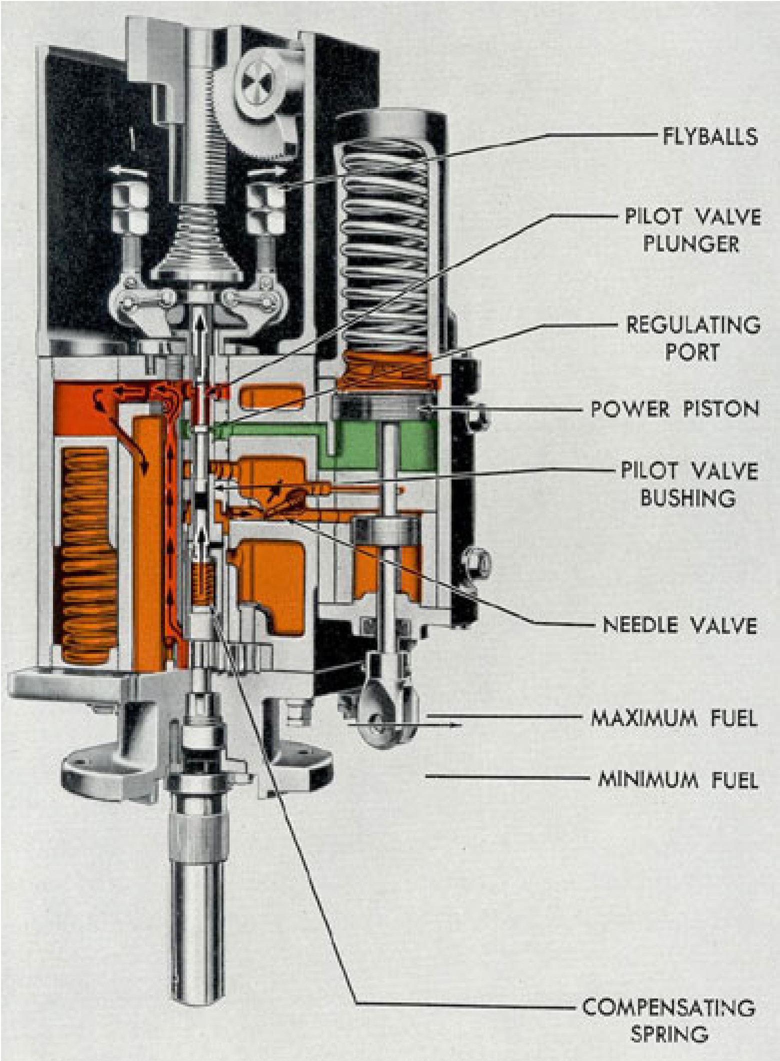


Figure 10-9. Governor cross section-normal speed, new load.

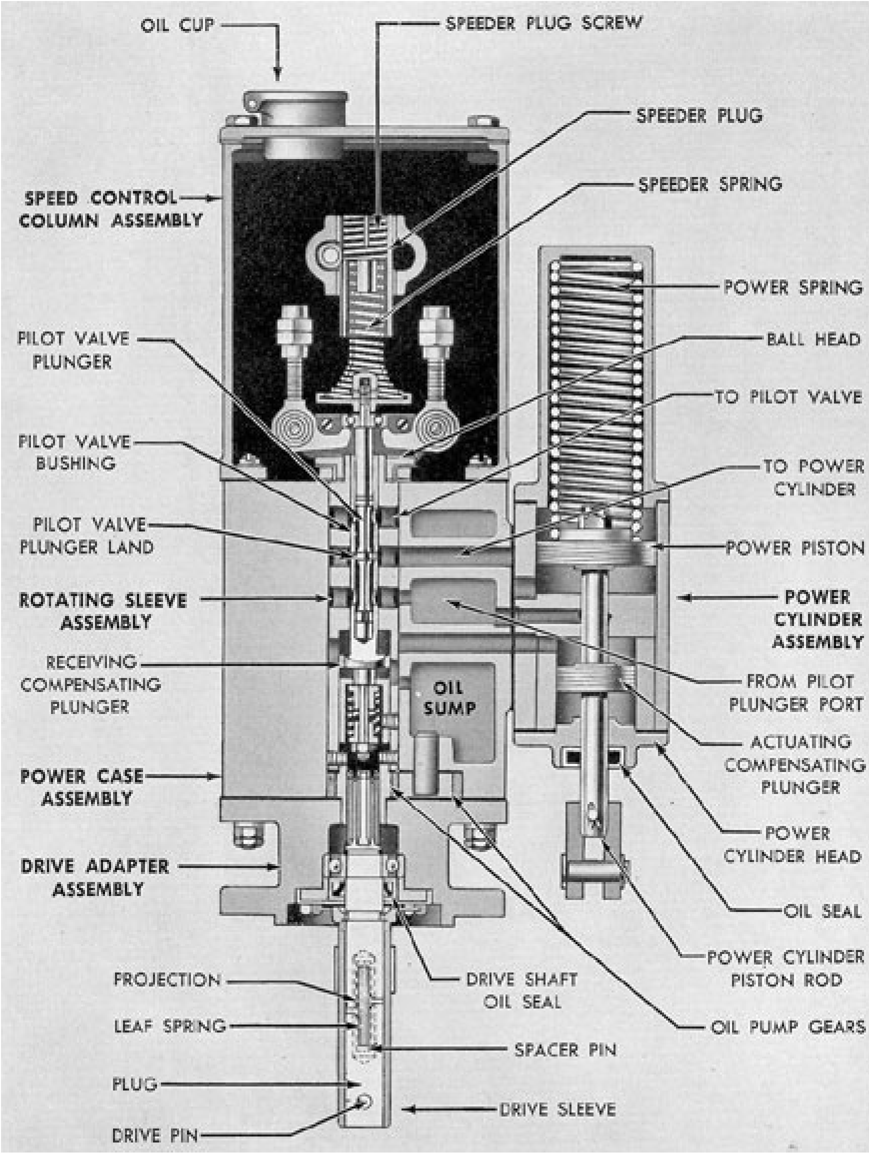


Figure 10-10. Governor-sections through adapter, power, case, power cylinder and rotating sleeve assembly.

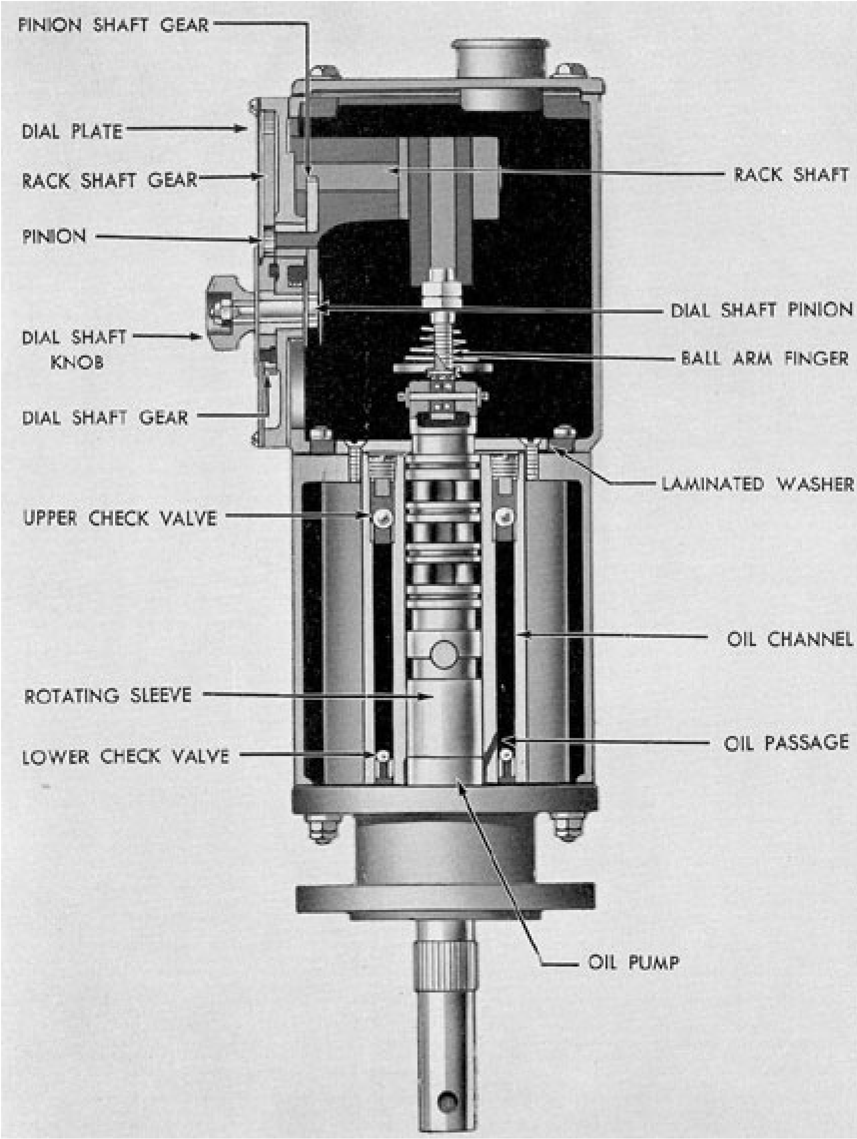


Figure 10-11. Governor-section through speed control column.

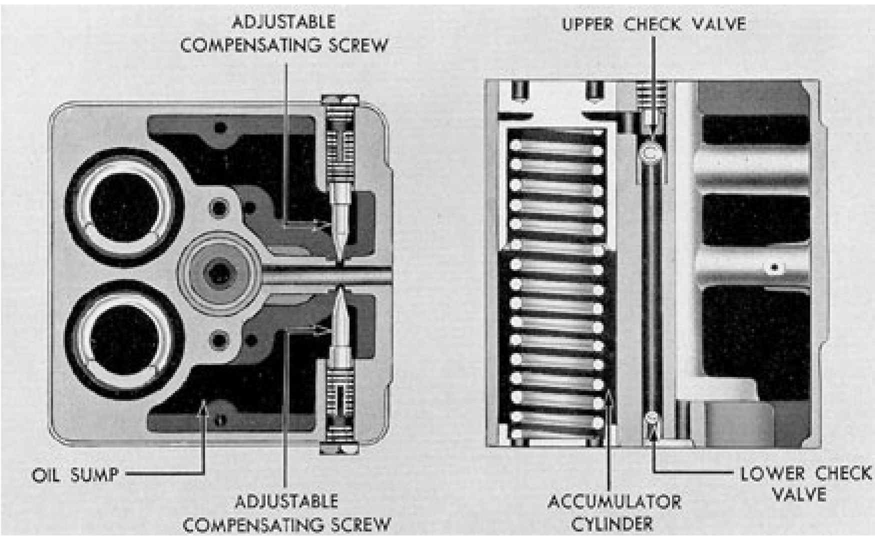


Figure 10-12. Governor-section through accumulator cylinder.

tapered ports in the passage that connects the area above the actuating compensating plunger in the Servo motor and the space sump. No piston rings are used in the closely fitting piston. A shallow, helical groove permits equal oil pressure on all sides of

above the receiving compensating plunger in the pilot valve bushing of the rotating sleeve assembly. These ports open the compensating oil passage to the oil sump tank. Only one needle valve and one port are necessary for operation, but two are provided so that adjustment can be made on the one that is more accessible.

c. **Power cylinder assembly.** The power cylinder assembly consists of the cylinder, power piston, piston rod, power spring, and the actuating compensating plunger. The power piston is single acting. Any oil pressure acting on the lower side forces the piston up against the power spring, thereby increasing the fuel flow. If no oil pressure is present, the power spring acting on the upper side forces the piston down to decrease the fuel flow.

The area underneath the power piston is connected to the pilot valve regulating ports.

An oil drain is provided in the space above the power piston to permit any oil that may leak by the piston to drain into the governor case oil the piston, thus preventing wear and binding.

An adjustable load limit stop screw is provided in the power cylinder. This screw prevents the power piston from traveling beyond the predetermined load limit. The screw can be adjusted by removing the cap nut on top of the power cylinder, loosening the lock nut, and turning the screw up or down with a screwdriver.

d. Speed control column. The basic speed control column assembly includes the speeder plug screw, speed adjusting spring, rack shaft, rack shaft gear, and the speed adjustment knob with gear train. The gear train consists of the dial shaft gear, dial shaft pinion, and the pinion shaft gear and pinion. Movement of the gear train changes the compression of the speed adjusting spring. The amount of compression determines the speed at which the flyballs will be vertical. Hence, the compression determines the engine speed. The speeder plug screw allows the adjustment of the governor speed setting to

match the actual speed of the engine.

e. Rotating sleeve assembly. The principal parts of the rotating sleeve assembly (Figure 10-13) are: the pump drive gear, pilot valve bushing, pilot valve plunger, ballhead, and flyballs. The central bore in the power case forms a bearing for the compensating plunger.

In the 1/2-inch and 1-inch diameter bores in the rotating sleeve are the pilot valve bushing and receiving compensating plunger, the compensating spring retainer, two compensating spring collars, compensating spring, and adjusting nut.

entire rotating sleeve. The port grooves in the sleeve align with the ports in the power case (Figure 10-10). Since these grooves extend completely around the diameter of the rotating sleeve, the results are the same as if the sleeve were stationary and the ports were permanently in line with those in the case. From top to bottom the ports are as follows: accumulator pressure to pilot valve, regulating pressure to power cylinder, drain from the lower end of the pilot plunger, compensating pressure from the power piston to the receiving compensating plunger on the pilot valve bushing, and drain from the lower side of the receiving

The nut is threaded on the stem at the lower end of the pilot valve bushing just tightly enough so that the compensating spring is slightly compressed between the collars, and so that the dimension between the outer faces of the spring collars exactly equals the depth of the 1-inch hole in the spring retainer. With the nut in this position, the face of the lower spring collar will be flush with the lower end of the compensating spring retainer and the upper end of the pump drive gear, and there will be no movement of the pilot valve bushing without compression of the spring.

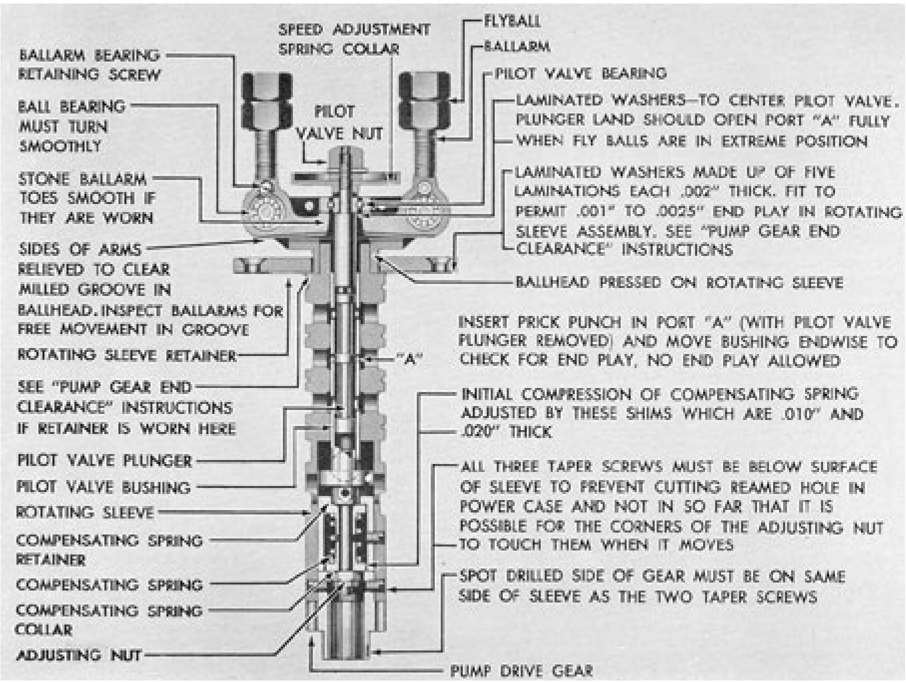


Figure 10-13. Governor-rotating sleeve assembly.

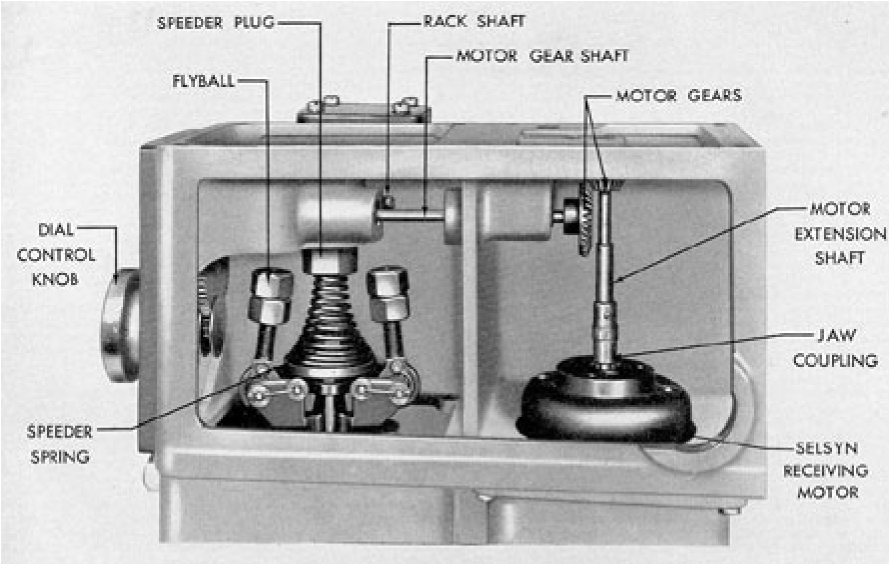


Figure 10-14. Governor-speed control mechanism.

The pilot valve plunger land slightly overlaps, the regulating ports in the valve bushing. Therefore any slight movement of the valve will produce a corresponding power piston movement.

The ball bearing clamped between the spring collar and the upper shoulder serves as a support for the ballarm fingers. Mounted on ball bearings, the flyballs are free to move at the slightest change in speed, and their motion is transmitted to the pilot valve through the horizontal fingers on the ballarms.

10B3. Adjustments. a. Speed adjustment. The speed setting of the governor is changed by increasing or decreasing the compression of the speed adjusting spring which opposes the centrifugal force of the flyballs. Increasing the spring compression will make it more difficult for the flyballs to move outward; consequently a higher flyball (and engine) speed must be attained to move the flyballs outward and thereby reduce the fuel supply.

Conversely, decreasing the compression of the speed adjusting spring will permit the flyballs to move outward when they, and the engine, are running at a lower speed. Thus, decreasing the spring compression decreases the speed at which the engine will run.

Speed adjustments may be made manually at the governor, or electrically from the governor control cabinet in the maneuvering room as follows:

1. Manual adjustment. The manual adjustment is made by means of the speed control knob located on the front of the regulating governor. This knob is connected through a gear train to the rack shaft which in turn is- geared to a rack on the speed adjusting plug. The knob also actuates a pointer that travels over a dial graduated to show engine speeds corresponding to deflection of the speed adjusting spring.

2. Electrical adjustment. For electrical control, a Selsyn receiving motor is also geared to the rack shaft. This receiving motor operates in parallel with a Selsyn transmitter generator in the governor control cabinet mounted on the main control cubicle instrument panel in the maneuvering room. When the speed setting is changed at the transmitter generator, the receiving motor in the governor moves to establish the same setting in the governor.

b. Compensating needle valve adjustment. This adjustment is made with the engine running from 200 rpm to 300 rpm as set by the speed adjustment knob or by remote control.

Either of the two needle valves may be used for adjustment. The one not used must be turned in against its seat. When performing the adjustment, the more accessible valve is opened a full turn or more and the engine is allowed to surge for approximately 30 seconds to eliminate trapped air. Then the valve is closed until surging is just eliminated.

The needle valve will usually be open about one-fourth of a turn for best performance. However, the adjustment depends on the characteristics of the engine. The needle valve should be kept open as far as possible to prevent sluggishness. Once the valve has been adjusted correctly for the engine, it should not be necessary to change the adjustment except for a

2. Remove the clevis pin from governor link and power piston tail rod connection.

3. Remove the nuts that hold the governor to the governor and tachometer drive housing.

4. Lift the governor straight up, being careful not to damage the splined shaft.

With the governor removed from the engine, remove the cover, turn the governor upside down, drain and flush thoroughly with clean light-grade fuel oil or an approved solvent solution to remove any foreign matter. Drain thoroughly, flush and refill with clean lubricating oil. Follow the above procedure whenever the governor is removed from the engine for any reason.

If it is not possible to shut down long enough to remove the governor from the engine, drain the oil from the governor by removing one of the plugs in the lower part of the power case. Fill with fuel oil and run for approximately 30 seconds with the needle valve open. Then drain and refill with clean lubricating oil.

b. Oil seals. When it becomes necessary to add oil to the governor too frequently, the oil seals should be replaced. To replace the drive shaft oil seal, remove the lockwire and capscrews that secure the drive shaft assembly to the base, then pull the assembly out of the base. Remove the snap ring and press the drive shaft out of the bearing. Remove the bearing retainer and press out the oil seal. Carefully

permanent temperature change affecting the viscosity of the oil.

10B4. General maintenance and internal adjustments. a. Oil changes. The governor oil must be clean and free of foreign particles. Under favorable conditions the oil may be used for approximately 6 months without changing. If adjustment of the compensating needle valve does not result in proper operation, dirty oil may be the cause of the trouble.

To change the oil, remove the governor from the engine as follows:

1. Disconnect all electrical connections to the governor. Tag the wires and connecting points to make certain connections will be properly replaced.

press the new seal into the retainer and reassemble the unit. Make sure the lip of the new seal faces upward.

To replace the piston rod oil seal, remove the power cylinder from the governor. Drive out the tapered pin and press the piston rod out of the rod end. Remove the cylinder head, pry out the oil seal and press a new seal into position making certain that the oil seal lip faces upward. Reassemble the unit being careful not to damage the lip of the new seal.

c. Ballarms and bearings. Erratic governor performance may indicate the need for replacement of ballarms, ballarm bearings, or pilot valve plunger bushing.

If the toes of the ballarms are worn too. badly to be refinished, new ballarms should be installed. Set the flyballs at the same position on the new ballarms as on the old ones. Ballarm bearings should be replaced if worn excessively. If the ballarm pins do not fit tightly in the inner race of the ballarm bearings, they should be interchanged with the ballarm stop pins.

If the pilot valve plunger bearing is grooved, it should be either turned over or replaced. Extreme care must be used in disassembling the pilot valve plunger assembly, to avoid damaging the ground finish. After disassembly and reassembly of the pilot valve uncovered for both inner and outer positions of the ballarms.

Movement of the regulating land on the plunger can be observed through the regulating port in the bushing while holding the plunger assembly against the toes of the ballarms and moving the ballarms through their full travel. The amount of port opening for inner and outer positions of the flyballs should be the same and correct within .005 inch. Openings need not be completely uncovered at each extreme. If the regulating port is not fully uncovered at each end of ballarm travel, the position of the plunger in relation to the ballarms can be changed by varying the washer thickness under the bearing on the plunger.

plunger assembly, the pilot valve adjustment should be checked.

d. Pilot valve adjustment. The pilot valve adjustments should be checked after doing any work on flyballs, pilot valve plunger, or pilot valve bushing.

The regulating port should be completely

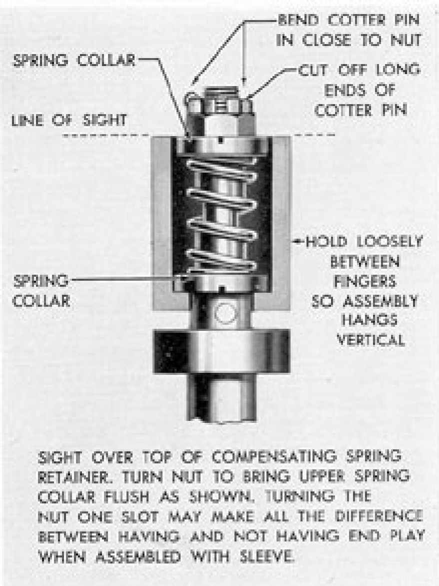


Figure 10-15. Governor-measurement of precompression.

Removing one layer from the laminated washer will raise the plunger a distance of 0.002 inch.

e. Pump drive gear end clearance. Pump drive gear end clearance is determined by the thickness of the laminated washers under the

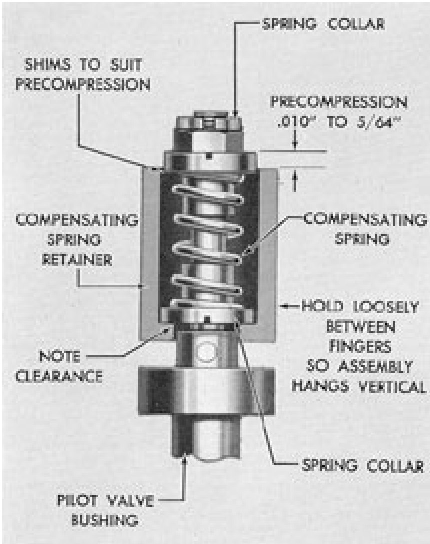


Figure 10-16. Governor adjustment of compensating spring length.

rotating sleeve retainer. To obtain proper end clearance of the pump drive gear, remove one lamination at a time from the washer under each end of the retainer until the rotating sleeve assembly turns hard, then replace one lamination under each end. The clearance should be from 0.001 to 0.003 inch. Insufficient end clearance will cause wear and possible seizure. Excessive clearance will reduce pump capacity.

After the laminated washers have been completely removed, due to repeated adjustment, the

dial plate, dial shaft nut, speed adjusting knob, and dial disk. While doing this, place a finger against the inside end of the dial shaft to prevent its being forced through the bushing by the dial shaft spring. Replace the knob and nut. Pull the gear forward, unmeshing it from the pinion.

Start the engine and turn the speed adjusting knob to the desired maximum (or minimum) speed. Remesh the gear in position where its maximum (or minimum) stop pin is against the pin in the dial panel. Stop the

retainer should be replaced. To replace the retainer, remove the rotating sleeve assembly from the power case and press the sleeve out of the ballhead. Reassemble the unit using a new retainer and new laminated washers. Adjust pump gear end clearance as before.

f. Compensating spring adjustment. Compensating spring adjustment should not be made without first making the compensating needle valve adjustment and changing the oil. Then, if operation is still not satisfactory, remove the tapered screws and pull out the drive gear and pilot valve bushing assembly. Back off the adjusting nut and change the precompression on the compensating spring.

This precompression may vary from 0.010 to 0.078 inch depending upon engine characteristics and load. To eliminate a slow engine hunt, remove shims to reduce precompression. To eliminate a surge, add shims to increase precompression.

Adjust compensating spring length and reassemble with the rotating sleeve and the drive gear. Check for end play. None is allowed.

g. Speed limit adjustment. Speed limit adjustment must be made only after it has been determined that the engine linkage is in proper adjustment. If the desired maximum or minimum engine speed cannot be obtained by turning the speed adjusting knob, the limits can be changed engine, remove the nut and knob, and reassemble all parts.

Note whether the engine speed, as shown by the tachometer, corresponds to that shown on the governor dial. If not, recheck the speed limit adjustment.

10B5. Governor control cabinets. The purpose of the control cabinet is to permit adjustment of the speeds of any engine or any

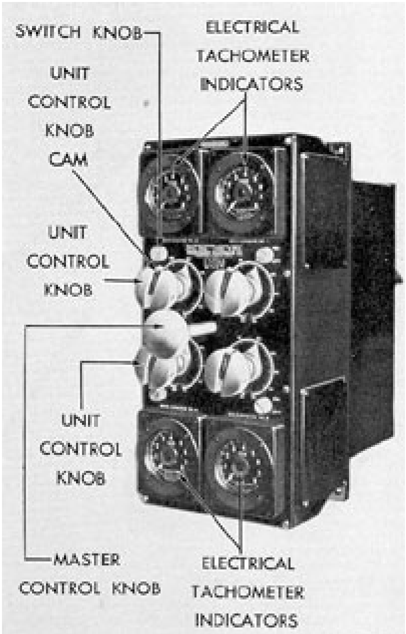


Figure 10-17. Governor control cabinet.

by turning the speed adjusting plug screw. If the limits cannot be changed sufficiently by adjusting this screw, or if the adjusting plug is not equipped with such a screw, the adjustment can be made by changing the position of the stop pins with respect to the speed adjusting plug.

With the engine shut down, remove the

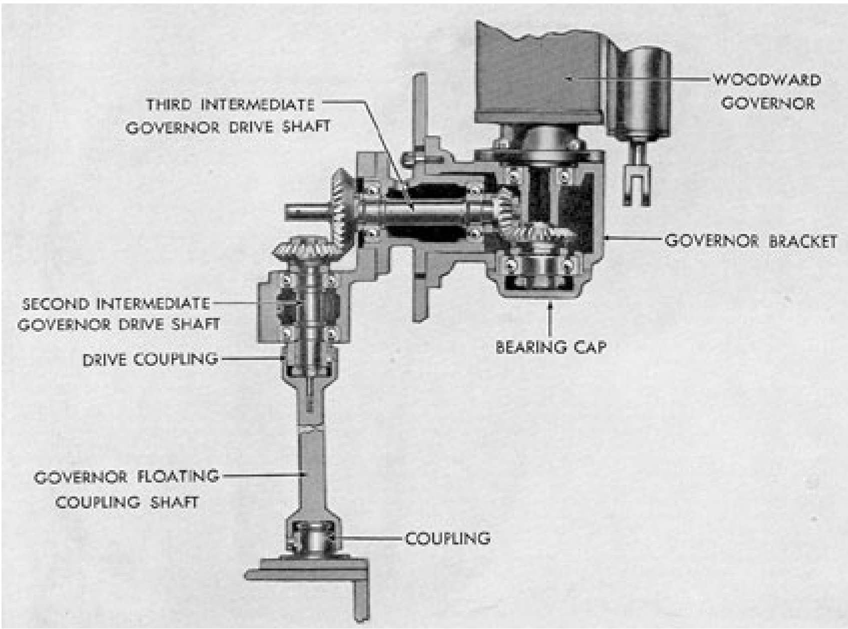


Figure 10-18. F-M governor drive.

combination of two or more engines from the maneuvering room. There are two type of cabinets. The common type (Figure 10-17) is a single unit that can control all four engines. The split type found on some late fleet type submarines is composed of two units, each of which normally controls two engines, but one unit may, through a switching arrangement, control a maximum of three or a minimum of one engine. The control cabinets also mount electrical tachometer indicators.

causes the receiving motor, which is geared to the rackshaft, to change the governor speed setting to that set on the knob on the control cabinet.

The speed setting of any of the four governors can be changed independently through a unit control knob for each governor. For independent operation the cams on the knobs must be in the latched position.

Any or all of the four unit control knobs can be operated simultaneously and identically by

The common type control cabinet contains four Selsyn transmitter generators, one for each main engine. Each of these transmitters is connected by three wires to a Selsyn receiving motor in the regulating governors on the engines. When a transmitter generator rotor is turned by means of a knob on the control cabinet, the phase relationship between the transmitter generator and its receiving motor is disturbed. This the master control knob at the center of the control panel. To permit simultaneous operation, it is necessary to unlatch the cams on the desired unit control knobs first. Next set the speed of the engines together by means of the unit control knobs as indicated by the tachometers, then relatch the cams. This meshes the gears that link each unit control knob to the master control knob.

In the split type installation, two control cabinets are installed to control the four main engines. Normally one is used for control of the two port engines and the other for control of the two starboard engines.

By means of switches, it is possible to surrender control of one engine over to the other control cabinet. Any one cabinet may control a maximum of three engines or a minimum of one.

Each control cabinet contains two direct current position motors, one for each of the engines on the same side of the ship as the control cabinet, and one master position motor which may be electrically interconnected with the other control cabinet, or mechanically with the position motors of its own pair of engines.

C. GOVERNOR DRIVES AND OVERSPEED GOVERNORS

10C1. Governor drives. a. F-M flexible drive. The governor and the fresh and salt water circulating pumps, as well as the lubricating and fuel oil pumps, are driven from the lower crankshaft through a flexible gear drive. The governor drive (Figure 10-18) transmits power from the coupling at the top of the flexible pump drive to rotate the regulating governor. The coupling shaft is designed to float between the pump drive in order to absorb vibration that might pass through the gear train from the lower crankshaft.

The ball bearings and gears of the governor drive are lubricated with oil thrown off from the timing chain in the control end compartment.

b. GM governor and tachometer drive. On the GM governor and tachometer drive (Figure 10-19), the governor is driven through bevel

and the second intermediate
drive shaft,

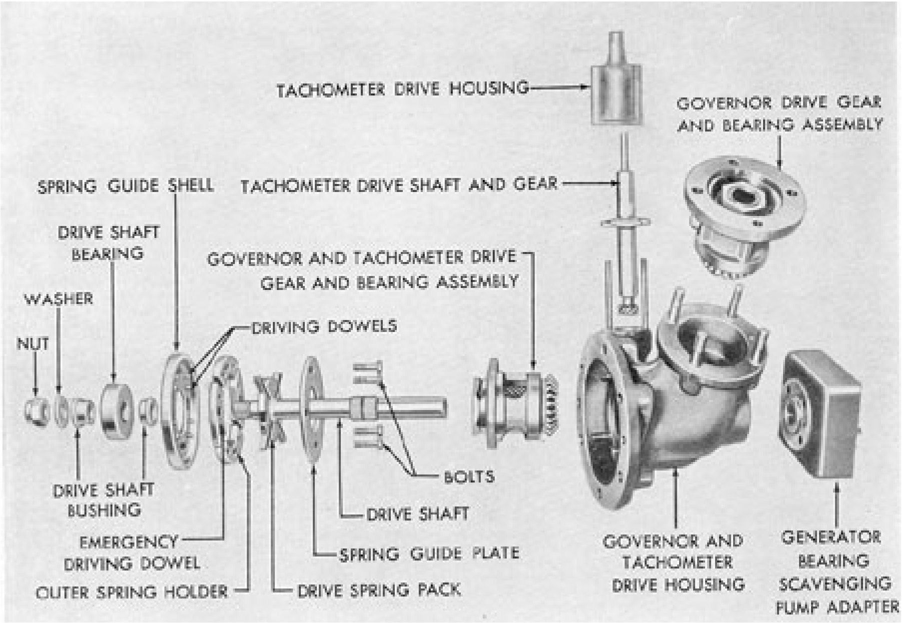


Figure 10-19. Governor and tachometer drive, GM.

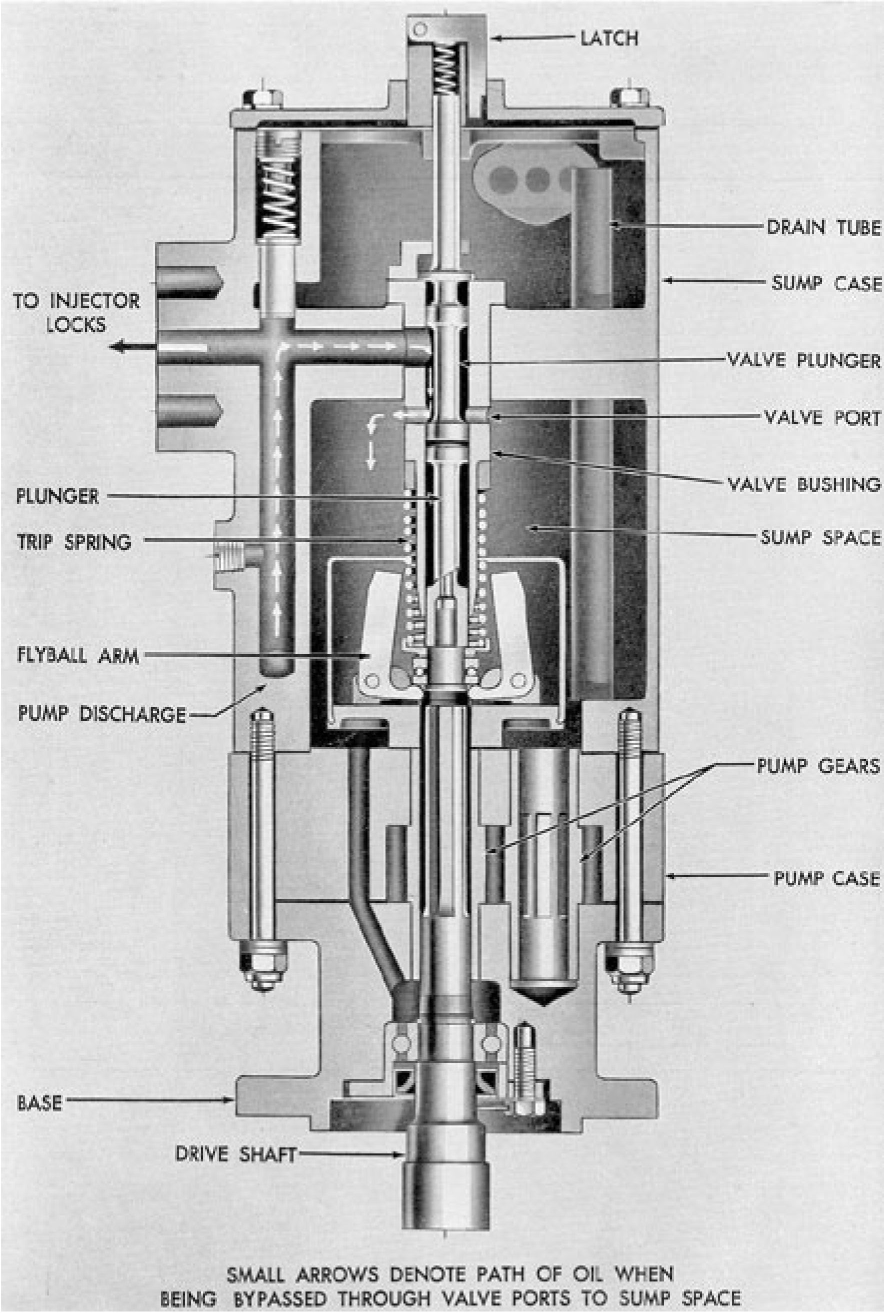


Figure 10-20. GM hydraulic Type overspeed governor.

gears mounted in a housing on the camshaft drive housing. The drive shaft is driven from the camshaft gear through a flexible radial leaf spring type coupling. This shaft drives a bevel gear, through serrations in the shaft, which in turn drives a mating gear, the hub of which is serrated to fit the governor drive shaft. The drive gear is housed in a carrier which also includes a worm gear for the tachometer drive. The entire assembly is

10C2. Overspeed governors. a. Function. Overspeed governors are provided to shut off fuel to the engine in the event of regulating governor failure, jammed linkage, or any other cause that may prevent reduction of the fuel flow by normal methods.

b. GM hydraulic type overspeed governor. The hydraulic type overspeed governor is similar to the regulating governor. It also employs the centrifugal force of a

enclosed in a bracket housing bolted to the camshaft drive housing and is lubricated with oil flowing through the center of the camshaft. The generator bearing scavenging pump, if used, is also driven through this assembly.

In the event of spring failure in the flexible coupling, a dowel pin in the driving member will come into contact with the side of a wide groove in the driven member and thus continue to transmit power.

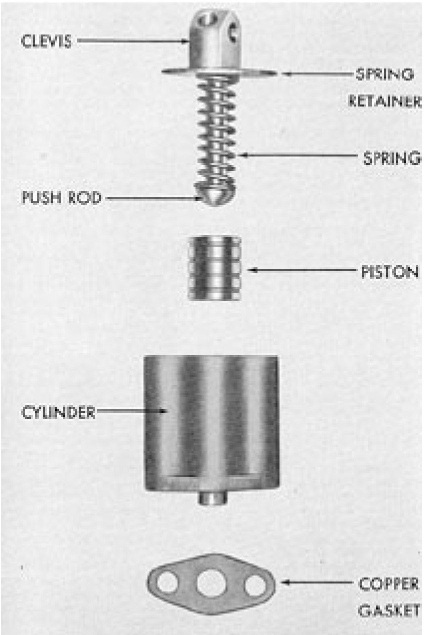


Figure 10-21. GM overspeed shutdown Servo motor.

pair of flyballs acting against the pressure of a spring raising and lowering a plunger (Figure 10-20). The plunger regulates the ports of a gear pump which can, under certain conditions of engine speed, provide oil under pressure to a small Servo motor at each injector rocker arm, causing the Servo motors to stop the function of the injectors.

The overspeed governor is driven from the camshaft of the engine. The drive shaft of, the governor drives the gear pump and the flyballs. At normal engine speeds the flyballs are not acted upon by sufficient centrifugal force to raise the plunger, therefore, the drain bypass ports in the valve bushing remain open. Oil discharged from the pump then flows through a passage in the side of the case into the space surrounding the plunger and through the valve ports into the sump space. The oil level is held at the top of the drain tube by metering the engine lubricating oil that flows into the case. An oil passage between the drive shaft housing and the sump space in the case prevents pressure from being built up by oil leaking from the pump into the housing. Such pressure would blow out the oil seal below the ball bearing.

When the engine reaches its maximum allowable overspeed of 107 percent of normal full speed, the flyballs move outward from the normal center of rotation, raising the plunger against the force of the trip spring to close the drain bypass ports. Pressure built up by the pump forces the oil through a tube to the top pipe in the multiple manifold assembly on

each cylinder bank. The oil flows from the manifold, through tubing, to a passage in the cylinder head and under the piston of a Servo motor attached to each cylinder head. The Servo motor actuates a lever that holds down the injector rocker arm. This action holds the cam roller on the injector

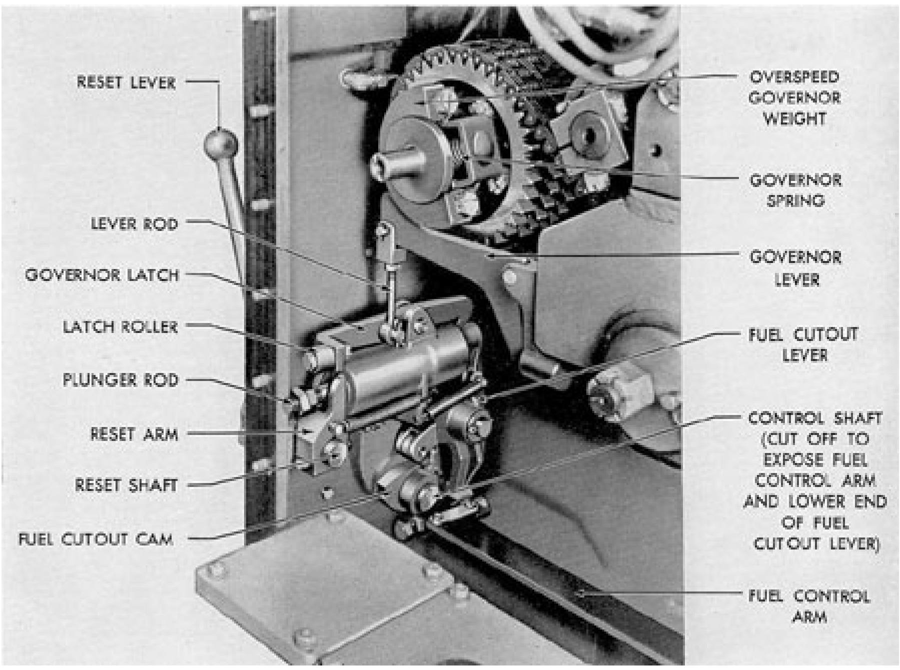


Figure 10-22. F-M overspeed governor and emergency stop mechanism.

rocker arm clear of the injector cam on the camshaft and prevents the injector from operating.

The overspeed governor is equipped with a latch on top of the governor case. This latch holds the valve plunger in position to keep the ports covered and therefore to keep the fuel injectors locked. The latch must be reset manually by the operator before the engine can again be started. This is accomplished by releasing the latch and by pushing the valve plunger into the open position.

centrifugal forces overcomes spring pressure, and the weight moves outward, forcing the overspeed governor lever and its rod downward. This motion trips the overspeed governor latch and permits the plunger spring to force the plunger rod against the fuel cutout lever. This lever then moves the fuel control arm to the no fuel position, stopping the engine.

10C3. F-M manual emergency stop and reset lever. The injection of fuel can be stopped by means of the emergency stop push button which extends through the control end cover near the reset lever. The button acts through

A relief valve in the governor casing allows the pump discharge pressure to be relieved if it exceeds a given set value.

c. F-M mechanical type overspeed governor. The mechanical type overspeed governor consists essentially of a single weight and a spring. The spring is adjusted with shims to prevent the weight from moving until the maximum safe engine speed is reached. When this occurs, linkage and the emergency stop shaft cam to depress the latch roller (Figure 10-22) and thus trip the overspeed governor latch. The result is the same as that obtained by moving the control shaft lever to the STOP position, thus causing the fuel cutout cam on the control shaft

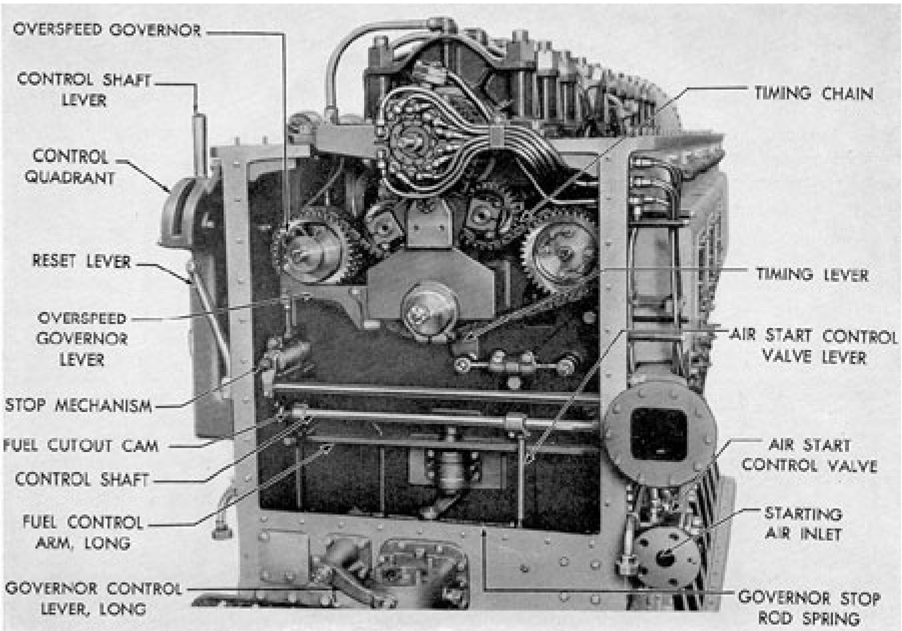


Figure 10-23. F-M control shaft and control mechanism.

to move the injection pump control rod to the no fuel position.

When the engine has been stopped, either by the emergency stop or the overspeed governor, it cannot be started again until the over speed stop plunger has been returned to its normal spring-load position. This is accomplished by moving the reset lever in the direction indicated on the name plate.

performing the same function as though the plunger had been moved manually by means of the emergency stop push button.

b. General Motors. In this installation an air cylinder operated from the maneuvering room operates the hand control lever on the engine. The air cylinder piston is connected to the hand control lever shaft through a slotted link and a lever. The compressed air power stroke moves the lever to the STOP

10C4. Remote control engine stop. a. Fairbanks-Morse. The engine can be stopped pneumatically from the maneuvering room. Operation of the remote control lever at that station permits compressed air to enter a cylinder at the emergency stop mechanism on the engine. The air moves the stop plunger, thereby position. A spring returns the air cylinder piston to the idling position when the air is shut off and the line vented. The slotted link allows the hand control lever to be moved through the full length of quadrant travel once the air has been discharged.

[Previous Chapter] [Sub Diesel Home Page] [Next chapter]

11
JOURNALS, BEARINGS, AND ALIGNMENT

A. GENERAL

11A1. General. The operator of any piece of machinery should thoroughly understand the various adjustments that are necessary for perfect operation. It is not enough for him to know merely which valve to open and close or the position of maneuvering levers in order to start, stop, and reverse his machine. He must possess knowledge of the functioning of each of its systems when he manipulates this gear. He should be alert to note the difference between efficient and poor performance by the sound, smell, and touch of the machinery. Instruments, such as gages, thermometers, and tachometers, however, should be the guides that the operator uses in detecting the approach of trouble so as to take corrective measures before anything serious occurs.

The modern diesel engine demands greater skill on the part of the designer and builder than any other kind of engine. Likewise in its operation it is far from being foolproof and requires intelligent attention. The adjustments are precise and to narrow limits. Overhaul and fitting of the pistons, rings, bearings, valves and fuel pumps are beyond the capacity of the sixty-fourths and thirty-seconds of an inch are not recognized in diesel engine work; the increment of measure for everything is thousandths of an inch.

There should be a regular routine for checking the different systems of the engine and performing upkeep functions. At this time all indications of wear, parts renewed, and adjustments made should be recorded in a systematic log book to be used as a history from which information may be obtained at future overhaul periods. This is always done during submarine refit periods and at any other time when it is found necessary.

11A2. Construction of bearings. The method used in construction of a bearing depends upon the type, the bearing metals to be used, and the type of use required. In the case of precision type bearings, it is necessary that the two halves form a true circle when finished. This requires rather ingenious practice, and shop procedures will vary.

Other than the shop procedure, there are only a few items concerning the construction of bearings that are worthy of mention. The first of these is the question of oil grooves. Bearing lubrication in the 2-stroke cycle

ordinary machinist and demand the efforts of a skilled mechanic. If it is properly adjusted, a diesel engine, once started, will run until it is stopped. The extent of its reliability over a long period of operation depends upon the intelligence and skill of its operator.

Care must be used when operating oil engines of any make, regardless of whether the engine is of the 2- or 4-stroke cycle, vertical or horizontal, air or mechanical injection. The working principles are the same, and the same care must be exerted to have everything properly adjusted before starting or operating the engine. Otherwise, there may be trouble. Every bearing should be adjusted as tightly as possible. This is a task for the real mechanic and should not be entrusted to unskilled personnel. The upkeep of the engine is an important duty, and one in which the real engineer shows his value. It should be kept in mind that measurements of engine is more difficult than in the 4-stroke cycle engine, since in the latter, the point of contact between bearing and journals of both main and crankpin journals rotates around the bearing and assists in the distribution of the oil. In the 2-stroke cycle engine, the point of contact swings back and forth across the lower bearing shell and hence, in this engine, it is usually necessary to provide oil grooves on the unloaded side to carry sufficient oil to the loaded half of the bearing. Where bronze or other flat bearings are used as wrist pins, ample grooving must be provided. Grooves may be cut axially, circumferentially, diagonally, or helically across the face of the bearing, but should never extend to the edge since this would allow the oil to spill from the bearing. All grooves should have rounded

edges, as the sharp edge of a groove has a tendency to act as a scraper and may impair the oil film.

In order that the flow of oil between the bearing halves may not be restricted, bearings are beveled for an arc of about 20 degrees at the joints where the bearing halves come together, except for a narrow strip at the ends, where the full thickness of the metal must be retained to surfaces may make physical contact and rupture the oil film.

From the above, it is obvious that an oil film must be maintained at all times in order to carry the load. This condition is called stable lubrication. When the oil film is destroyed and lubrication of the bearing depends entirely upon the oiliness of the lubricant, we have what is known as unstable lubrication. This latter condition exists when the bearing shaft is

prevent the loss of oil. The spaces formed by beveling are called oil cellars.

11A3. Bearing loads. The only bearings in a diesel engine that require careful consideration due to the heavy loads placed upon them are the main, crankpin, and wrist pin bearings. Other bearings are not so limited in size, and little attention need be given them in so far as their ability to carry the load is concerned. The following discussion pertains principally to the above three heavily loaded bearings that are usually limited in size by the space available.

The study of bearing loading brings two things to mind: 1) the temperature at which the bearing must operate, and 2) the maximum pressure per unit area that will be exerted upon the bearing. Too much pressure will squeeze out the oil film and ruin the bearing, and too much heat will reduce the viscosity of the oil until the film can no longer be maintained. Both of these are factors of loading, although the latter is a product of loading and speed of rotation.

In a diesel engine operating at variable loads, the successful bearing design is generally the result of experimentation directed toward the discovery of a satisfactory bearing for all loads. The loads that a bearing can withstand are based upon the assumption that the surfaces of the journal and bearing are smooth and parallel, that proper clearances are provided, and that sufficient lubrication is provided. Too much oil clearance at the running at too low a speed to build up an oil film or when the bearing is overloaded.

11A4. Bearing metals. Compared with the journal, the bearing metal should be sufficiently soft so that any solid matter passing through in the oil stream will wear the bearing instead of the journal.

Roller and ball bearings are frequently used in diesel engines for smaller shafts, such as camshafts, and in governors, because they greatly reduce the bearing friction and because their smaller clearances keep the shaft more rigid. In at least one opposed piston type of diesel engine of medium power, ball bearings are used as main bearings. For wrist pins, roller bearings of the needle type are used extensively in other types of large engines. With these bearings, lubrication is made simpler, and the amount of freedom of motion and friction is reduced.

Wood is used in the tail shaft bearings of naval vessels that are submerged in water and constantly lubricated and cooled. Lignum vitae is the wood commonly used for this purpose since it is of a greasy character and extremely hard and dense. Other types of materials used for this purpose include hard rubber strips and phenolic resinous materials.

Bronze bearings are used where the pressures are very high such as at the wrist pin. Here the load on the bearing is the total gas pressure less the inertia of the piston. In most modern diesel engines, bronze is used as the

ends of a bearing will cause excessive oil leakage and subsequent reduction in load-carrying ability. If the bearing were closed at the ends, the pressure would be uniform over its entire length, and much greater loads could be carried. If the shafting is of in alignment, or vibrates severely, as when running at a critical speed, the faces of the journal and bearing will not be parallel, and the metallic bearing metal for the wrist pin bearing.

There is no material known that is suitable for all types of bearings. There are four general types of alloys used today but each has its own particular uses determined by the maximum unit pressure and temperature at which the

bearing will operate, and by the hardness of the journal.

Bearing metals should be of such composition that the coefficient of friction is low. They should be sufficiently hard and strong to carry the load, but must not be brittle. If they are too soft, they will wipe or be pounded out, destroying the clearance and reducing the bearing area. In grooved bearings the grooves will become filled with wiped metal. When this trouble arises the oil film is squeezed out, the metal is burned, and failure results.

The four commonly used types of bearing linings are: high-lead babbitts, tin-base babbitts, cadmium alloys, and copper-lead mixtures.

The backs for bearings are made either of steel or bronze in the case of the babbitts, while only steel backs are used for cadmium alloy and copper-lead bearings. In some bearings, an intermediate layer of metal is shells are either forged or cast, and the linings are made of lead-base babbitt metal.

In the naval service the most frequently encountered bearing metal used in precision bearings is that known by the trade name of Satco. The composition of this metal is as follows:

	Percent
Calcium	.30- .70
Mercury	.40- .90
Tin	1.00-2.00
Aluminum	.15- .17
Magnesium	0.00- .05
Lead	Remainder

11A5. Bearing installation and adjustment. In order to insure its successful operation, the bearing must fit the journal perfectly; the bearing and journal surfaces must be smooth and parallel, and the bearing clearance must be correct. Too great a clearance will allow the oil to spill out at the ends of the bearing, while too small a clearance will cause the bearing to run hot. In general, the least

used between the backs and the bearing metals.

The hardness of the above bearing metals naturally varies with the percentage of alloying employed. In general, however, the copper-lead and cadmium alloys are the hardest, while the high-lead and tin-base babbitts are the softest. The temperature at which the bearing metals melt is a rough measure of their degree of hardness, the softer metals melting at the lower temperatures. The softness of the bearing metal is also a measure of the maximum allowable unit pressure. The harder the bearing metal, the greater is the load that a given size bearing will carry without failure.

Where two metallic surfaces are moving in contact with each other, such as a journal rotating within a bearing, wear will inevitably take place. Since it is easier and cheaper to renew the bearing, the journal should "be harder than the bearing. Therefore, when using relatively hard bearing metals, such as cadmium alloy and copper-lead, it is necessary to use a hard alloy steel journal or else to harden the surface of the journal.

The precision type of bearing is rapidly coming into universal use for crankpin and main bearings. There is an increasing use of very thin bearing linings on steel shells. The clearance that will allow the successful operation of the bearing is desirable.

In the modern high-speed engine the precision type of bearing is generally used. No scraping-in is done, and no shims are used between the faces of the two halves. The bearing is accurately machined to the correct diameter and the only fitting necessary is an occasional filing down of the faces of the two halves in order to obtain a close and even fit when the bearing caps are brought together. In connection with the fitting of precision type bearings, too much emphasis cannot be placed upon the importance of having the backs of the bearing shells fit evenly against the bearing support. Recent experience with bearing failures due to this improper fitting has shown its importance. The areas not in contact fill with oil or air, both of which are relatively poor conductors of heat, and the transfer of heat from the bearing is reduced, causing the bearing temperature to increase. In addition, if an even fit is not obtained, a flexing of the bearing shell may result, causing the bearing metal to crack and flake off.

To assure an even fit the backs of the bearing should be examined at

bearings should be fitted to their supports in the same manner that the bearings are fitted to the journal. Since the back usually is made of steel, it is necessary to file down the high spots rather than scrape them down as is possible with softer bearing metals.

11A6. Bearing failures. When an engine bearing fails in service it can generally be attributed to one or more of the following causes:

1. Poor operating conditions and improper maintenance such as:

a. Improper or insufficient lubrication.

b. Insufficient cooling water.

c. Grit or dirt in oil.

d. Water in oil.

e. Bearings out of alignment.

f. Installing the bearing with improper clearances or uneven bearing surface.

g. Excessive load on the bearing.

2. Faulty design of the bearing or of the engine itself.

a. Improper dimensions of length and diameter.

b. Improper bearing material.

c. Improper lubrication. The lubricant, free from all foreign matter, must be supplied in ample amounts.

d. Improperly cooled.

e. Improperly grooved.

once. Also the lubricating oil gage pressure to the system and the passage of cooling water through the oil cooler should be checked. Sometimes the overheating may be due to foreign matter in the lubricating oil. The oil should be rubbed between the fingers to detect the presence of grit or dirt. An inspection of the filters will also reveal any abnormal amount of foreign matter deposited there. Since used oil generally is slightly acid, the presence of salt water may be detected by inserting a strip of red litmus paper in a sample of the oil. If salt is present to any degree, the litmus paper will turn blue. If salt water is detected in the oil, the crankcase and sump tank should be drained and refilled with new oil after flushing the system thoroughly. If possible, the cause of the salt water in the system should be determined. At the first opportunity the system should be well cleaned to remove any particles of salt that may have been deposited there.

As a rule, hot bearings may be traced to one or more of the following causes:

1. Improper or insufficient lubrication.

2. Grit or dirt in the oil.

3. Bearings out of line.

4. Bearings set up too tightly.

5. Uneven surface of bearing or journal.

6. Bearing overloaded.

If the temperature of the bearing continues to rise after the oil

f. Improperly baffled. Proper baffles must be fitted to prevent loss of oil, or its passage to adjacent parts of machinery, such as generator armature, where damage would result to the commutator. Also in some cases baffles are used to prevent the mixing of water with the lubricating oil.

3. The use of inferior lubricants, or the use of a good lubricant which does not meet the requirements of the piece of machinery.

a. Corrosion of bearings.

4. Inferior workmanship and material in the manufacture of the bearings and engine parts.

A bearing that is not operating properly will overheat. When this occurs, and the reason is not immediately known, the oil supply to

supply has been increased, the condition known as a hot bearing arises. The danger of a hot bearing lies in the fact that the babbitt expands until it grips the journal, thus causing a constant increase in friction and heat. When the temperature reaches the melting point of the bearing metal, the metal will run or wipe.

The treatment of heated bearings involves two main items: the removal of the cause, and the restoration of the bearing to its normal condition. If the trouble is due to improper or insufficient lubrication and is discovered before the metal has wiped, an abundant supply of oil usually will be sufficient to control the situation and gradually bring the bearing back to its normal temperature. Should the trouble be caused by an accumulation of dirt on the bearing, the abundant supply of oil will generally flush out

the impurities sufficiently to permit operation.

If the trouble is caused by foreign matter in the oil, the oil will have to be renovated or renewed. If the bearings are out of alignment, if they are set up too tightly, or if they have been improperly fitted, the fault cannot be fully remedied until the improper adjustments have been rectified. This usually involves stopping the engine.

In all cases the temperature of the bearing can be lowered by slowing down and thus

condition can become serious enough to cause bearing failures, and the only remedy is to machine or grind down the journal until it is again cylindrical. This, of course, will reduce the diameter and necessitate using a bearing of a different bore in order to effect the proper bearing clearances.

Journals should be kept smooth, even, and free of rust at all times. To remove spots of rust or ridges, the journal should be dressed with a fine file and then lapped with an oilstone or with an oilstone powder. Carborundum may also be used. If Carborundum is used,

decreasing the amount of load on the bearing. If the trouble has reached an advanced stage, it may be found necessary to stop the engine. When stopped, the bearing cap can be eased up a slight amount, thus increasing the clearance between the bearing and journal. However, the greatest care must be exercised in easing up on the bearing cap, for if too great a clearance is given, trouble will be experienced from pounding.

When the trouble is inherent in the bearing -as for example, if the machinery is not properly lined up, or the bearings are of insufficient area, or not in proper condition-only temporary relief can be secured from using the various means suggested above. The most effective treatment of a hot bearing is probably the operation of the machinery at a low or moderate power until such time as the needed readjustments, changes, or repairs can be effected.

To summarize the treatment for a hot bearing, the measures to be taken may be selected according to the special circumstances, from the following:

1. Lubrication.

2. Slowing down, and consequent reduction of load, or stopping.

3. Cooling water to oil cooler.

4. Easing up bearing caps.

Even though relief is obtained by the above measures, it should be borne in mind that once a great care must be taken to remove all particles, as these, if allowed to remain, will cause cutting and grinding of bearings.

When bearings have been removed for long periods, such as during a major overhaul, it is customary to wrap the journals with canvas in order to protect them from accidental damage. When this is done, only new canvas should be used. There have been cases where journals were wrapped with old rags or burlap that contained some acid. The action of this acid corroded and pitted the journals and it was found necessary to renew the entire shaft.

Each time a bearing is removed for any reason the journal should be carefully inspected. Any evidence of pitting or general corrosion indicates the presence of acid or water, and the lubricating oil should be analyzed immediately. When a bearing clearance exceeds the allowable tolerance, or when the bearing fails due to scoring, wiping, spalling, or cracking, looseness of the bearing metal, or for any other reason, it must be renewed.

To renew a precision type bearing it is first necessary to have available a spare bearing. These are manufactured to size and are available from the manufacturer. They are bored to correct dimensions, so that only a slight amount of scraping in and filing of the edges of the shell faces is required to produce an accurate fit. There seems to be a tendency to renew Satco bearings before it is necessary. A slight amount of spalling is not necessarily an

precision bearing has wiped, it is necessary to renew the bearing as soon as possible.

The wear on journals rotating in bearings is seldom, if ever, evenly distributed over the entire surface. Consequently the journal wears until it becomes eccentric or egg-shaped. This indication that the bearing properties of the metal are destroyed.

B. COUPLINGS

11B1. GM elastic coupling. The crankshaft of the GM 16-278A engine is connected to the generator shaft by means of an elastic coupling. The elastic coupling connects the engine to the generator flexibly by means of radial spring packs. The power from the engine is transmitted from the inner ring, or spring holder of the coupling, through a number of spring packs to the outer spring holder, or driven member. A large driving disk connects the outer spring holder to

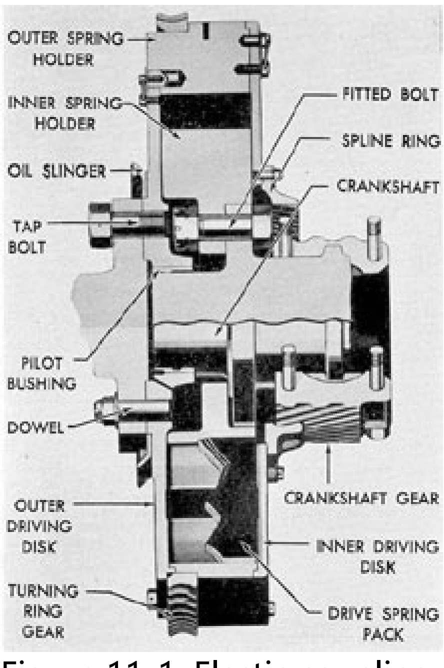


Figure 11-1. Elastic coupling

the inner driving disk. This helical internal gear fits on the outer part of the crankshaft gear and forms an elastic drive through the crankshaft gear which rides on the crankshaft. The splined ring gear is split and the two parts bolted together with a spacer block at each split joint. This makes it possible to engage separately the two parts of the splined ring with the crankshaft gear teeth, and to slide them into position with the idler gear in place.

The parts of the elastic coupling are lubricated with oil flowing from the bearing bore of the crankshaft gear through the pilot bearing.

11B2. F-M flexible coupling. The crankshaft coupling on an F-M installation consists of three parts: the engine coupling driving half, the laminated rings, and the generator coupling driven half. The coupling driving half is fastened to the lower crankshaft with fitted bolts, and the coupling driven half is likewise fastened to the generator shaft. Power from the engine is transmitted through

cross section, GM.

the flange on the driven shaft. The pilot on the end of the crankshaft fits into a bronze bushed bearing on the outer driving disk to center the driven shaft. The turning gear ring gear is pressed onto the rim of the outer spring holder.

The inner driving disk through which the camshaft gear is driven is fastened to the outer spring holder. A splined ring gear is bolted to

the laminated rings by means of a third set of fitted bolts held in place by ring bolt spacers.

Pilot rings between the ends of the generator shaft and the crankshaft form a safety guide in the event of failure of other parts. Tapped holes for jackscrews and drilled holes for body fitted bolts are provided in the lower flanges of the cylinder blocks. To permit fitting of the coupling bolts to the generator shaft, it is necessary to remove the lower and upper halves of the end cover back of the coupling driver half, the lower bearing cap, and the lower crankcase side cover at the vertical drive compartment.

Guards and two jackscrews of different lengths are furnished with the tools by the engine manufacturer for use in removing and installing the coupling bolts. The guards protect the bolt threads and are tapered to facilitate entry of the bolts when fitting.

When installing coupling bolts in either set, the shorter jackscrew should be used for starting the installation and the longer jackscrew for completing it.

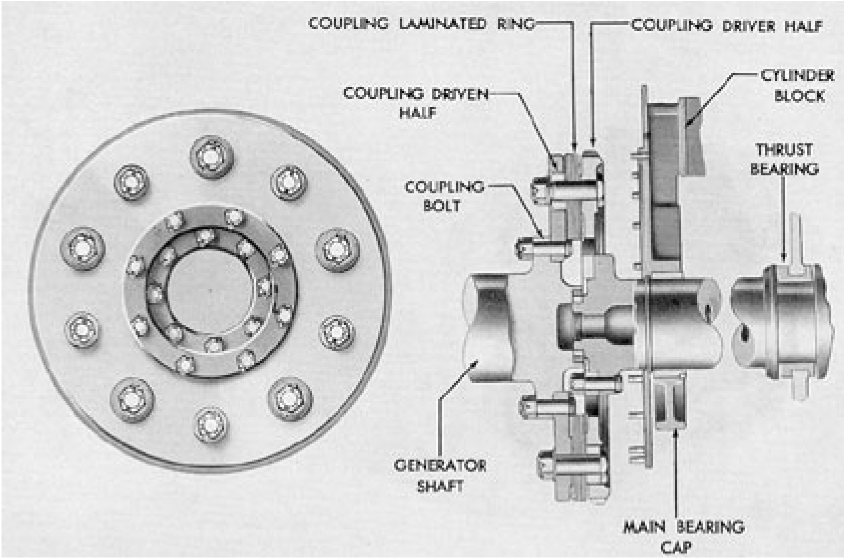


Figure 11-2. Crankshaft coupling, F-M.

C. ALIGNMENT

11C1. General. Good engine and generator performance can be obtained only if the original coupling installation is made with the components in correct alignment and with correct clearances. The problem of originally aligning a generator set and subsequent checking arise quite frequently during submarine wartime operations. The original alignment, of course, is extremely important as it greatly influences future operation and adjustment of the engine. During navy yard overhauls it is common practice to take motors and generators out of the ship for overhaul, and the young engineer officer or new leading chief motor machinist's mate is frequently called upon to check an alignment job being done by naval shipyard personnel. It as also become routine to check crankshaft alignment to some degree after an engine overhaul in which many of the engine parts have been renewed. This may be only a checking of the crank cheek deflections with the good idea as to the status of the alignment of the equipment. The most important and most difficult job of alignment is the complete installation, of a generator set. The salient points of these installations will be covered in the following paragraphs. When the principles involved in a complete alignment job are understood, smaller alignment problems become relatively simple.

NOTE. Alignment tests and corrective measures should never be undertaken when a vessel is in drydock because the alignment of the shafting is not the same when the vessel is waterborne as when it is in dock.

11C2. Strain gage readings. The strain gage is basically a micrometer for measuring the differences in distance between the two webs or cheeks of a crankshaft during a revolution of the shaft. As previously stated, one of the basic alignment procedures is the taking of strain gage readings. This is a relatively simple

use of a strain gage, but even this will give the operating personnel a

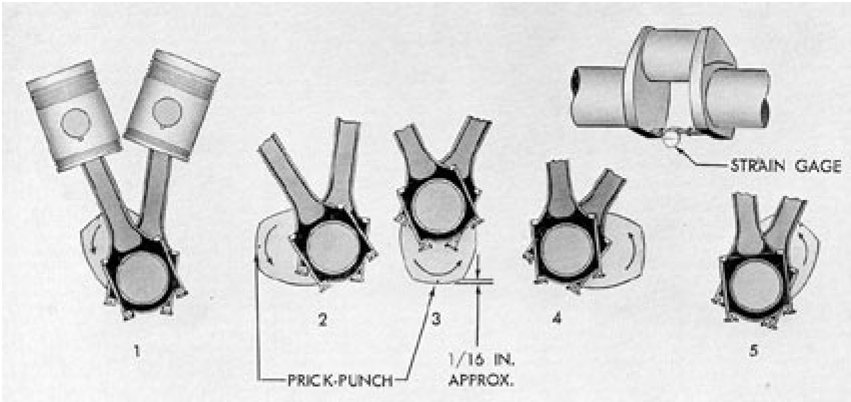


Figure 11-3. Position of crankshaft for strain gage readings.

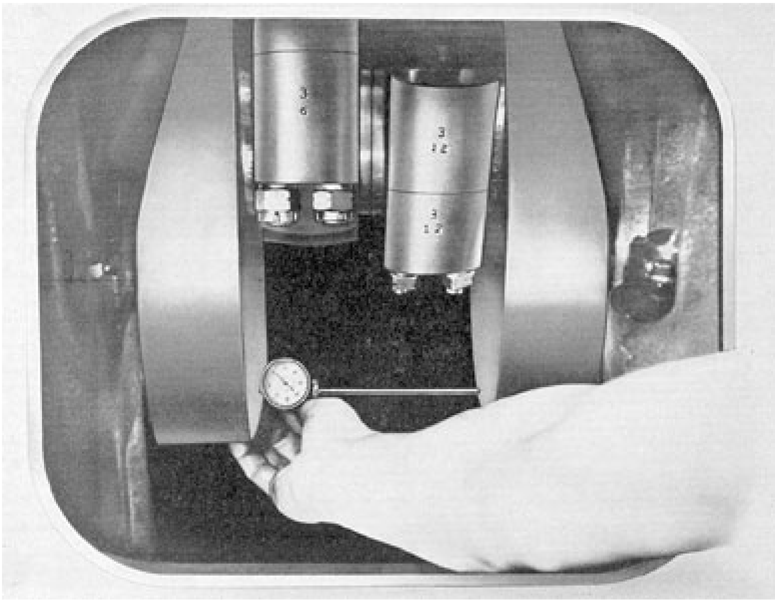


Figure 11-4. Measuring crank check deflection with a strain gage.

undertaking but it is important that the procedure be followed exactly for best results. A series of strain gage readings of a crankshaft gives a measurement of the crank cheek deflection for various angular positions of the shaft. The measurement is accomplished by placing the gage between the engine crankshaft cheeks. The gage should be installed with its two endpoints in the crankshaft

prick-punch marks. The crankshaft should be turned to its initial position so that the gage will be as close to the top position as possible without touching the connecting rod. The dial of the strain gage is then set on zero, and the crankshaft is slowly jacked over to subsequent positions as shown in Figure 11-3 and the readings taken. When taking the readings, the gage should not be allowed to rotate about its end-points.

After the readings have been taken for one revolution of the crankshaft, they should be compared, and the maximum crank deflection obtained. Large variations in the individual readings indicate some type of misalignment in the installation.

11C3. Alignment of engine crankshaft with one bearing generator. This type of installation is that normally found on F-M generator sets. There are many recognized methods of accomplishing alignment of engine and generator. The following procedure is one method and is discussed more from the standpoint of alignment principles than of a standardized alignment procedure.

Generators and crankshafts that are being coupled together must be in alignment. This condition is attained by moving the shaft bearing supports vertically and horizontally until the two halves of the coupling are true to each other or until the axes of the two shafts coincide at the point where they are coupled. The operation usually involves

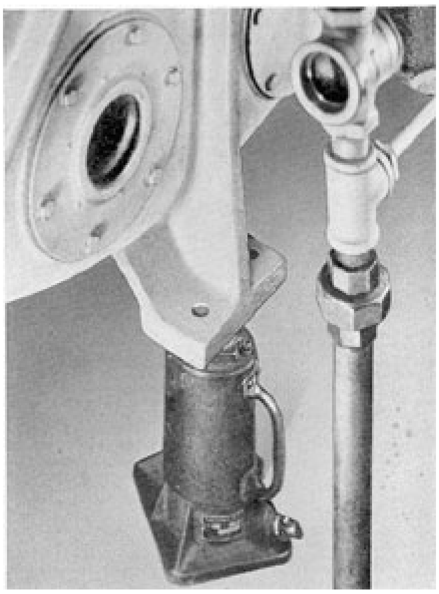


Figure 11-5. Using hydraulic jack to adjust height of generator body for proper vertical alignment.

In all modern submarines the engines are attached to a generator rather than directly to propeller shafts. When a generator is being installed, it should be originally placed as nearly as possible in final alignment. Subsequent procedure is as follows:

1. Attach the driven half of the crankshaft coupling to the driver half by installing the outer row of bolts around the coupling. Tighten the bolts evenly.

2. Secure the generator shaft to the flexible coupling by installing coupling bolts through the flange on the end of the generator shaft into the driven flange of the coupling.

3. Check the strain gage measurements to determine whether or not the coupling operation has affected the original reading. If a large change is noted at a particular position of the crankshaft, it indicates that the coupling has placed a strain on the crankshaft.

movement of the entire generator casing.

In the following alignment it is assumed that the engine is already located. Before starting alignment, the amount of crankshaft cheek deflection should be known and recorded in order to be able to make a comparative check during and after the alignment has been completed. The crankshaft cheek deflection readings should agree within approximately 0.002 inches.

4. Check the thickness of the flexible coupling with a micrometer. The measurements should be made at the top and bottom, inboard and outboard. Compare the measurements with

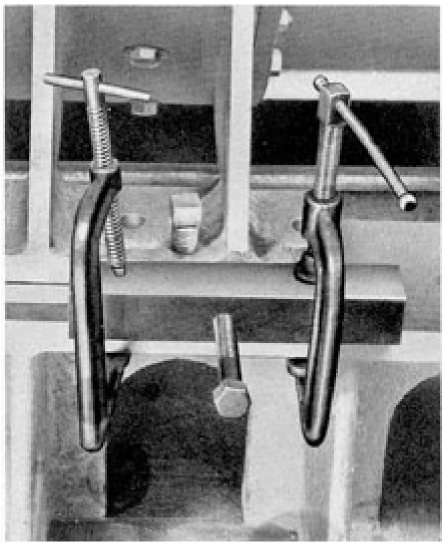


Figure 11-6. Using portable block and jack screw to adjust generator body for proper lateral alignment.

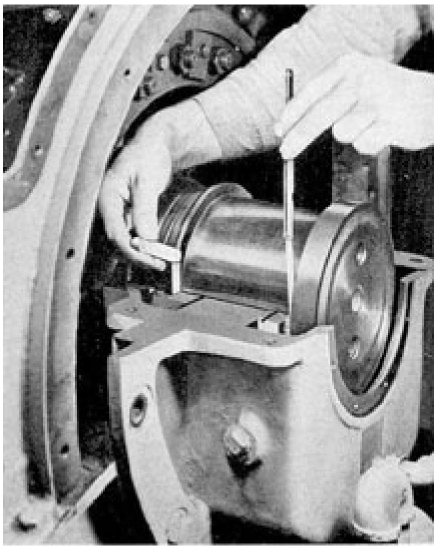


Figure 11-7. Measuring generator thrust bearing clearances.

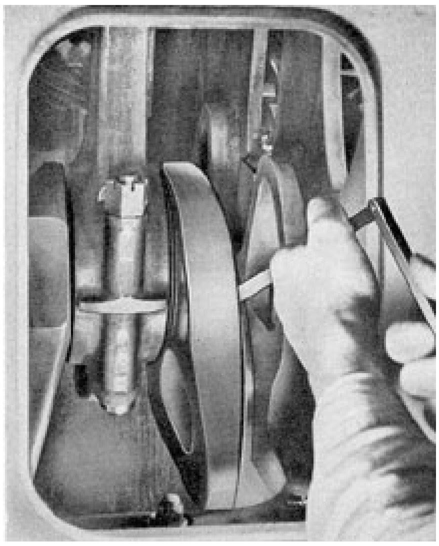


Figure 11-9. Measuring crankshaft

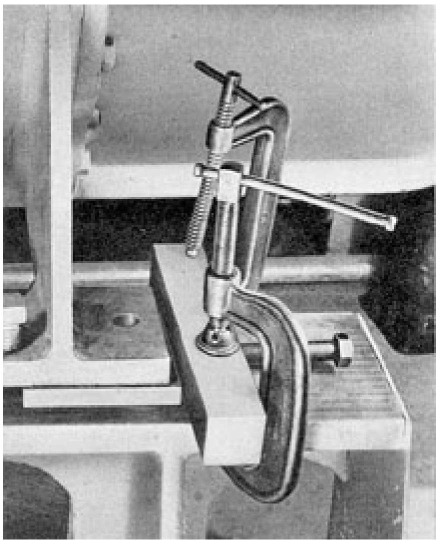


Figure 11-8. Using portable block and jack screw to adjust generator for proper thrust clearance.

thrust bearing clearance toward control end of F-M engine.

225

the established dimension stamped on the flange by the manufacturer. For example, the manufacturer's dimension is 5.225 inches. The outboard measurement as made with the micrometer is 5.250 inches. The inboard measurement is 5.200 inches. This indicates that the generator shaft is placing a strain on the inboard side of the coupling and is probably also affecting the strain gage reading on the crankshaft. Therefore, the generator casing and shaft must be moved outboard 0.025 inches to balance the readings on the coupling and to remove the strain from the crankshaft. Normally, this should bring the strain gage readings back to the original readings.

If a difference in the measurement of the coupling, against the stamped dimension, occurs at the top or bottom of the coupling, the generator casing and shaft will necessarily

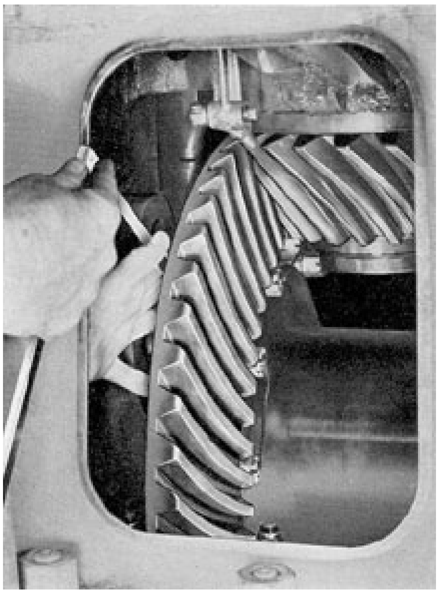


Figure 11-10. Measuring crankshaft thrust bearing clearance toward generator end of F-M engine.

and generator thrust bearing clearances.

6. Measure the crankshaft thrust bearing clearances by inserting a feeler gage between the crank cheek and the face of the bearing (Figure 11-9), and between the vertical drive gear and the generator end of the bearing face (Figure 11-10). The total clearance

have to be raised or lowered to effect a balanced condition. The height of the generator casing and shaft may be adjusted by means of hydraulic jacks placed under the casing as shown in Figure 11-5. When the proper height is attained, block up the casing, remove the jacks, and install shims between the feet and the permanent pedestals on the deck.

The generator shaft and casing may be moved inboard or outboard by installing a portable block and jack screw against the edge of the side foot mountings of the casing as shown in Figure 11-6. To check the distance of the movement, attach a dial indicator on the opposite pedestal with the indicator pointer touching the edge of the opposite foot of the casing.

5. Remove the generator thrust bearing cap and measure the generator shaft thrust bearing clearances (Figure 11-7). The clearances should measure 0.0075 inch (approximately) at each end and on each side of the bearing in an F-M installation. If it is found, for example, that there is no clearance at the thrust face away from the engine, the generator casing must be moved toward the engine 0.0075 inch with a portable block and jack screw (Figure 11-8). This operation, however, should not be accomplished until the crankshaft thrust bearing clearances have been measured, since it is possible that only one movement of the casing will be needed to correct both crankshaft should measure between 0.004 and 0.010 inch evenly distributed on both sides. If the clearance on one side is greater than on the other, it will be necessary to move the generator shaft in one direction or the other to balance the measurements.

Any movement of the shaft will affect the clearances at the generator thrust bearing. It may also affect the strain gage readings and the setting of the flexible coupling. Additional movement, therefore, of the shaft, or the casing may be necessary to bring about a balanced condition. A check should be made after every move and steps taken to correct any offset condition which may have been brought about by a previous move.

7. Movement of the generator casing, or

the shaft, will probably have some effect on the generator air gap (the space between the armature windings and the pole pieces). The air gap must be uniform around the diameter of the armature. The clearance should be kept within the limits specified by the manufacturer of the generator. Air gap measurements are taken with long thickness gages furnished for this purpose. The gages are inserted between the armature winding and each pole. When the air gap is found to be greater at the top than on the bottom, the generator casing will have to be lowered by loosening the jacking screws located on the side feet of the casing. If the gap is greater at the bottom, the casing must be raised with the jacking screws, and shims inserted between the side feet and the pedestals.

Assuming that the shaft alignment has been completed and is true, it will be necessary to secure the rear foot of the generator casing to the rear pedestal with a C-clamp. This will hold the alignment of the shaft while the casing is moved for adjustment of the air gap.

After attaining a balanced air gap, correct shims should be installed. A complete recheck of all clearances should be made to verify the alignment installation. This check must include another set of strain gage readings. Before this final check, the generator casing should be than on the other, the crankshaft must be moved in whichever direction will balance the clearances. This may be accomplished with a pinch bar placed between the crank cheeks and the engine framework. After clearances have been balanced, the crankshaft must be blocked with hardwood wedges placed between the crank cheeks and the framework, to prevent movement of the crankshaft during the coupling operation.

3. Determine the amount of fore-and-aft movement of the elastic coupling. This measurement is made by placing the pointer of a dial indicator against the face of the outer driving disk of the coupling. The indicator may be secured to the upper half of the coupling housing and the pointer should touch the driving disk near the center. The coupling is then forced as far forward or aft as possible, with a pinch bar, and the dial indicator is set on zero. Make a prick-punch mark on the face of the outer spring holder in line with the jacking gear pointer. Then force the coupling as far as possible in the opposite direction and make another mark. The dial indicator reading denotes the full fore-and-aft movement of the coupling, which normally is about 0.0125 inch. In order to divide the coupling thrust evenly between the engine and the generator, the coupling must now be moved to the center of its thrust or 0.0625 inch using the dial indicator as a guide. Make a third prick-punch mark between the two previously

rigidly secured in position with a C-clamp to prevent any possible movement of the casing.

11C4. Alignment of engine crankshaft with two-bearing generator. This type of installation is typified by the GM generator set. The GM engine is connected to the generator by means of an elastic coupling. A procedure to follow in aligning a generator to the coupling is as follows:

1. Take strain gage readings to determine the amount of crank cheek deflection. The maximum permissible deflection in a GM engine is 0.0035 inch. The measurement should be re-recorded for reference after completing the alignment.

2. Check the engine crankshaft thrust bearing clearances with feeler gages. Clearance should total approximately 0.030 inch, equally divided on both sides of the bearing. If the clearance is greater on one side of the bearing

made. This mark is the reference mark used to check the center of the coupling thrust after alignment has been completed.

4. Remove the outer driving disk from the coupling and bolt it to the flange on the generator shaft. Check the amount of deflection of the face of the disk with a dial indicator by turning the generator armature one complete revolution. The deflection should not exceed 0.001 inch. Next, place the indicator pointer against the rim of the disk, rotate the shaft one revolution, and check the amount of deflection. This measurement should also be within 0.001 inch. If the amount of deflection, on either the face or the rim of the disk, is greater than 0.001 inch, the condition may be corrected by loosening the bolts and recentering the disk or by cleaning the inner surfaces of both disks.

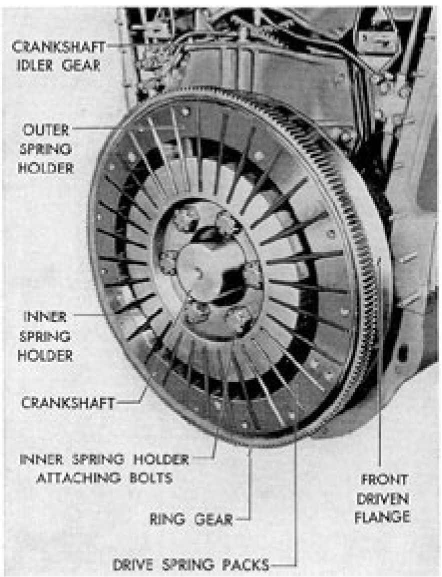


Figure 11-11. Elastic coupling,

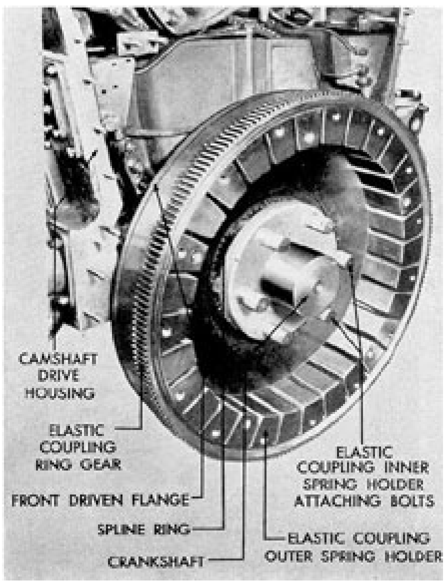


Figure 11-12. Elastic coupling,

outer driving disk removed, GM.

inner spring holder removed, GM.

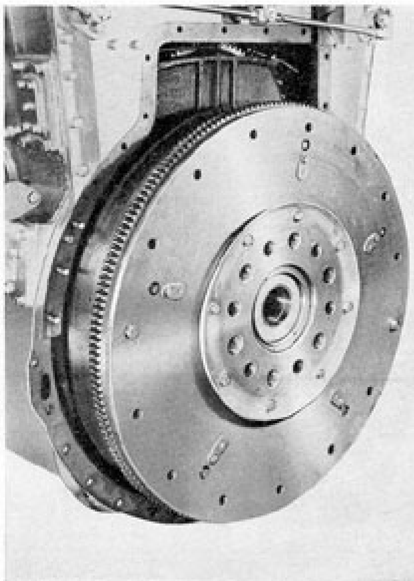


Figure 11-13. Mastic coupling, outer driving disk mounted, GM.

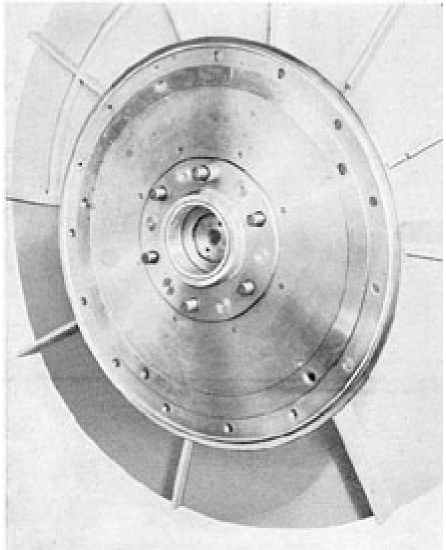


Figure 11-14. Elastic coupling, outer driving disk mounted on generator, GM

After obtaining deflection readings within 0.001 inch, the bolts, the driving disk, and the generator shaft flanges must be marked so that they may be replaced in their respective positions when the generator is coupled to the elastic coupling. Before removing the flange, the dowel holes must be reamed for the body-bound dowels. Dowels and dowel holes must also be marked so that they will be replaced in their respective positions.

5. Remove the driving disk from the generator shaft and reinstall it on the elastic coupling.

6. Install the upper half of the elastic coupling housing. Move the generator toward the engine. Approximate axial alignment may be attained with the jacking screws on the generator feet. Inboard and outboard alignment may be attained by use of with feeler gages. The thrust should be evenly divided between the two sides. If the thrust clearance is greater on one side than on the other, the generator housing must be moved until a balanced condition is attained.

8. Remove the hardwood blocks securing the crankshaft. Recheck the engine crankshaft thrust bearing clearances and the setting of the elastic coupling in relation to the center prickpunch mark. If either the bearing or the coupling has moved, the condition must be corrected by moving the crankshaft, the coupling, or the generator. If any move is made, it will be necessary to recheck the engine crankshaft thrust, the coupling, and the generator thrust bearing.

When all clearances are correct, a strain gage reading must be taken and checked against the recorded original reading. A change in the

portable blocks and jack screws working against the edges of the generator feet. When an alignment as nearly perfect as possible has been attained, the generator is moved farther toward the engine and the generator shaft carefully inserted into the bore of the driving disk.

Align the marked dowels with their corresponding dowel holes. If the generator is properly aligned, the dowels will slide into their dowel holes. No attempt should be made to force the dowels. If they cannot be inserted by hand, the generator must be moved until perfectly aligned.

After installing the dowels, secure the coupling to the generator shaft flange by installing the tap bolts.

7. Remove the generator thrust bearing cap, then remove the bearing. Carefully inspect and clean the bearing. Replace the lower half of the bearing and check the thrust clearances strain gage reading indicates misalignment, a condition which, at this point, can be corrected only by moving the generator.

After perfect alignment has been attained, measure the space between the generator feet and the pedestals and install suitable shims. Back off the generator feet jack screws so that the full weight of the generator will be on the shims. Another strain gage reading must then be taken to check whether or not the shims have affected the setting of the generator. If a change is noted, it can be corrected by cutting down or adding to the thickness of the shims. If no change is noted, drill the dowel holes and install the dowels; then drill the bolt holes and install the bolts.

A final check is made by rotating the engine with the jacking gear several revolutions in the direction of rotation and then rechecking all clearances. A slight variation in clearances, if found at this time, is permissible.

12
AUXILIARY ENGINES

A. GENERAL MOTORS 8-268 AND 8-268A ENGINES

12A1. General. The General Motors 8-268 or 8-268A engine is used on board modern submarines as an auxiliary engine. It is located in the lower flats of the after engine rooms, and may be used for directly charging the batteries or carrying the auxiliary load, and indirectly for ship propulsion. The GM 8-268 is an 8-cylinder, in-line, 2-cycle, air started engine rated at 300 kw generator output at 1200 rpm. In general, the individual parts of the engine are similar to, but smaller than the corresponding parts in the GM 16-278A. For example, the camshafts, exhaust valve and rocker lever assemblies, injectors, pistons, cylinders, liners and connecting rods are almost miniature replicas of the 16-278A parts. The main differences between the engines appear in the construction and design of the various systems such as the scavenging air, exhaust, lubricating oil, and fuel oil systems, as well as in the fact that the 8-268 is an in-line engine.

12A2. Engine stationary and moving parts. a. Cylinder block. The cylinder block is the main structural part of the engine. It is composed of forgings and steel plates welded together, access to the crankcase. Eight are located on one side and seven on the other. The remaining handhole is covered by the air maze which may be moved. Seven of the covers are of the safety type, each having four spring-loaded plates, which in an emergency, relieve any undue pressure in the crankcase.

The main bearings are lubricated from the lubricating oil manifold located in the crankcase.

b. Crankshaft. The crankshaft is a heat-treated steel forging finished all over, having eight connecting rod throws or crankpins 45 degrees apart. The crankshaft is held in the cylinder block by nine main bearing caps. The bearing at the drive end of the engine acts as a combination main and thrust bearing. Lubricating oil is supplied under pressure from a main manifold located in the crankcase, and is forced through tubes to the crankcase crossframes, where it flows through oil passages to the main bearings. From the main bearings the oil flows through drilled holes, in the crankshaft to the adjoining crankpin and lubricates the connecting rod bearing. The combination main and thrust bearing journal No. 9 is not connected by drilled holes to a crankpin. There is a 1/4-in. diameter radial oil hole in the

combining strength with light weight.

The upper and lower decks of the cylinder block are bored to receive the cylinder liners. The space between the decks is the scavenging air chamber. The bore in the lower deck is constructed with a groove which serves as a cooling water inlet for the liner. The cylinder liners are located in the cylinder block by means of dowel pins in the upper deck.

The camshaft bearing lower support is an integral part of the cylinder block located at the extreme top of the block. The bearing cape and bearing supports are match-marked and must be kept together.

The forged transverse members in the bottom of the cylinder block form the main crankshaft upper bearing seats. Again the bearing caps and bearing supports are match-marked and must be kept together.

Fifteen removable handhole covers permit

surface of this journal into which a capscrew, with the head ground off enough to clear the bearing seat, may be inserted for rolling out the upper shell.

c. Elastic coupling. The power from the engine crankshaft is transmitted through spring packs from the inner spring holder of the elastic coupling, or flywheel, to the outer spring holder, and from there through the driving disk to the generator armature shaft flange. A pilot on the end of the crankshaft fits into a ball bearing in the armature shaft. The turning gear pinion engages a ring gear shrunk on the rim of the outer spring holder.

The inner cover of the elastic coupling, through which the camshaft gear train is driven, is fastened to the outer spring holder. A helical

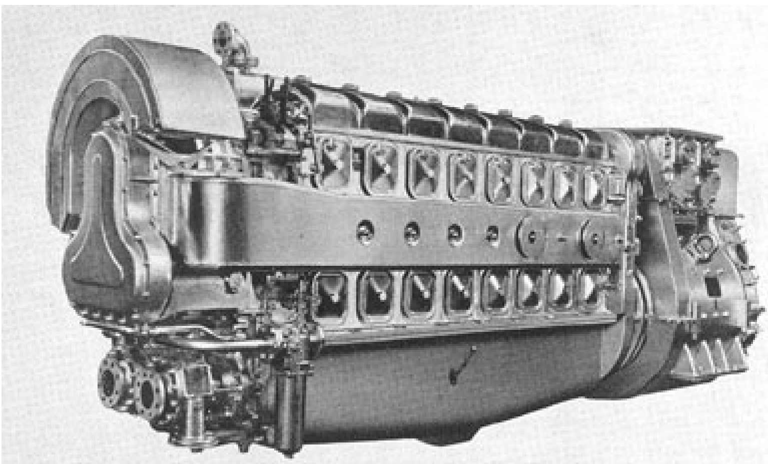


Figure 12-1. Blower end control side of GM 8-268 auxiliary engine.

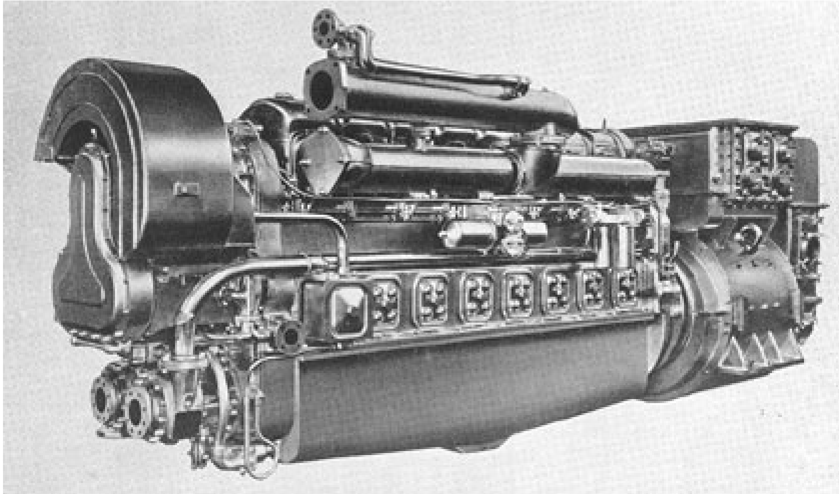


Figure 12-2. Blower end exhaust header side of GM 8-268 auxiliary engine.

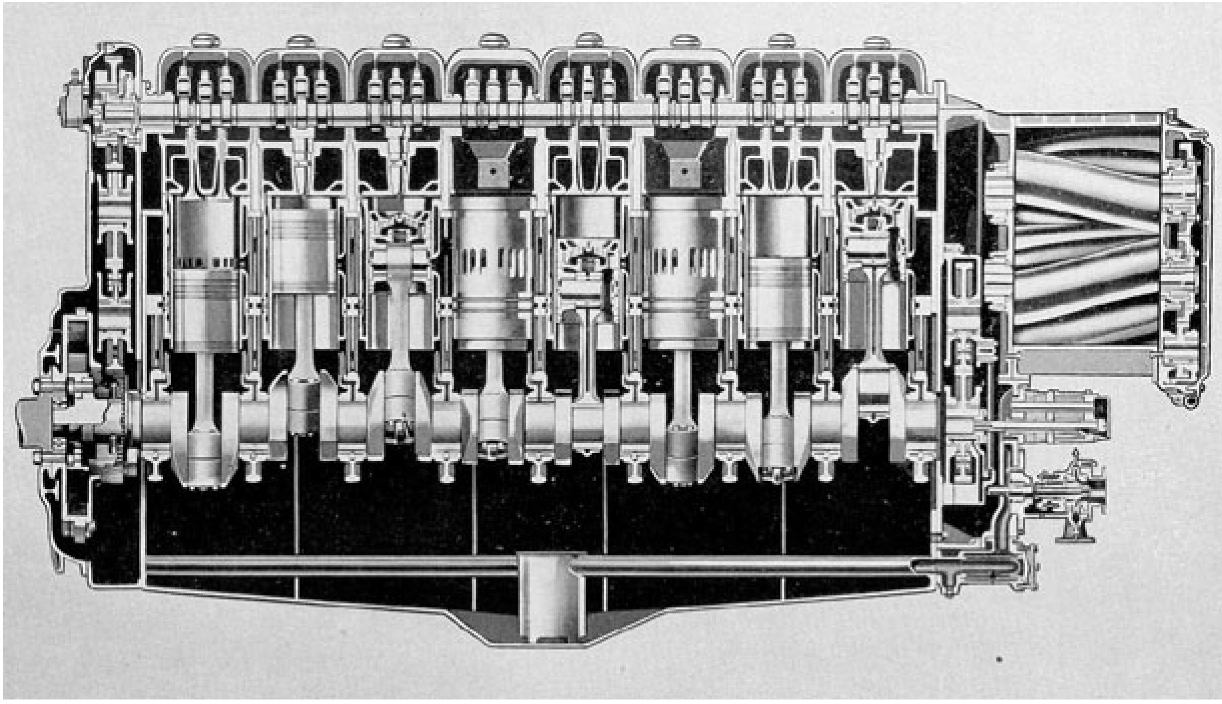


Figure 12-3. Longitudinal cross section of GM 8-268 auxiliary engine.

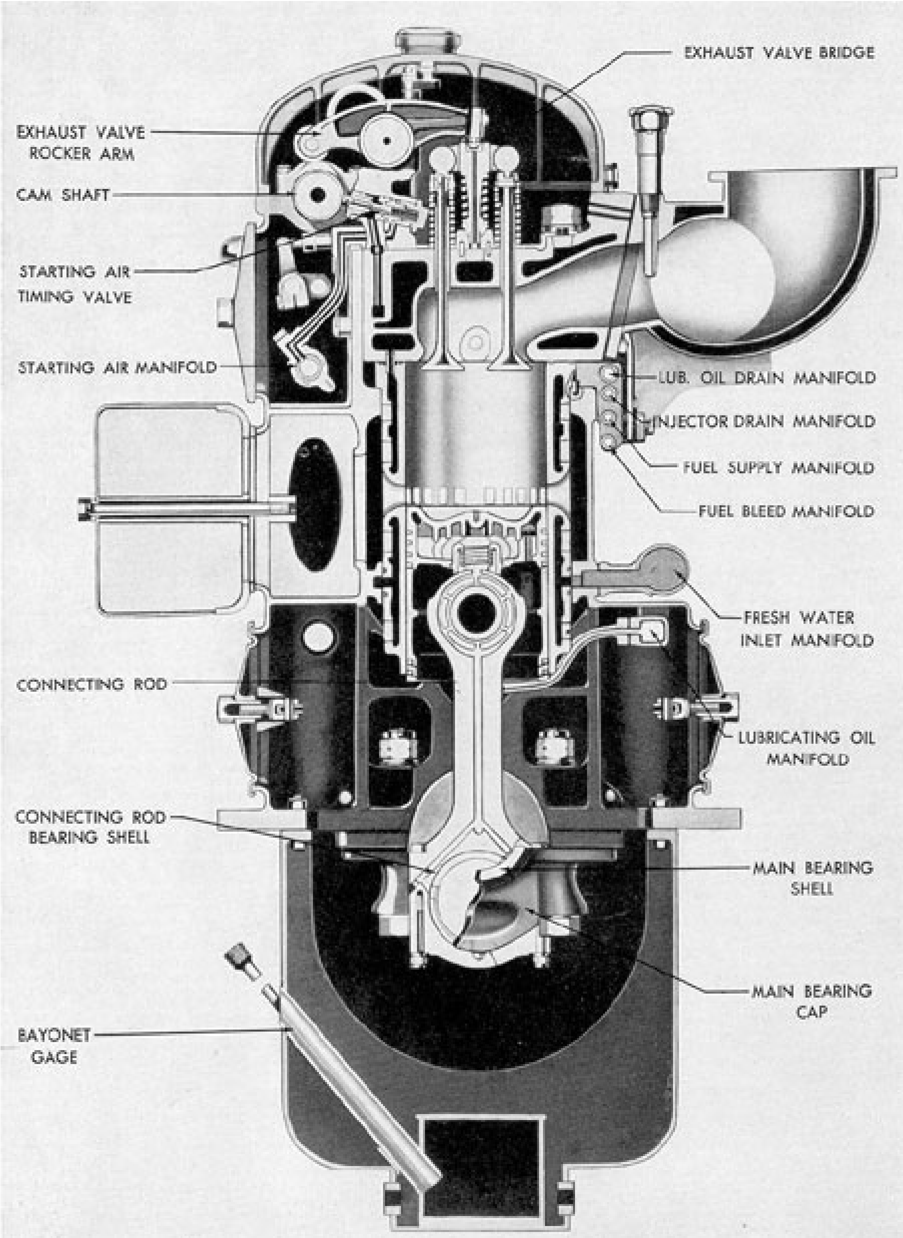


Figure 12-4. Transverse cross section of GM 8-268 auxiliary engine.

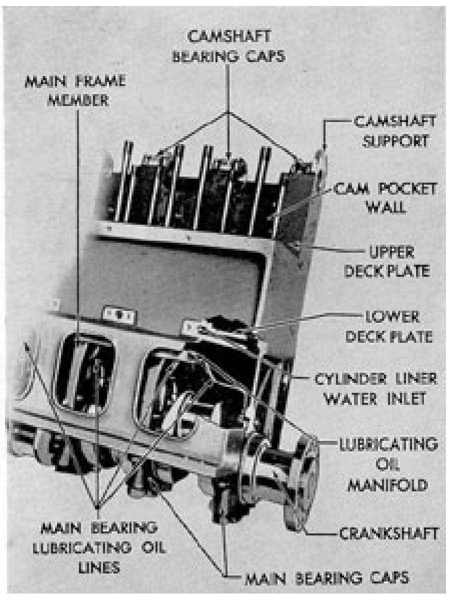


Figure 12-5. Cutaway of frame,

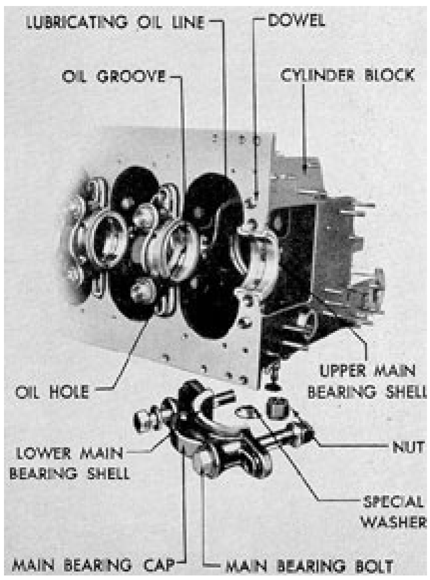


Figure 12-6. Lubrication of main bearings, GM 8-268

GM 8-268.

internal gear, cut in the inner bore of the elastic coupling cover, meshes with the crankshaft gear, forming a splined drive connection to the crankshaft gear which has a loose mounting on the crankshaft.

The bearing bore of the crankshaft gear. hub receives oil that flows from the adjacent main bearing through passages in the crankshaft. The parts of the elastic coupling are lubricated with the oil that flows from the bearing bore of the crankshaft gear hub.

d. Main bearings. Each main bearing consists of an upper and a lower double-flanged, bronze-backed, precision bearing shell. The centrifugally cast lining is a high lead bearing metal called Satco which contains a special hardener.

The lower shell is mounted in the bearing cap and the upper shell in its seat in the cylinder block crossframe. The joint faces of the upper and lower bearing shells project a very small amount above the seat and cap. That is to insure that the backs of the shells will be forced into full contact when the cap is fully tightened. A drilled hole in the lower shell fits on a dowel pin in the cap. The dowel pin locates the lower shell in the bearing cap and prevents both the upper and lower shells from rotating.

Each bearing shell is marked on the edge of one flange. For example, 2-L-B.E. indicates that the shell so marked is for the No. 2 main bearing, the lower bearing shell, and the flange so marked must be toward the blower end of the engine. The main bearing nearest the blower end of the engine is the No. 1 main bearing. Upper and lower bearing shells are not interchangeable.

Crankshaft thrust loads are taken by the rear main bearing. The thrust bearing shells are the same as the other main bearing shells except that the bearing metal is extended to cover the flanges. Each main bearing cap is marked with its bearing number and is marked Blower End on the side that should face the blower end of the engine.

Lubricating oil enters the oil groove in the upper shell through a hole in the top and then

flows to the lower shell. The bearing surface of the lower shell has an oil groove starting from the joint face at each side and extending partially around the inner surface of the shell.

of upper and lower bearing shells. The bearing shells are lined with Satco metal and are of the precision type. Each connecting rod bearing shell is marked on the edge of one flange. For instance, 1-L-B.E. indicates the shell is marked for the No. 1 connecting

e. Pistons. The pistons are made of an alloy cast iron. The bored holes in the piston pin hubs are fitted with bronze bushings. The outer ends of the bore for the full-floating alloy steel piston pin are sealed with cast iron caps.

A cooling-oil chamber is formed by an integral baffle, and the piston crown lubricating oil under pressure flows from the top of the connecting rod, through a sealing member, into the cooling chamber. The oil seal is a spring-loaded shoe which rides on the cylindrical top of the connecting rod. The heated oil overflows through two drain passages.

Each piston is fitted with six cast iron rings, four compression rings above the piston pin and two oil control rings below. These rings are of the conventional one-piece, cut-joint type.

f. Connecting rods. The connecting rod is an alloy steel forging. The connecting rod bearing in the lower end of the connecting rod consists rod, and lower bearing shell, and the bearing flange so marked must be toward the blower end of the engine. No shims are used between the connecting rod and the bearing cap. The upper and lower bearing shells are not interchangeable.

The lower shell is mounted in the bearing cap and the upper shell in its seat in the connecting rod. The joint faces of the upper and lower bearing shells project a very small amount above the seat and cap. This is to insure that the backs of the shells will be forced into full contact when the cap is fully tightened. A drilled hole in the lower shell fits on a dowel pin in the cap. The dowel pin locates the lower shell in the bearing cap and prevents both the upper and lower shells from rotating.

The piston pin is of the full floating type. The piston pin bronze bushing is a shrink fit in

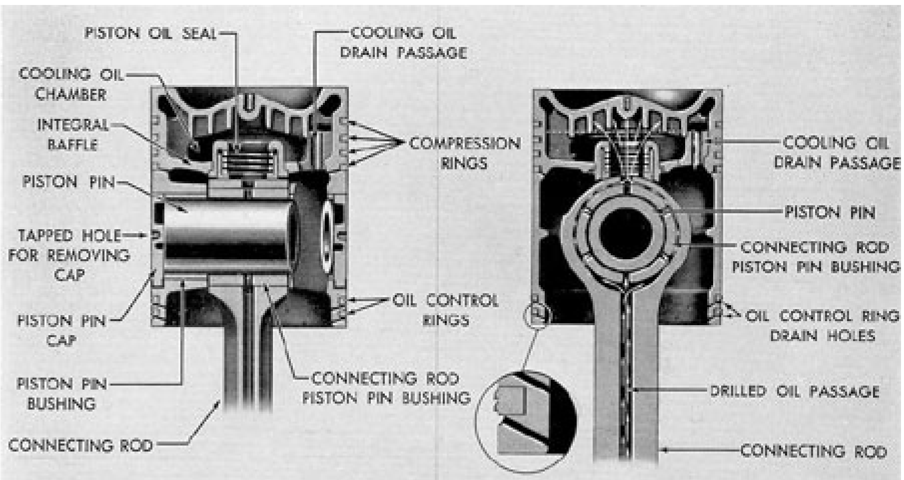


Figure 12-7. Cross section of piston, GM 8-268.

the upper hub of the connecting rod. The ends of the pin oscillate in the bronze piston pin bushing hubs of the piston.

g. Cylinder liner. The cylinder liner is a cylindrical alloy iron casting with cored annular spaces between the inner and outer surfaces between the inner and outer surfaces through which cooling water is circulated. The liner is accurately bored to a smooth finish.

The cylinder liner is held in the engine block by the lower deckplate and a recess in the upper deckplate. The cylinder head forces the liner against the cylinder block. The lower deckplate has a groove that serves as the water inlet into the passages in the cylinder liner. It is made watertight by two synthetic rubber ring gaskets, called seal rings. The cooling water flows up through the cylinder liner and into the cylinder head through ferrules made watertight by synthetic rubber gaskets. The air intake ports, through which scavenging air from the blower enters to supply the cylinder with fresh clean air, are located around the circumference of the liner. When the piston reaches the bottom of its stroke, these ports are completely open and the air space above the piston is charged with fresh air.

The joint between the cylinder liner and the cylinder head is made gastight by an inner bronze gasket while an outer copper gasket which has notches in it serves to seat the head squarely against the cylinder

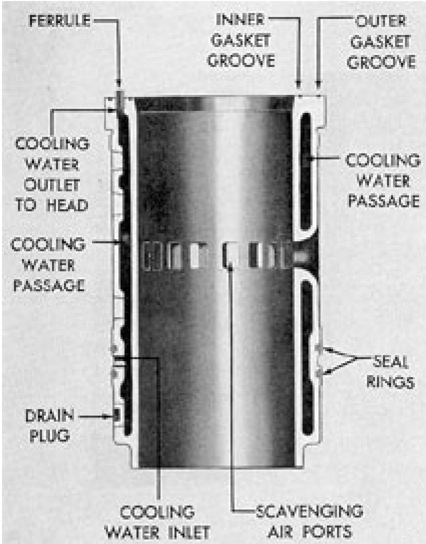


Figure 12-8. GM 8-268 cylinder liner cross section showing cooling water passages.

Cooling water flows from the cylinder liner into the head and then flows into the water jacket of the exhaust manifold.

Each cylinder head is fitted with four exhaust valves, the unit injector, rocker lever assemblies, air starter distributor valve, an over speed injector lock, the air starter check valve, and the cylinder test and safety valves.

i. Rocker lever assembly. Each cylinder head is equipped with three rocker levers, two of which operate the two pairs of exhaust valves, and the third operates the injector. The rocker levers are made of alloy steel forgings. Bushings are pressed into the lever hubs and are reamed for a bearing fit on the rocker lever shaft.

The three rocker levers rock on a fixed shaft which is clamped in a bearing support. They are fitted with cam rollers, which operate in contact with the exhaust and injector cams. Each of the three cam rollers turns on a bushing and the bushing turns on a sleeve that has a loose mounting on the roller

liner. The drain plug in the lower part of the jacket of the cylinder liner should be removed for draining water when freezing temperatures are expected and an anti-freeze solution is not in use.

h. Cylinder heads. The engine cylinders are fitted with individual cylinder heads which are made of alloy cast iron. Studs in the cylinder block hold each head against the cylinder liner flange. The joint between the head and the liner is made gastight with an inner bronze and an outer copper gasket. The outer gasket serves to seat the head squarely on the liner. The shallow milled grooves show leakage of exhaust gas or water.

The head is also fastened to the vertical wall of the cam pocket with tap-bolts. The joint is made oiltight with a synthetic rubber gasket.

pin. Each of the exhaust valve rocker levers operates two valves

through a bridge. Each of the valve rocker levers is fitted at the valve end with a nutlocked adjusting screw, which has a hardened ball end that fits into the ball socket in the valve bridge. The injector rocker lever is fitted at the injector end with a nut-locked adjusting screw, which has a hardened ball at the lower end. This ball is fitted with a hardened steel flexibly mounted shoe. The shoe bears on the injector plunger follower and transmits the rocker lever motion to the injector plunger.

the sequence of events essential to the operation of the engine will be in the proper order. The forged steel crankshaft gear, which is driven by, the crankshaft through the elastic coupling, is keyed on a split collar and drives the camshaft gear through the crankshaft and camshaft idler gears. A spacer ring is doweled to the crankshaft gear.

Steel-backed babbitt-lined bearing shells support the inner and outer hubs of the forged steel helical idler gears. The inner and outer supports are bolted and doweled together before being mounted in the camshaft drive housing. The

The rocker lever shaft is made of alloy steel and is ground to size. The shaft is clamped in the bearing support by two bearing caps and is held in its correct location by a dowel pin in one of the bearings. A rocker shaft thrust plate is bolted to each end of the shaft, and a plant fiber gasket is placed in the joint between the thrust plate and the rocker lever shaft. The bearing support is fastened to the cylinder head with two studs and positioned by two dowels, and is also held against the head by two of the cylinder head hold-down studs.

The rocker lever assembly is lubricated with oil received from one of the camshaft bearings. The oil flows from the top of the camshaft bearing through a tube to the plate connection that is fastened to one end of the rocker lever shaft. From this connection, the oil flows through drilled passages in the rocker lever shaft to the three bearings in the rocker lever hubs.

A drilled passage in each of the rocker lever forgings conducts the lubricating oil from a hole in the hub bushing to the camshaft end of the lever. The rocker lever motion permits oil to flow intermittently under pressure from the hole in the shaft, through one hole in the bushing and rocker lever to the cam roller. The bearing in each of the cam rollers receives oil through drilled holes in the roller pin and in the bearing bushings.

j. Camshaft drive. In 2-cycle engine operation the camshaft rotates at the same speed as the fuel oil pump and governor are driven from a gear that meshes with the lower idler gear. A pair of bevel gears drives the vertical governor shaft which is mounted in ball bearings.

The lower idler gear also drives the quill shaft gear, which is splined for the quill shaft that drives the blower and accessory gear trains. A splined coupling, which rotates in the babbitt-lined center bearing, joins the two sections of the quill shaft.

The overspeed trip weight assembly and the camshaft gear are bolted and doweled to a hub that also serves as a bearing journal for this assembly. The hub is splined to fit on the end of the camshaft.

Lubricating oil for the camshaft drive gear train and bearings is piped from the end of the lubricating oil manifold in the cylinder block. Oil is supplied under pressure to the hollow camshaft through the camshaft gear bearing. Open jets spray oil on the gear teeth.

Complete dynamic balance of the engine is obtained by balance weights mounted in a certain relation to each other on the gears in the front and rear gear trains.

k. Accessory drive. The accessory drive, located between the end of the crankcase and the blower, consists of a train of helical gears driven from the camshaft drive gear train through the quill shaft. The gears in the accessory drive are match-marked with a definite relationship to the match-marks on the gears in the camshaft drive gear train, to maintain the

crankshaft. The camshaft drive gears are located at the power takeoff end of the engine. They transmit the rotation of the crankshaft to the camshaft. It is necessary to maintain a fixed relationship between the rotation of the crankshaft and the rotation of the camshaft so that

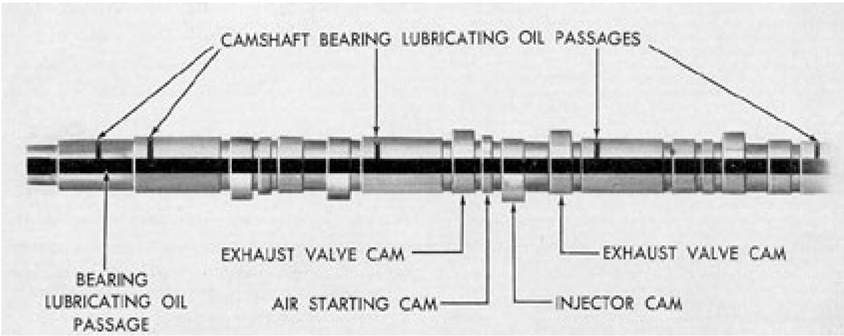


Figure 12-9. Cross section of camshaft, GM 8-268.

relationship between the balance weights in both trains.

The accessory drive gear drives the upper idler gear. This upper idler gear drives the lower idler gear. A plate with a splined hub for driving the lubricating oil pump is bolted to the hub of the lower idler gear. The fresh water and sea water pump drive gears are driven from the lower idler gear. The hubs of the water pump drive gears have a spline cut in the bore for the fresh water and sea water pump shafts. The hubs which project from each side of the lower idler and water pump gears run in steel-backed babbitt-lined bearings mounted in the inner and outer bearing supports. These bearing supports are bolted together and the assembly is fastened in place on the inside of the accessory drive housing.

steel backed and babbitt lined, are held on their seats in the cam pocket with bearing caps.

There are four cams for each cylinder. The two outer cams operate the exhaust valves, and the center cam operates the injector. The fourth cam, which is narrower than the other three, operates the air timing valve.

The camshaft drive end of the camshaft is splined for a driving connection in the hub of the camshaft gear which is driven from the crankshaft gear through a train of idler gears.

Lubricating oil under pressure is supplied to the camshaft bore through the splined drive connection. The oil is then delivered to the camshaft bearings through radial holes in the camshaft. Oil for lubricating the rocker lever mechanisms flows

Lubricating oil is piped to the accessory drive from the main lubricating oil manifold in the cylinder block. Oil lines and connecting pass ages in the bearing supports supply oil to the bearings in the drive.

The accessory drive cover should be removed periodically and the gear train inspected for excessive wear of any parts. Lubricating oil lines and passages should be checked periodically to insure that they are not broken or clogged. All nuts and capscrews should be tight.

1. Camshaft. The camshaft is of the one-piece type with integral case-hardened cams and bearings. The bearing bushings, which are through tubes from the camshaft bearing caps.

m. Engine control. The governor, which is located at the generator end of the engine, controls the engine speed for any setting.

The movement of the governor power mechanism is transmitted through lever and link connections to the injector control shaft in the cam pocket. Each fuel injector rack is connected to a control shaft lever through a slipjoint link. A micrometer adjusting screw on this link increases or decreases the amount of fuel injected into the combustion chamber.

A slip joint is connected to each injector rack so that in case the control rack in one injector binds, the compression of the spring in

the slip-joint link allows normal operation of the other injectors. Each spring is preloaded to limit the force that can be applied by the governor to move the injector control racks. When the link is either shortened or lengthened by a load greater than its assembly load, the spring is compressed.

The start and stop lever is used for manual control when starting or stopping the engine, and its movements are transmitted through a connection that provides for unrestricted governor control when the start and stop lever is latched in the RUN position. The governor connections to the injector control shaft include an extensible spring-loaded link filter on the cylinder head to a jumper tube that supplies the injector. The injector inlet contains another filter to further prevent solid matter from reaching the spray valve.

The surplus fuel is bypassed in the injector and flows through another filter in the injector outlet passage so that any reverse flow of fuel cannot carry dirt into the injector. The surplus fuel passes from the injector through a tube to a fuel bleed manifold, which is the bottom pipe in the multiple oil pipe assembly. The fuel from this bleed manifold flows to the metering block, through the metering valve which sets up enough resistance to maintain the required pressure in the fuel

which permits the injector control shaft to be turned manually without moving the governor power piston.

When the governor or any part of the injector control system is renewed, the governor power piston should be linked in the correct relation to the injector rack.

n. Overspeed trip. The overspeed trip mechanism stops the injection of fuel oil to the combustion chambers when the engine speed exceeds 112 percent of rated speed.

The overspeed trip weight assembly, mounted on the camshaft gear, is fitted with a spring-loaded flyweight. The spring tension is adjusted so that, at a predetermined engine overspeed, the centrifugal force moves the flyweight radially until it strikes a roller latch, releasing the spring-actuated injector lock shaft in the cam pocket at each engine cylinder. The injector lock carries a lever on the shaft that moves a pawl engaging a notch on the injector rocker lever. The injection of fuel stops when the locked rocker lever holds the injector plunger at the lower end of its pumping stroke.

The overspeed trip is manually reset with a hand lever on the shaft which projects from the camshaft drive housing.

12A3. Fuel oil system. a. Description. The fuel oil pump draws oil from the clean fuel oil tank and forces it through the fuel block and the fuel oil strainer and filter. From the filter, the oil flows to the fuel supply supply manifold, and then flows back to the clean fuel oil tank.

Fuel oil leakage from the injector plunger and bushing is drained through an injector body ferrule, through a cylinder head passage into a manifold connection clamped between the cylinder block and cylinder head. The injector drainage is conducted through this connection to the second manifold from the top in the multiple oil pipe assembly and then it flows through the drain to the fuel oil tank or bilge.

b. The unit injector. On this engine, the fuel pump and spray valve are combined into a single and compact unit called a unit injector, which meters the fuel and also atomizes and sprays it into the cylinder. This injector is similar to that used in the GM 16-278A and its operating principle is identical. The unit injector is held in position in a water-cooled jacket in the center of the cylinder head: At the lower end, the injector forms a gastight seal with the tapered seat in the cylinder head. All the injectors in this engine are alike and interchangeable. Fuel is supplied through jumper tubes with spherical type gasketless connections.

The pumping function of the injector is accomplished by the reciprocating motion of the constant stroke injector plunger which is actuated by the injector cam on the engine camshaft, through the injector rocker lever.

The position of the plunger, and thereby the timing, is adjusted by means of the ball stud and lock

manifold, which is the third pipe from the top in the multiple oil pipe assembly, and then through a small jet

nut at the injector end of the rocker lever.

The quantity of fuel injected into each cylinder, and therefore the power developed in

that cylinder, is varied by rotating the plunger by means of the injector control rack. A rack adjustment (called the microadjustment) located on the control linkage permits balancing the load of each cylinder while the engine is running,

c. Fuel block. The fuel block is located under the exhaust manifold at the camshaft drive end of the engine and in front of the fuel oil pump. The fuel block contains a metering valve, a priming valve, and an adjustable pressure relief valve.

d. Jet filters. The cylinder head jet filters are located on each head, just above the exhaust manifold connection. The element in each cylinder head is of the edgewise-wound metal ribbon type. This filter is correctly assembled when the helical spring and cap are placed over the long end of the filtering element to hold the element flange against the shoulder at the inner end of the filter wall.

e. Fuel pump. The fuel oil pump is located under the exhaust manifold at the camshaft drive end of the engine and is of the positive displacement, spur gear, rotor type. Fuel enters the pump through the top port in the end of the pump and is discharged

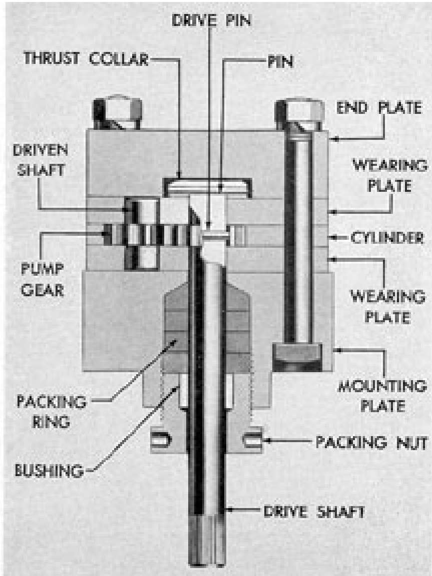


Figure 12-10. Cross section of Northern fuel oil pump used on GM 8-268 engine.

or both filtering units. In normal operation both filtering units are in operation.

The arrows under the valve handles show the positions of the valve handles for using either one or both of the units. The flanges are also marked IN and OUT indicating the direction of flow of fuel oil through the filter. When the valve handles are between the two positions indicated on the valve handle base, or with the valve handles directly above the inlet and outlet flanges, fuel oil is passing through both units. If the valve handle on the IN end of the filter is in one of the positions indicated by the arrow on the casting, the valve handle on the OUT end of the filter must be in the corresponding position. The

from the lower port on the side of the pump. Each pump gear is keyed to its shaft by a pin.

f. Fuel oil strainer. The fuel oil strainer contains two straining units, each with an inner and outer winding. The space between the windings on the inner and outer elements is 0.001 in.

Fuel oil enters the strainer case, flows through the outer and inner windings, through the center of the elements, and out through the strainer head. Provision is made for using either one or both strainer units. When the handle on the unit is shifted to the No. 1 position, the oil is flowing through the No. 1 unit. This applies also to the No. 2 position. When the control valve is in the Both position, oil is flowing through both units. This is the position of the control valve for normal operation. The positions of the control valve and the number of the corresponding straining unit are cast into the strainer head at the control valve.

g. Fuel oil filter. The fuel oil filter is a duplex filter with provisions for using either one flow of fuel oil to the engine will be stopped if both valve handles are not pointing in the same direction when using only one filtering unit.

12A4. Lubricating oil system. a. Description. The lubricating oil pressure pump, mounted directly below the blower, draws hot oil from the oil pan through a strainer in the

pump suction line. A spring-loaded pressure relief valve is built into the discharge passage of the pump body, which bypasses excess oil into the engine oil pan. The pump forces the oil through the strainer and the cooler into the engine lubricating oil system. The upper connecting rod conduct lubricating oil to the piston cooling chamber in the top of the piston.

The camshaft drive gears are lubricated with oil from the generator end of the lubricating oil supply manifold in the engine block. Oil is piped from this

engine inlet connection, on the blower and pump drive housing, is fitted with a spring-loaded relief valve. The spring pressure is adjusted by means of a regulating screw to maintain the correct pressure. Any surplus oil is returned to the oil pan.

Lubricating oil is supplied to the lubricating oil manifold in the cylinder block. From this manifold, oil is forced through tubes to the crankcase crossframes, where it flows through oil passages to lubricate the main bearings. The crankpin bearings are lubricated with oil received from an adjacent main bearing through oil passages in the crankshaft. Oil holes in the manifold to the camshaft drive gear bearing support and to the lubricating oil distribution block in the camshaft drive housing. Lines from the distribution block carry oil to the other gear bearings in the camshaft drive and the mating teeth of the gears in the camshaft drive. The lubricating oil from the camshaft drive housing is returned to the engine oil pan by the camshaft drive housing scavenging pump.

Oil under pressure is supplied to the camshaft bore through the splined drive connection. The oil is then delivered to the camshaft

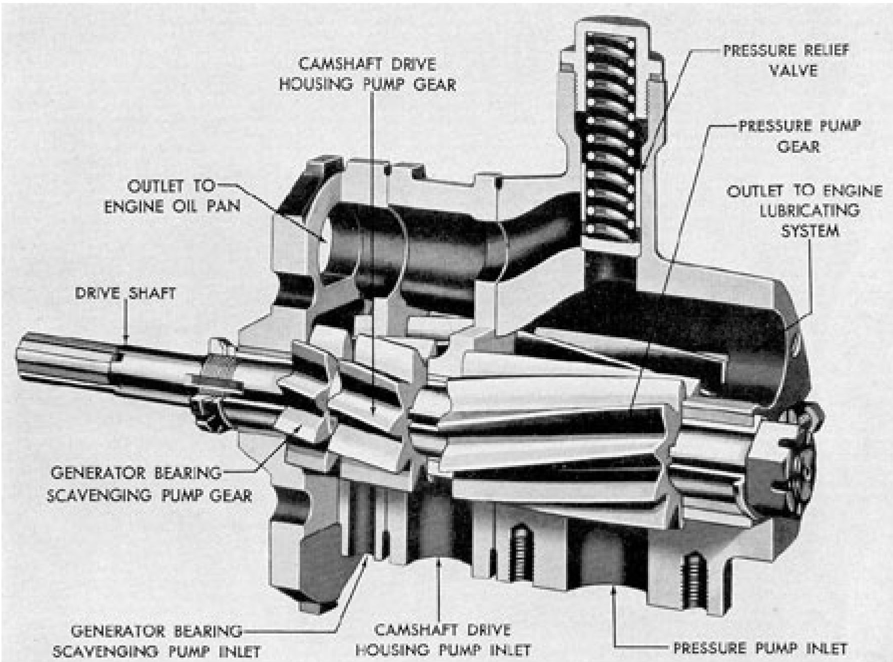


Figure 12-11. Cutaway view of GM 8-268 lubricating oil pump.

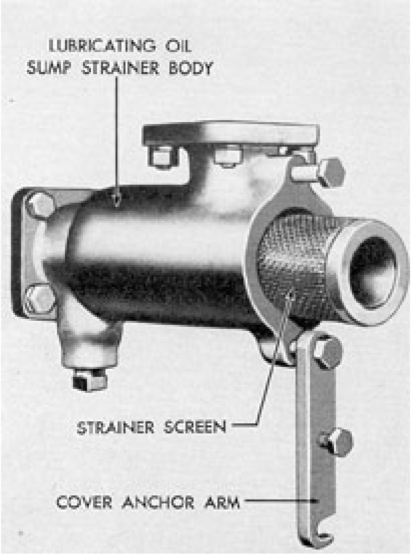


Figure 12-12. Lubricating oil suction strainer, GM 8-268.

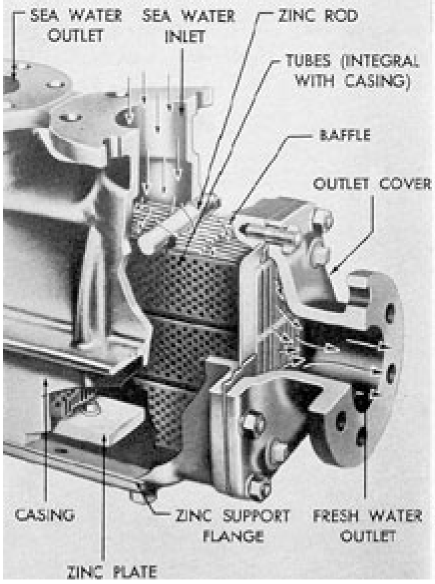


Figure 12-13. Cutaway of lubricating oil cooler GM 8-268.

bearing through radial holes in the camshaft. Oil for lubricating the rocker lever mechanism flows through tubes from the camshaft bearing caps. This oil also furnishes lubrication for the valve assembly. The oil then drains to the oil pan.

The blower and accessory drive gear bearings receive oil from the blower end of the lubricating oil pressure manifold in the engine block. Oil for the blower bearings and gears is received from the relief valve connection on the main lubricating oil manifold, and then is conducted through the tubes under the rotor housing to passages in the blower endplates, and returned to the oil pan.

b. Lubricating oil pump. The attached lubricating oil pump unit is mounted below the blower. The pump unit is of the positive displacement, helical gear type, and consists of a lubricating oil pressure pump, a camshaft drive housing scavenging pump, and a generator bearing scavenging pump. The lubricating oil draws the oil from the camshaft drive housing and returns it to the engine oil pan. The generator bearing scavenging pump draws the excess oil from the generator bearing and returns it to the engine oil pan. The pump housing is made in four separate parts: the bearing flange, the generator bearing scavenging pump housing, the camshaft drive housing scavenging pump housing, and the lubricating oil pressure pump housing. The driving gear shaft bearings are located in the pump housing. The driven gears, fitted with bronze bushings, rotate on the stationary idler gear shaft.

c. Strainers. Two types of strainers are used in this installation. The lubricating oil suction strainer is located in the pump intake line at the blower end of the engine and strains the oil entering the pump from the engine lubricating oil pan. The straining element is made of wire screen and is in the shape of a cylinder. The pump draws oil through the open end of the strainer element and sends it out through its side.

pressure pump supplies lubricating oil to the engine. The camshaft drive housing scavenging pump

Figure 12-14. SALT WATER COOLING SYSTEM, GM 8-268 AND 8-268A.

Figure 12-15. FRESH WATER COOLING SYSTEM, GM 8-268 AND 8-268A.

The other strainer in the system is the supply line strainer which is similar to the strainer found in GM 16-278A engines. The strainer case contains a cylindrical straining element of the edgewise-wound metal ribbon type. A handle on the top of the unit is used to revolve the straining element under metal cleaning blades. The strainer should be cleaned frequently when the engine is running, by turning the cleaning handle one or more complete revolutions.

The direction in which to rotate the cleaning handle is indicated by an arrow. The pressure drop through the strainer is an indication of the condition of the straining element.

The other lever on the strainer operates the bypass valve. When the lever is in the ON position the lubricating oil is flowing through the strainer. When the lever is in the BYPASS position the oil is flowing directly through the head of the unit, and the strainer case and element can be removed and cleaned. The ON and BYPASS positions are indicated on the strainer case.

manifold and into the cylinder liners through the lower deckplate in the engine block. The water is then pumped upward to the cylinder heads through the ferrules in the top of the liner. From the cylinder head the cooling water flows to the water jacket around the exhaust manifold, to the fresh water and lubricating oil coolers, and back to the pump. The fresh water system is filled through the expansion tank. Control of the fresh water temperature is by means of a temperature regulator identical with that found on 16-278A engines.

d. Fresh water and salt water pumps. The fresh water and salt water pumps are of the, centrifugal type. Water enters the center of the impeller and is thrown outward through the impeller vanes by the rotating motion of the pump.

The pump impeller is keyed to the tapered end of the driving shaft and rotates in the pump housing on two pairs of replaceable bronze wear rings.

A packing sleeve is keyed to the shaft and butts against the impeller. A watertight seal is provided by three 1/8-in. square

d. Lubricating oil cooler. The lubricating oil is cooled in a Harrison type cooler that is made up of a core assembly and an enclosing case. The oblong tubes enclose a series of baffles which form a winding passage for the flow of oil. The tubes are fastened to header plates at the ends. The core assembly is permanently attached to the casing.

12A5. Cooling system. a. General. The cooling system is of the closed type, employing fresh water to cool the engine, with salt water in the generator air coolers and acting as the cooling agent in the fresh water cooler.

b. Salt water system. The salt water pump draws water from the sea chest through a strainer and forces it through the engine water cooler and out through the overboard discharge. The pump also forces sea water through a branch line to the generator coolers. The valve controlling the flow of salt water through the generator coolers should be set to keep the temperature of air in the generator at the temperature specified in the manufacturer's instruction book.

c. Fresh water system. The fresh water pump forces the water into the engine water

plastic metallic packing rings that fit in a recess of the packing sleeve. This packing is tightened by rotating the locking sleeve with a spanner wrench, thereby compressing the packing. The sleeve is locked in place with a setscrew. The packing gland must be removed and the setscrews loosened before the locking sleeve can be tightened.

A finger, locked to the shift with a setscrew, throws off any water that may work its way along the shaft toward the ball bearing. The ball bearing is pressed on the shaft and can be removed only with the bearing puller furnished for this purpose. This bearing is lubricated by splash from the accessory drive. A leather seal prevents the oil from leaking out of the bearing housing.

e. Fresh water cooler. The engine water is cooled in a Harrison type cooler consisting of a core assembly and an enclosing case. The oblong tubes are baffled to form winding passages for the flow of engine water. The tubes are fastened to header plates at the ends. The core assembly is permanently attached to the casing.

12A6. Air intake and exhaust systems. a. General. An air blower scavenges the engine

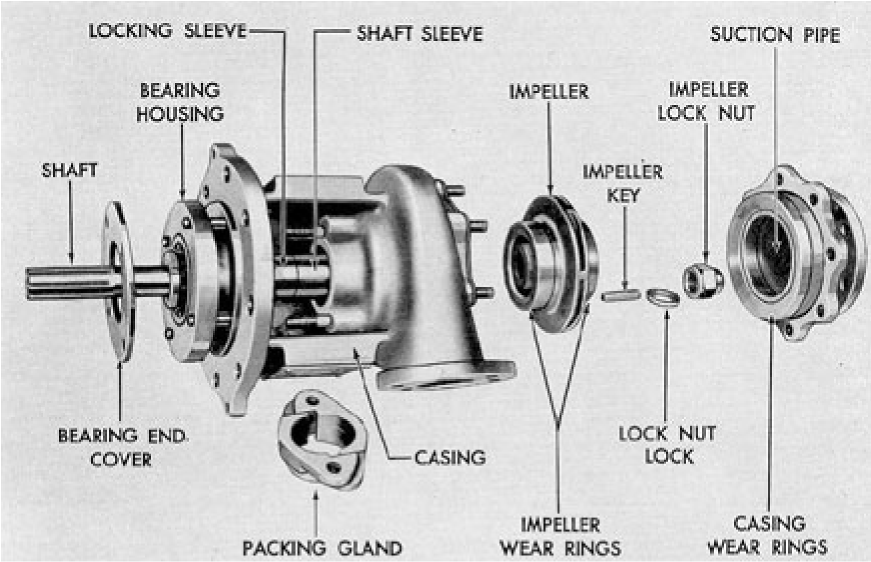


Figure 12-16. GM 8-268 water pump disassembled.

cylinders by forcing air through the intake ports in the liners as the pistons approach the end of their power strokes. This air forces out the burned exhaust gases through the open exhaust valves in the cylinder head.

Air is drawn by the blower through an intake silencer and is discharged through a distributor manifold into the air box surrounding the cylinders. Air is admitted to each cylinder when the piston uncovers the intake ports. These ports are designed to produce a swirling flow of air upward through the cylinder toward the exhaust valves which open for the discharge of the exhaust gases. This results in complete scavenging and filling of the cylinders with clean air.

The exhaust gases from each cylinder are discharged into a water-jacketed manifold, which in turn discharges the gases into one of the main engine exhaust pipes (usually No. 3ME) and thence to the atmosphere.

The cooling water flows from each cylinder head into the water passages of the manifold. From the manifold the water passes through an elbow into the expansion tank. (See Section 12A5.)

Thermocouples for measuring the temperature of the exhaust gases from each cylinder are located in the manifold.

b. Blower. The blower consists of a pair of rotors revolving together in a closely fitted housing. Each rotor has three helical lobes which produce a continuous and uniform displacement of air. The rotors do not touch each other or the surrounding housing. Air enters the housing at one side and fills the spaces between the rotor lobes as they roll apart. The air is carried around the cylindrical sides of the housing, into the closed spaces between the lobes and the housing, and is forced under pressure to the discharge side of the housing as the lobes roll together. Then the air passes through a distributor manifold into the air box around the cylinder liners.

Each rotor is carried by a tubular serrated shaft. Endwise movement is prevented by two taper pins. No gaskets are used between the

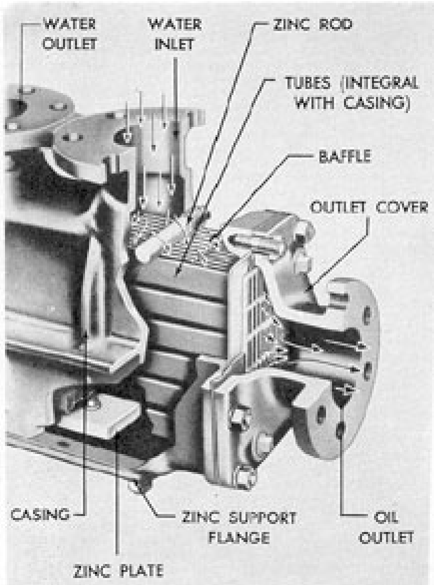


Figure 12-17. Cutaway of fresh water cooler, GM 8-268.

endplates and the housing due to the importance of maintaining the correct rotor end clearance. A fine silk thread around the housing, inside the stud line, together with a very thin coat of non-hardening gasket compound, provides an airtight seal.

Large babbitted bearings in the endplates accurately locate the rotors in the two half-bores of the housing so that the clearances between the rotor tips and the housing bores can be held to a minimum. Both ends of the rotor bearings at the gear end of the blower are made with thrust surfaces that locate the rotor endwise and prevent contact between the rotors and the endplates.

The power to drive the blower is transmitted directly to the rotor gear train by a drive shaft that extends through a passage in the blower housing. Closely fitted hexagon head lockscrew, threaded in the rotor shaft, holds a thrust collar as a spacer between the gear hub and the end of the rotor, maintaining the clearance between the rotors and the blower endplate.

The blower rotor gears are bolted to the gear hub flanges and are located angularly by hardened dowel pins. Due to the importance of having the rotors roll together without touching, yet with the least possible clearance, it is necessary to locate the dowel pins during the assembly for a given set of gears and hubs. This is the only adjustment provided for timing the gears with respect to the rotors.

Oil passages in the endplates conduct lubricating oil under pressure to the bearings. Oil seals are provided at each bearing to prevent oil from entering the rotor housing.

c. Air maze. A breather system is used to prevent contamination of the engine room atmosphere by heated, fume-laden air that otherwise would escape from the engine crankcase. This ventilation of the crankcase also reduces the formation of sludge in the oil and prevents the accumulation of combustible gases in the crankcase and oil pan.

Atmospheric air for the breather system enters the engine through the cylinder head cover breathers. The blower draws air from the crankcase through the air maze

helical rotor gears are rigidly attached to both rotor shafts to prevent the rotors from touching each other as they roll together. Each hub is pressed on the serrated rotor shaft. A which prevents oil mist from being drawn into the blower.

The air maze element consists of a number of fine steel and copper wire screens. Oil laden air is drawn through the air maze screens. The oil deposited on the screens, drains to the bottom of the air maze housing. This separated oil is returned to the accessory drive cover through a drain tube.

12A7. Air starting system. High-pressure air is piped to a lever-operated air starting valve. When the lever opens the valve, it allows the air to flow through the starting air manifold in the cam pocket of the crankcase to the individual air distributor valves or air timing valves in the rocker lever bearing support at each cylinder. The distributor valve is of the poppet type and is operated from the narrow earn in each group of four on the engine camshaft. Starting air from each distributor or timing valve is conducted

through passages in the rocker lever bearing support and cylinder head to the air starter check valve. The joint between the rocker lever bearing support and the cylinder head is made airtight with a metal ferrule and a neoprene gasket. The air starter check valve prevents exhaust gases from entering the air starting passage. It is opened by the high-pressure starting air and closed by a spring when the starting air from the timing valve is cut off. From the check valve the starting air flows into the space above the piston and forces the piston downward until the air distributor valves closes and the exhaust valves open.

B. FAIRBANKS-MORSE 38E 5 1/4 ENGINE

12B1. General. The F-M 38E 5 1/4 7-Cylinder engine is used as an auxiliary engine on submarine

12B2. Operation. The opposed piston engine is of the solid injection, inlet and exhaust port,

whose main propulsion engines are Model 38D 8 1/8 Fairbanks-Morse engines. Like the GM 8-268 engine, it is located on the lower deck level of the after engine room and may be used to carry the auxiliary load, to charge batteries and indirectly for propulsion. The engine is of the opposed piston type, with 7 cylinders in line and air started, and is rated at 300 kw generated output at 1200 rpm. It works on exactly the same principle as the F-M 38D 8 1/8, and most of the parts are identical in design, the only difference being in the size and dimension of the parts.

scavenging blower type and is designed to use a variety of fuels. The two pistons in each cylinder work vertically against each other, forming a single combustion space between the pistons at the center of the cylinder. The cross sections of the engine show the relative positions of the blower, crankshaft, pistons, and generator.

The engine operates on the two-cycle principle in which two strokes of each piston and one complete revolution of each crankshaft are necessary to complete the cycle. The cycle begins with the movement of the pistons from their outer dead centers. As the pistons move

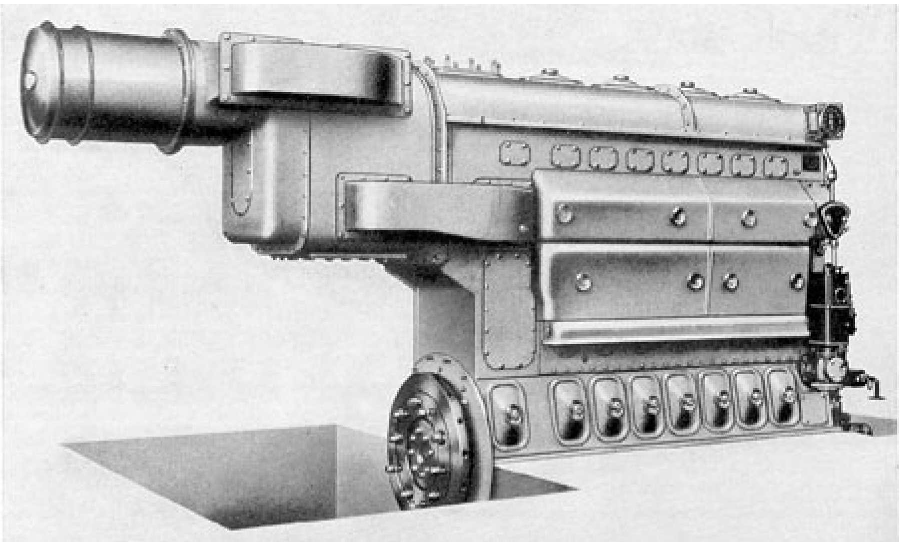


Figure 12-18. Control side of 7-cylinder F-M auxiliary engine.

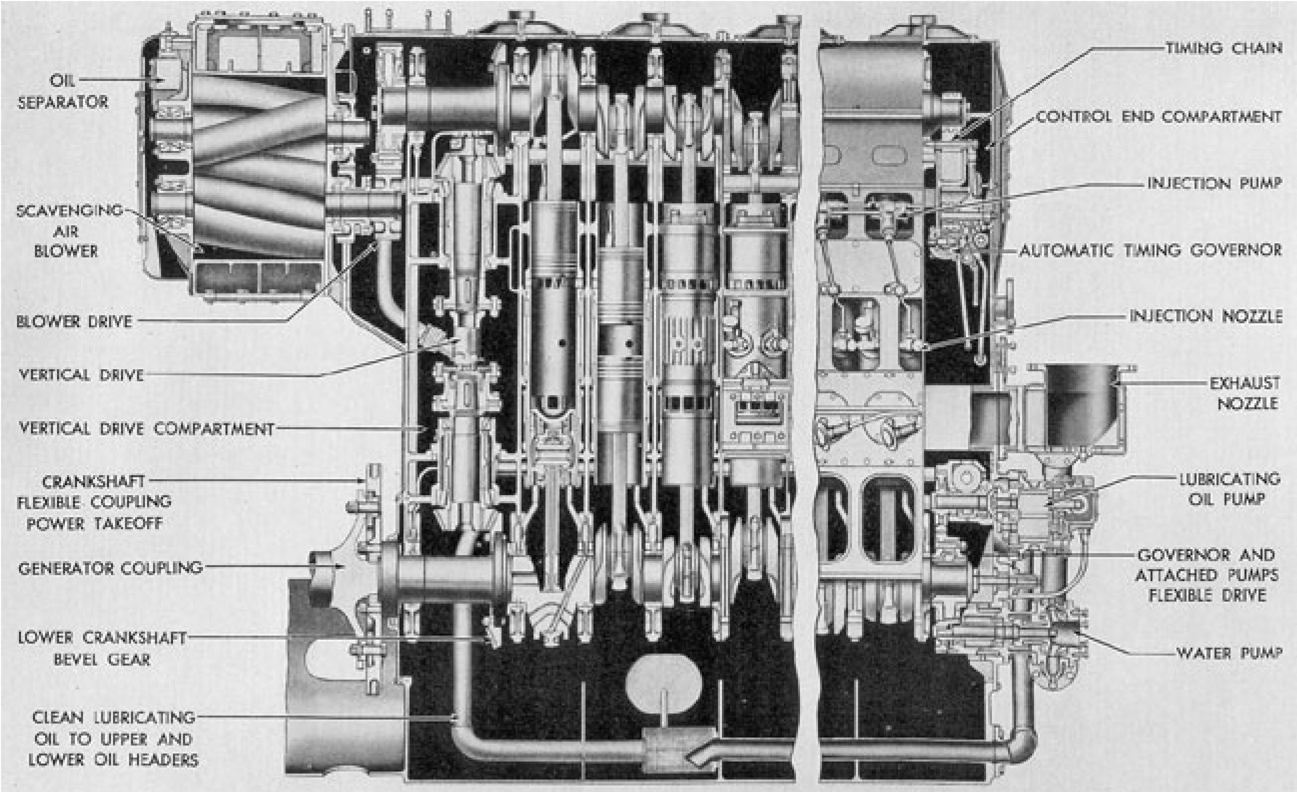


Figure 12-19. Longitudinal cross section of 7-cylinder F-M auxiliary engine.

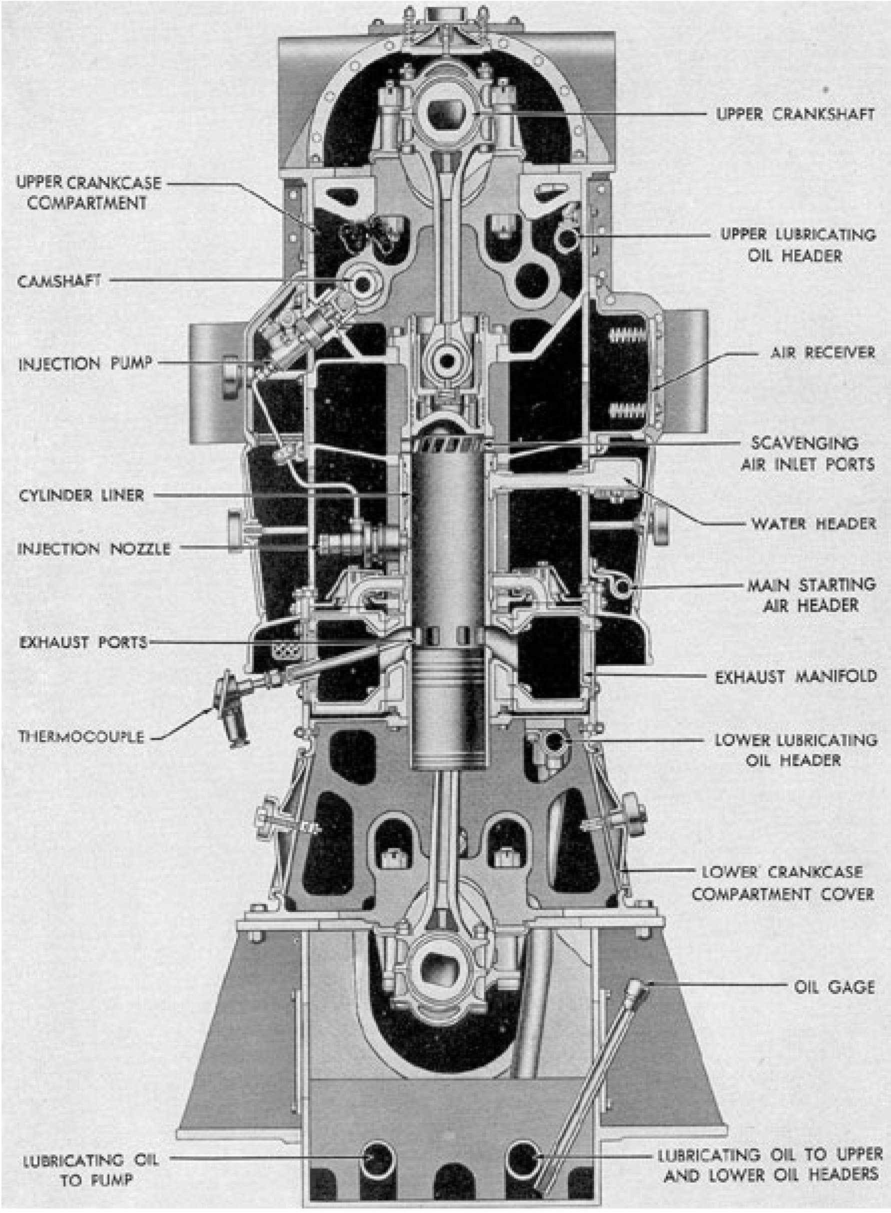


Figure 12-20. Transverse cross section of 7-cylinder F-M auxiliary engine.

inward they cover the exhaust and inlet ports and start to compress the air in the cylinder. As the pistons approach combustion dead center, fuel is injected into the combustion space in a fine spray. The fuel immediately starts to burn and expansion follows, forcing the pistons outward and delivering work to the crankshafts. The shafts are connected by a vertical gear drive.

Toward the end of the expansion stroke, the lower pistons uncover the exhaust ports and allow the

Figure 12-20 shows a transverse cross section of the working cylinder.

12B3. Engine main moving and stationary parts. a. Cylinder block. The cylinder block is the main structural part of the engine and is designed to give it the necessary strength and rigidity. It is constructed of hot rolled steel plates of the proper dimensions welded into a single unit, combining compactness and strength with lightness of weight.

burned gases to escape to the atmosphere through the exhaust system. Soon afterward, the upper pistons uncover the air inlet ports. At this point the pressure in the cylinder is about atmospheric. As the inlet ports are uncovered, the scavenging air under pressure in the air receiver rushes into the cylinder with a whirling motion, sweeping the cylinder clear of any remaining exhaust gas and filling it with fresh air for the next compression stroke. The whirling motion or turbulence of the air is obtained by the tangentially cast inlet ports. This turbulence persists throughout the injection period and aids in the mixing of the air and fuel.

With this arrangement of the pistons and crankshafts as described above, the lower crankshaft will lead the upper crankshaft by approximately 12 degrees. The difference in the crankshaft setting is referred to as the lower crank lead. The two pistons will be the nearest together when the upper piston is approximately 6 degrees ahead of inner dead center and the lower piston is 6 degrees past inner dead center. The point midway between the two pistons when they are in this position is called the piston dead center.

From the foregoing it can be seen that when the upper piston reaches inner dead center, the lower piston will have completed 12 degrees of the expansion or power stroke. This causes the lower piston to receive the greater part of the expansion force with the result that at full load about 70 percent of the

Transverse vertical members together with horizontal decks form enclosures, housings, and fastenings for the operating parts of the engine. The four horizontal decks are bored to receive the cylinder liners along the centerline of the engine. An extension of the block is provided for attaching the scavenging air blower at the vertical drive end of the engine.

The cylinder block is separated into the following compartments:

1. The control end compartment which forms an enclosure for the timing chain, controls, overspeed and timing governors, and drives for the regulating governor and attached pumps.

2. The vertical drive compartment which houses the vertical gear drive that interconnects the upper and lower crankshafts. Used oil from the drive gears and upper crankcase compartment drains down through this compartment to the oil pan or engine sump.

3. The upper crankcase compartment which forms the bearing saddles for the upper crankshaft bearings and bearings hubs for the camshaft bearings. The saddles and bearing hubs are drilled for passage of lubricating oil to the bearings from the upper oil header. The used oil from these parts collects on the floor of this compartment and drains way from the center of the block toward each end where it passes through openings to the engine sump or oil pan.

4. The air receiver compartment which extends the full length of the block and completely surrounds the cylinder liners at the

total load is delivered by the lower crankshaft. The remaining power is delivered to the upper crankshaft where it is partially utilized in driving the scavenging blower. Any residual power is transmitted through the vertical drive gear to the lower crankshaft which is connected to the generator.

air intake ports, forming a passage for scavenging air from the blower to the inlet ports of the cylinder liners.

5. The injection nozzle compartments

which form enclosures for the injection nozzles, air start check valves, and cylinder relief valves. The injection pump, which is cam actuated, is located at an angle in the underside of the upper crankcase on the control side and furnishes fuel under pressure to the injection nozzle.

6. The exhaust compartment which extends the full length of the block on each side. The exhaust decks are bolted to the block and surround the cylinder liner, forming water-cooled passages for the exhaust gases from the combustion spaces to the exhaust manifolds. The manifolds are water-cooled welded steel units, bolted to the cylinder block and exhaust decks.

7. The lower crankcase compartment which forms bearing saddles for the lower main and thrust bearings. These saddles are drilled for the passage of lubricating oil to the bearings. The used oil from the engine collects in the oil pan or engine sump.

After welding, the block is sand blasted and stress relieved to remove the internal strains at cylinder liner jackets which are pressed on the outside of the liners. They extend from the bottom of the scavenging air inlet ports to a short distance above the exhaust ports.

Openings for the air start check valve, cylinder relief valve, and injection nozzle are located in the liner and liner jacket at the point where the pistons arrive at inner dead center.

The channels directing the cooling water up and around the cylinder liner are cast in between the ribs on the liner. The heat of combustion that accumulates in the cylinders, as well as the heat conducted to the air start check valve, cylinder relief valve, and injection nozzle, is transferred to the cooling water which flows into these passages through regular fittings from the water jacket of the exhaust manifold. The water leaves the cylinder liner near the top of the jacket by means of an outlet pipe to the water header.

c. Crankshafts and main bearings. The crankshaft is a heat-treated, cast iron shaft with an allover finish. It is held in the cylinder block by main bearing caps. The

certain vital points. It is then magnafluxed to check for the presence of any construction faults. The various compartments are provided with covers. The upper crankcase compartment is closed by a top cover bolted to the block. The cover has several inspection covers along the top, one of which is an explosion cover to relieve any excess pressure built up in the upper crankcase. An explosion cover on the control end of the block above the exhaust nozzle prevents the possibility of excess pressure building up in the lower crankcase.

The main and thrust bearing saddles are machined together with the forged steel bearing caps. These are match-marked to show proper position. When a replacement cylinder block is ordered the bearing caps are always furnished in place.

b. Cylinders and cylinder liners. The cylinders are formed by cylindrical sleeves or liners located on the centerline of the engine and spaced to correspond with the crankshaft throws. The cast iron liners have specially designed air inlet and exhaust ports for the passage of scavenging air to, and of exhaust gases from the combustion space.

The liner cooling spaces are formed by

bearing at the blower end of the upper crankshaft and at the coupling end of the lower crankshaft act as combination main and thrust bearings. The crankshaft is drilled from each main bearing journal to each connecting rod journal so that oil, furnished to the main bearing saddles from the oil headers, flows into the crank and into each connecting rod journal. The combination main and thrust bearings are not connected to any connecting rod journal; accordingly they are not drilled, but receive their oil from separate tubes direct from the oil headers.

d. Vertical drive assembly. The power developed by the upper crankshaft is transmitted to the lower crankshaft by means of a gear drive which is turned by the crankshaft gear bolted to the blower end of each crankshaft. The drive assembly consists of two tapered shafts with pinions, connected together by a flexible coupling with laminated rings between the hubs, coupling shaft, and adjusting flange. The upper and lower housings contain a large and a small set of thrust bearings.

e. Connecting rod and connecting rod bearings. The connecting rod is an alloy steel forging with a closed eye at one end and a removable

cap at the other end. The connecting rod crankpin bearing is made up of a cap bearing shell

are the bearings at the blower end of the camshaft stub shaft, which are held in place by a bearing nut.

and a rod bearing shell. The bronze-backed bearing shells are lined with a high lead bearing composition containing a special hardener and known under the trade name of Satco metal. Each connecting rod crankpin bearing shell is identified by a mark stamped on the edge of the shell. New shells may be installed without fitting or scraping. The piston pin bearing consists of two rows of hardened steel rollers or a bronze bushing fitted in the space between the piston pin and the connecting rod bushing.

f. Piston and piston pin assembly. The upper and lower cast iron pistons are identical except for the position of the fuel opening in the cup at the top of the piston. The pistons are marked and should be installed in their proper places, so that the fuel openings line up correctly with the injection nozzle openings in the liners.

The pistons are connected to their respective crankshafts by the piston pin bracket, piston pin, needle bearings or bronze bushing, and forged steel connecting rods, and are cooled by lubricating oil as described later.

Each piston is fitted with seven cast iron rings, four of which are compression rings located above the piston pin and three of which are oil rings located below the piston pin. Of these three rings there is one oil scraper, one oil drain, and one oil cutter ring.

The condition of the lower piston and rings may be observed

Oil is supplied to the control end bearings by a pipe that leads directly to the camshaft bearing saddle from the upper oil header. The oil flows through the hollow shaft and supplies oil to the rest of the bearings by means of the radial holes drilled in the shaft.

h. Timing chain. The timing chain is the means by which the rotation of the upper crankshaft is conveyed to the camshafts, turning the camshafts at the same rate of speed as the crankshaft. Sprockets are fastened to the control end of the crankshaft, camshaft, and stub shaft. The chain is guided by special links over the crankshaft sprocket, under a timing chain sprocket, and over the camshaft sprocket. It then passes under a tightener sprocket, over the stub shaft sprocket, and under a second timing chain sprocket to the crankshaft.

The timing chain is a No. 766 Duplex 1/2 in. pitch, 2 in. wide, center guide, endless chain of 116 pitches. It is assembled to operate the links, to guide the chain on the camshaft and crankshaft sprockets, and to operate in slots in the tightener and timing sprockets.

i. Hand control lever. The engine is started and stopped by means of a hand-control lever. This lever has three positions: START, STOP, and RUN. In the STOP position, the fuel cutout cam on the control shaft moves the fuel injection pump control rod to the no fuel position. When the lever is in the START position, the control shaft mechanism moves the air control valve plunger and opens the control valve, admitting air to the engine air header and air

through the exhaust manifold and ports after the thermocouple or plain covers have been removed from the manifold.

g. Camshaft, camshaft stub shaft, and camshaft bearings. The camshaft is of the one-piece type with integral cams on a case-hardened alloy steel shaft. There is one cam on the camshaft for each cylinder. This cam actuates the injection pump plunger. There is only one camshaft on this model F-M engine. On the other side of the engine from the camshaft is a camshaft stub shaft which is used to drive various governor auxiliaries and to form a part of the timing chain system. This stub shaft is extremely short, extending for the length of one cylinder only.

The camshaft bearings are held in place in the block by special setscrews. Exceptions to this

distributor. In the RUN position, the engine is under full governor control.

j. Emergency stop and reset lever. The engine is equipped with a hand-operated emergency stop device, consisting of a push button that operates against the overspeed latch and releases the overspeed stop plunger, shutting off the delivery of fuel to the injection pumps. This emergency stop may be connected to the ship's air supply so that it can be operated from the maneuvering room by the use of a quick opening valve to stop the engine. When air pressure is admitted to the emergency stop housing, the overspeed stop latch is tripped and shuts off the

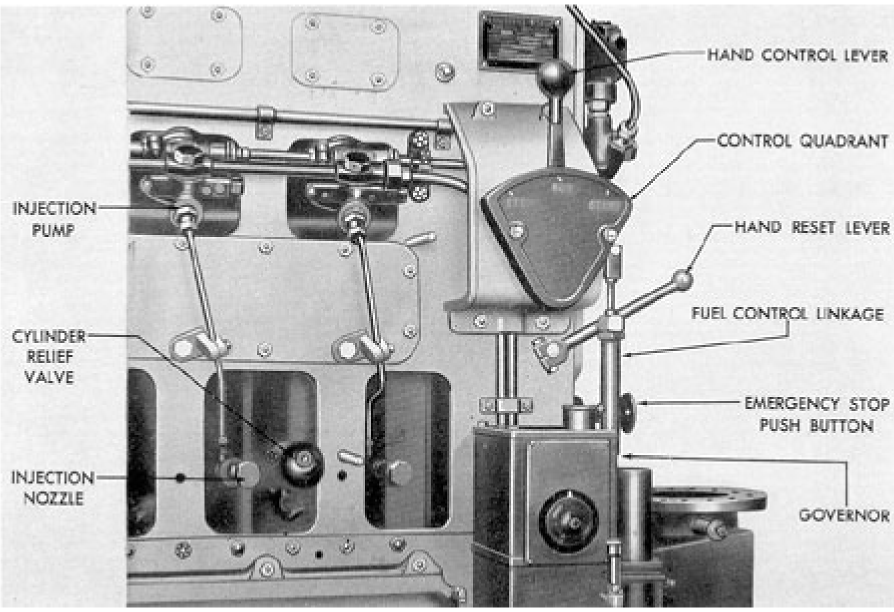


Figure 12-21. Engine controls, end view, F-M auxiliary engine.

supply of fuel as described in the preceding paragraph.

should be inspected to see that it is operating at the correct speed.

k. Overspeed governor. The overspeed governor mechanism automatically stops the engine when the main governor fails to hold the engine speed below the safe maximum of 1290 to 1370 rpm. This mechanism consists of a weight on the end of, and rotating with the camshaft. When an overspeed condition occurs, movement of the governor weights releases the spring-loaded stop device as described below.

When the predetermined safe speed of the engine is reached, the centrifugal force of the governor weight will overcome the opposing pressure of a spring and allow the governor weight to swing outward and strike the governor lever. This trips the overspeed stop latch and releases the overspeed stop plunger which moves fuel cutout lever and shaft, shutting off the supply of fuel to the injection pumps. The speed at which the governor weight will strike the governor lever can be adjusted by the addition or removal of shims. At regular intervals this governor

In order to start the engine after it has been stopped by either the overspeed governor or the emergency stop, the plunger must be returned to its spring-loaded normal position. This is accomplished by moving the reset lever which in turn moves the reset shaft and pulls the stop plunger up, thus compressing the spring to a position where the overspeed latch will be brought up back of the head on the plunger by the stop latch spring. The reset lever is then in its normal position and the engine is ready to operate.

1. Flexible drive. The flexible drive transmits power from the lower crankshaft to drive the governor, air start distributor, lubricating oil pump, and fuel oil and generator bearing drain pumps.

The pump drive hub is pressed on to the lower crankshaft, and the pump drive gear is bolted to it.

The lower torsion graph drive shaft is

bolted to the end of the louder crankshaft and aids in holding the hub on the shaft. The pump driven gear is keyed to the pump drive shaft which is located in the pump drive housing. The housing is bolted to the engine block and supported by the pump mounting plate.

The power take-off for the air start distributor and the

and the housing, and between the impellers and bearing plates are reduced to minimum. Under no circumstances should oil be allowed to leak into the blower housing or air receiver.

b. Oil separator. The engine crankcases, vertical drive, and control end compartments are vented by means of the suction of the blower. A slight vacuum is

governor is located in the pump drive housing and consists of a gear keyed to the pump drive shaft and driving a final spiral gear.

The lubricating oil pump is bolted to the pump mounting plate and is connected to the pump drive shaft by a positive drive coupling. At the end of the lubricating oil pump impeller driving shaft is a set of beveled gears that drive the fuel oil and generator bearing drain pumps. These pumps are located on each end of the fuel pump drive housing.

The fresh water and salt water pumps are driven directly from the pump drive gear.

13B4. Scavenging system and blower. a. General description. Scavenging air is supplied to the cylinders under a pressure of from 2 to 5 psi by means of a positive displacement type blower. The blower consists of a housing containing inlet and exhaust passages enclosing two three-lobe spiral impellers. Timing gears, driven by a gear drive from the upper crankshaft, interconnect the impellers.

Air is drawn from the atmosphere through the air silencer and enters the inlet passage of the blower. It is moved by the lobes along the walls of the blower housing and forced through the outlet passages. Pipes conduct the scavenging air to the air receiver compartments on each side of the cylinder block. These receivers are the full width of the cylinder block and extend to the produced by suction through an oil separator located inside the blower end cover. This should be adjusted to 2 in. maximum water vacuum by the screw on top of the oil separator in the blower end cover. Passage to the oil separator is through the hollow impeller shafts.

The separator consists of a metal box, with small holes drilled in the front piece, filled with copper gimp upon which the oil collects as the air passes through it. The accumulated oil is drained off each end of the separator, collecting in the lower compartments of the blower and draining back into the crankcase.

The separator should be removed and cleaned in kerosene at regular intervals. The excess oil should be blown from the copper gimp before placing the separator back into service. If for any reason the separator is neglected, the seepage of oil from the lower crankcase side cover or a possible smoky condition of the exhaust will indicate that it should be cleaned.

12B5. Fuel system. a. Description. The fuel oil supply system is composed of a standby and priming pump, an attached fuel oil pump, and a fuel oil strainer-filter with the necessary relief valves and piping.

The standby and priming pump is used to fill the complete fuel system prior to first starting the engine or after the engine has been over hauled. As the engine starts to turn, the attached fuel oil pump takes over the task of supplying fuel to the engine and

control end compartment. They completely surround the cylinder liners at the air inlet ports. The scavenging air enters the cylinder under pressure, and sweeps the exhaust gases out through the exhaust ports, producing complete scavenging. A quantity of scavenging air is trapped in the cylinder by the pistons, thus providing fresh air for the next compression stroke.

The scavenging air is discharged from the blower with a uniform velocity due to the design of the impeller lobes. For greatest efficiency the clearances between the impellers, the impellers

forces the fuel past a relief bypass valve through the fuel oil strainer-filter to the engine header where each individual injection pump is supplied. The overflow from this header is directed, via a relief valve, to the clean fuel oil tank. Dirty oil from the injection nozzle compartment flows back to the leakage tank or to the bilges depending on the particular installation.

The capacity of the supply pump is such

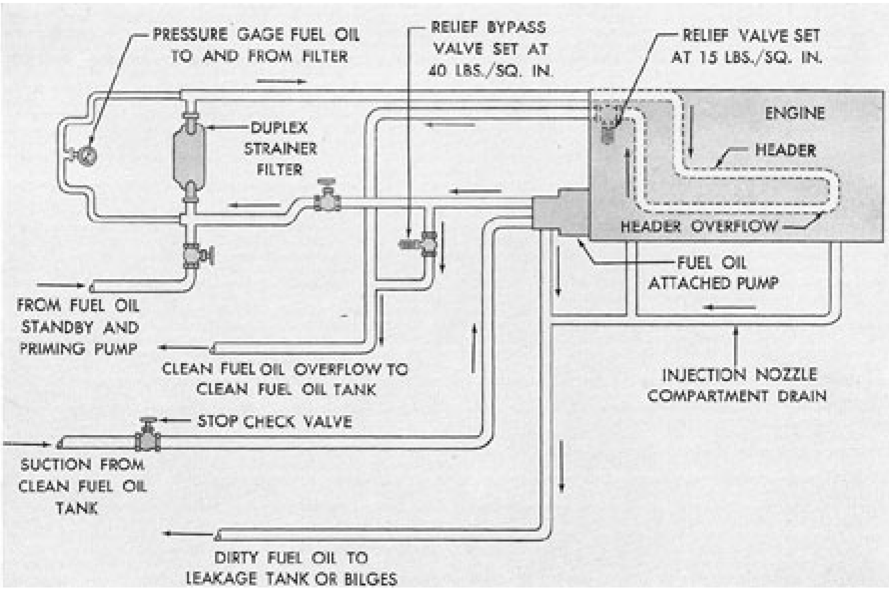


Figure 12-22. Fuel oil piping, F-M auxiliary engine.

that a sufficient velocity of fuel is obtained in the fuel oil header to insure rapid replacement of fuel as it is used by each individual injection pump at full speed.

b. Fuel pump. The attached fuel oil supply pump is a positive displacement pump. It should require no attention other than an occasional inspection. The

stroke, lapped plunger, cam-actuated type and is identical in principle to the pump used on the Model 38D 8 1/8 engine. The pump measures the correct amount of fuel and delivers it at the correct moment to the injection nozzle from which it is injected into the combustion space between the pistons.

packing gland should be tightened or repacked as found necessary. It is very important when installing pump packing that the rings be cut to the exact lengths. The joints of the packing should be alternated so that they do not come in line with each other. Leakage should be permitted through the gland after the packing is first installed. The gland should then be set up in small increments with several minutes between tightening in order to permit the packing to adjust itself to the shaft gradually.

c. Injection system. The injection system for each cylinder is made up of the injection pump, injection nozzle, and the tubing connecting the two units.

1. The injection pump is of the constant

The injection pumps are enclosed in the cylinder block in an inverted position on the control side of the engine below the camshaft. The camshaft is driven through a silent chain by a sprocket on the control end of the upper crankshaft. Each pump consists of a housing, plunger, barrel, control rack, plunger spring, delivery valve, delivery valve seat, and delivery valve spring. The push rod assembly transforms the rotary motion of the camshaft into linear up-and-down motion of the plunger.

The injection pump plunger moves in the plunger barrel with a constant stroke and delivers fuel through the delivery valve and injection tubing to the injection nozzle and on to the combustion space in the cylinder liners. The

plunger stroke remains constant, and the amount of fuel is varied by rotating the plunger in the barrel by means of the serrated control rack acting upon and meshing with the pump plunger control sleeve.

When the plunger is in its highest position, the fuel inlet port is uncovered and the pump barrel fills with fuel. As the plunger moves down, the port is covered and fuel is delivered to the combustion space. Delivery of fuel continues until the helical edge of the plunger uncovers the bypass port. Fuel not required for combustion is discharged through the bypass

discharge tubing, no oil can drain out of the tubing. The seating of the delivery valve causes the injection nozzle to close sharply and prevents a dribbling of oil from the nozzle.

2. The fuel injection nozzle consists primarily of a nozzle body, nozzle spring housing, needle sleeve, needle, push rod, spring, filter, shim and nozzle tip. On the down stroke of the injection pump plunger, fuel enters the injection nozzle through the injection tube and is forced through the nozzle filter. The nozzle filter removes any foreign matter which may have gone through the main filters.

port to the suction header. (See Figure 5-23.)

The exact position in the plunger stroke at which the helical edge uncovers the bypass port depends on the rotary position of the plunger. When the plunger is in the stop or no fuel position the vertical groove and the helical edge of the plunger keep the bypass port uncovered during the entire plunger stroke, bypassing all the fuel.

Rotary position of the plunger is controlled by the regulating governor through its linkage with the fuel control rod and injection pump control rack. The control rack is connected to the fuel control rod of the governor linkage at the shifter sleeve and to the injection pump at the plunger control sleeve. The sleeve is toothed at one end and slotted at the other end. Lugs on one end of the pump plunger fit into the slots of the control sleeve.

For an increase in fuel, the shifter sleeve moves the control rack through the shifter sleeve key. For a decrease in fuel, the control rack is moved by the control rod shifter sleeve spring. This flexible design is used to actuate the control racks so that if any of the racks sticks while the engine is running, the remaining pumps can be operated to decrease the fuel injected.

In order to cut out any individual pump while the engine is in operation, the shifter sleeve should be rotated until the slot in the sleeve can pass over the

The filter built into this nozzle is extremely simple. It is a close fit in the nozzle body with a clearance of .0015 in. to .0022 in. for fuel to pass from one groove to another. The longitudinal grooves are connected alternately with the annual grooves so that fuel entering the annular groove is forced through the space between the filter and the nozzle body into the annular groove connected to the opposite end of the filter. The filtered fuel is forced into the chamber through the flutes and holes, into the outside of the needle sleeve where it enters the chamber at the face of the needle seat.

The fuel under pressure acts against the face of the needle lifting it from its seat. The pressure of the oil is counteracted by the spring through the nozzle push rod. This permits the pressure of the fuel to build up to about 3000 lb. When it reaches this point, the nozzle needle opens, allowing fuel to escape through the three small holes in the nozzle tip into the combustion space of the cylinder in a fine spray. The fast action of the needle caused by the fuel, acting on the face of the needle, and the spring counteracting this pressure, insures quick opening and closing of the needle and eliminates dribbling or leaking.

12B6. Lubricating oil system. a. Description. Lubricating oil is supplied to the system under pressure to insure a continuous flow of oil to all surfaces requiring lubrication and to the pistons for cooling. The system is comprised of the attached lubricating oil pump, oil pan, strainer, oil

sleeve key of the fuel control rod. This will permit the movement of the individual control rack to the stop or no fuel position.

The delivery valve seats when the pressure of oil in the pump chamber is relieved. Because of the spring and the high oil pressure in the

separator, filter and cooler with the necessary valves and piping.

An oil gage connection to the gage board

is provided from the discharge of the lubricating oil pump and upper oil header. A marked oil gage stick is located at the control end of the oil pan for checking the level of the oil in the engine.

The lubricating oil pump, located on the control end of the engine, draws oil from the oil pan through a strainer and discharges it through the filter and cooler to the lower and upper oil headers. Through branches from these headers, oil is supplied to each main bearing and to the thrust bearing. From the main bearings, oil passes through holes drilled in the crankshaft and through tubes swedged into the crankshaft to each crankpin bearing. Oil is then forced through the passages drilled the length of the connecting rod to the piston pin needle bearings or bronze bushing and to the piston oil pockets for cooling the pistons. An opening through the upper crankshaft lubricates the timing chain.

The cooling oil from each lower piston is discharged through the

circulating water pump, lubricating oil pump, fuel oil and generator bearing drain pumps. A tube from the upper header supplies oil to the automatic timing governor for the operation of that unit.

The camshaft and stub shaft bearings are lubricated by pipes from the upper oil header to the first bearing of each shaft. Oil enters the hollow shafts and lubricates the rest of the bearings through openings drilled radially in the camshaft.

Used oil from the bearings, tappet housing, and pistons collects in the upper crankcase compartment where it drains either toward the control end or toward the blower end of the block and flows back to the oil pan. Used oil from the blower gears and bearings collects at the bottom of the blower inner bearing plate and at the bottom of the blower end cover. An oil tube connects the blower end cover compartment to the inner bearing plate compartment allowing oil to drain from these areas to the oil pan, via the vertical drive compartment.

An oil separator is fastened inside the blower end cover and vents

piston cooling bracket outlet pipe into the sump or oil pan. Oil from each upper piston is discharged through the piston cooling bracket outlet pipe into the upper crankcase compartment where it can drain toward either end of the engine and flow back to the engine sump.

Branches from the lower oil header supply oil to the thrust bearing, crankshaft vertical drive gears, and main bearings. The bushings of the flexible pump drive located on the control end of the lower crankshaft, receive lubrication through an opening in the lower crankshaft from the control end main bearing. The surfaces of the lower thrust bearing shell and the crankshaft flange are lubricated by openings in the bearing shell.

Connections from the upper oil header supply lubricating oil to the vertical drive gears and bearings, to the blower flexible drive wear rings, to the inner and outer blower impeller bearings, and to the injection pump tappet housing. The blower drive gears are lubricated by a splash system obtained from a special fitting that directs a spray of oil to the gears.

Fittings at the control end of the upper oil header supply lubrication to the timing chain and control mechanism. This oil drains down and also lubricates the governor drive and gears, the crankcase of fumes.

b. Lubricating oil pump. The lubricating oil pump is attached to the pump mounting plate on the control end of the engine below the exhaust nozzle. It is of the positive displacement type, driven by a pump driveshaft through the flexible coupling. A built-in relief valve is set to operate at 60 pounds' pressure to relieve the pump when excess pressure is built up. Oil from the relief valve flows directly into the oil pan.

The pump should require very little attention except for periodic removal for inspection and the cleaning and grinding of the seat of the relief valve with the tool furnished for that purpose. When reassembling the pump, carefully adjust the valve to open at the proper pressure.

Special pullers are furnished among the tools for removing the drive gear, the timing gear, and drive coupling. Spanner wrenches are provided for removing the locknuts on the drive gear, the timing gear, and coupling.

c. Generator bearing drain pump. A small gear type drain pump, mounted on the front of the attached lubricating oil pump at one end of

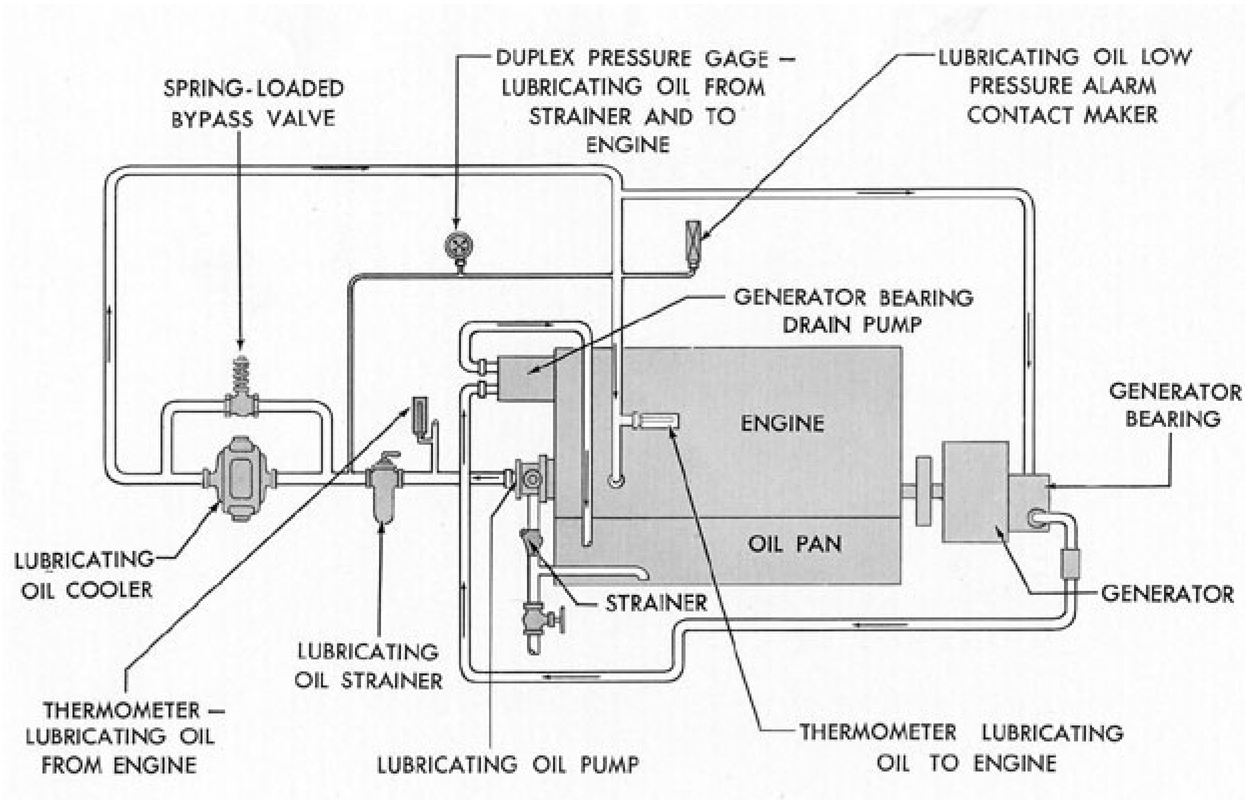


Figure 12-23. Lubricating oil piping, F-M auxiliary engine.

the fuel pump housing, is driven by a beveled gear on the extension of the upper impeller shaft of the lubricating oil pump. This pump takes the warm used oil from the closed generator bearings and returns it to the engine oil pan. Cool filtered oil is furnished to the generator bearings by the lubricating oil pump.

d. Suction strainer. The lubricating oil suction strainer is located in the pump intake line. All oil from the oil pan passes through the strainer before entering the pump. The strainer consists of a piece of screen supported by a cylindrical shell which is fastened in the enlargement in the oil piping. This strainer should be removed and cleaned frequently to insure an adequate supply of clean oil to the pump.

are identical they are discussed together under the fresh water system.

c. Fresh water system. The attached fresh water pump is driven through the flexible pump drive from the lower crankshaft and draws water from the fresh water cooler and forces it into the engine at the base of the exhaust nozzle. From the exhaust nozzle, water flows around the jacketed outside of the exhaust manifold, thence through fittings into the space between the cylinder liner and jacket on each cylinder. The heat of combustion is transferred to the cooling water as it travels up and around the cylinder liner, injection nozzle, air start check valve, and cylinder relief valve. The water leaves the jacketed space of the liner by means of outlet pipes connected to the water header on the side opposite the control side of the engine. The water flows past

12B7. Cooling system. a. General. An adequate supply of cool, clean, soft, fresh water, free of scale-forming ingredients, is essential for proper operation of the engine. The sodium dichromate water treatment should be used. Salt water should not be used for cooling the engine.

The cooling system is of the closed type in which fresh water is circulated through the engine and fresh water cooler. Salt water is circulated through the fresh water cooler and through the generator air cooling system. For more detailed discussion, the two systems are discussed separately. In older installations, salt water is used as the cooling agent in the lubricating oil cooler, but in later installations, fresh water is used.

b. Salt water system. The salt water system consists of an attached salt water pump which is driven through the flexible drive from the lower crankshaft. It draws water from the suction sea chest and strainer, forces it through the generator cooling system, the fresh water and sometimes the lubricating oil coolers, and discharges it overboard. The necessary piping, fittings, and valves complete the system. A hand-operated three-way bypass valve is installed in the discharge piping from the pump to the cooler, controlling the amount of salt water passing through the coolers and thereby, to some extent, the temperature of the fresh water and lubricating oil to the engine.

a mercury bulb thermometer where the outlet water temperature of the engine is registered.

Adjustments are possible so that part of the water can be bypassed around the cooler by means of a temperature regulator similar to that used on the Model 38D 8 1/8 engine. All of the cooling passages of the engine can be drained at the exhaust nozzle. The system should be drained if the engine is to be left in freezing temperature with no protection.

d. Circulating water pumps. The pumps are of the centrifugal type with the water entering at the center of the impeller and being forced outward by the rotating motion of the impeller vanes to the pump discharge.

These pumps require very little servicing beyond the tightening or replacing of the packing and the oil retainer ring. The packing should be tightened occasionally to keep the gland from leaking. If it must be replaced, the special packing hook should be used to remove the old packing. The new packing should be cut to the correct length and the joints alternated so they do not come in line. The gland nut should then be tightened in small increments to allow the packing to adjust itself gradually to the shaft. The bearings of these pumps are lubricated from the gear drive case.

12B8. Air starting system. The air starting system is furnished with air from the ship's high-pressure air line or from air bottles. A reducing

valve (3000 to 250 psi), a relief valve, gages, and necessary valves are in the piping leading to the engine. On the engine the control valve, the air start distributor, engine starting air header, pilot air tubing, and air start check valves, complete the system.

Engine starting is accomplished by the action of the compressed air on the pistons in their proper firing order. The operation is identical with that described in Chapter 4 for a Model 38D 8 1/8 F-M engine. When the hand control lever of the fuel and governor control rod is moved from STOP to START position, the machined portion of the control valve plunger acts as a cam, forcing the control valve open, allowing high-pressure air to enter the engine starting air header and the air start distributor. The air chamber of the distributor housing the pilot valves is under pressure when the control valve is open. This pressure is greater than the force exerted by the valve springs and accordingly it forces the pilot valves down against the air start cam.

The cam is designed so that the pilot valves on high cam are closed and vented to the atmosphere. The pilot valves on low cam are open, allowing pilot air to enter the pilot tubing from the distributor air chamber. The rotation of the cam, which is

the valve spring and high-pressure air. The valve is opened when the hand control lever is moved from STOP to START position. The clearance between the valve plunger and the valve stem should be 1/16 in. This clearance can be adjusted by adding or removing shims between the valve body and plunger body.

The valve will require little servicing beyond its removal, and reseating of the valve seat with the tools furnished for this purpose. After the valve seat has been refaced, the valve should be lapped to make a positive seat and prevent leakage of air from the air inlet piping.

The air start distributor, located on the side opposite the control side of the engine below the exhaust manifold at the control end, is properly timed and marked at the factory. The distributor directs a flow of air into the combustion chamber of the engine in time with the engine firing order. Under normal circumstances it will not need to be timed again; however, should a new cam be installed, it must be retimed.

The air start check valve is located in the cylinder liner, at the injection nozzle level, on the side opposite the control side of the engine. It is actuated by a supply of pilot air from the air start distributor which acts against the pressure of the valve spring forcing the operating piston to

driven through the flexible pump drive from the lower crankshaft, causes the pilot valves to open in time with the engine firing order, admitting pilot air up to the air start check valves. The action of this valve allows a full charge of air from the main engine starting air header to enter the combustion space in the cylinder liner, forcing the pistons apart and rotating the crankshafts.

When the engine begins to fire, the hand control lever is moved from START to RUN position. This raises the control valve spring to close the valve, shutting off the air to the engine header. The engine starting air header and air start distributor are vented to the atmosphere through an opening in the control valve. When the pressure is relieved on the air chamber, the valve springs raise the pilot valves free of the cam, thereby venting all pilot tubing to the atmosphere.

The control valve located in the engine header is held closed by the combined action of open. When the piston opens, it releases a charge of air from the engine starting air header to the combustion space between the pistons in the cylinder liner, forcing the pistons apart and causing the crankshafts to rotate.

When the air start distributor shuts off the supply of pilot air to the valve, the spring closes the operating piston. The air from the engine starting air header presses against the balancing piston with greater force than it does against the check valve so that at no time is there enough pressure to open the valve.

The valve is vented to the atmosphere through two holes in the valve body. Water from the cylinder liner jacket enters the space between the check valve sleeve and the sleeve water jacket to cool the valve and valve body.

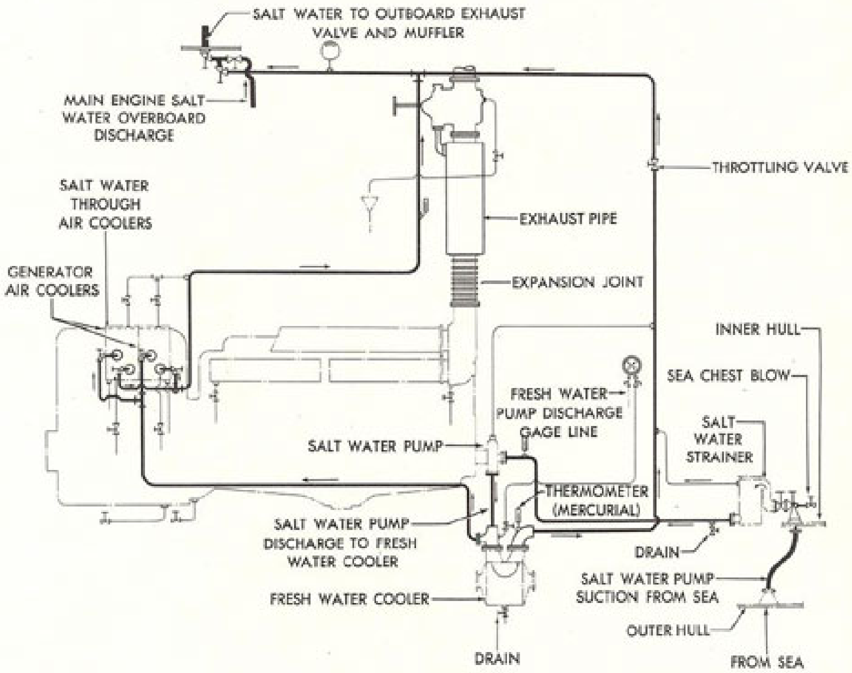
12B9. Exhaust system. The exhaust system conducts the gases from the engine combustion space through the exhaust ports to the

atmosphere. The hot gases are forced out of the cylinder liner when the upper and lower pistons uncover the exhaust and inlet ports The gases enter individual exhaust decks connected to the exhaust manifolds mounted in the cylinder block on each side of the engine. The gases pass on through the combined exhaust nozzle at the control end of the engine to the exhaust pipe and line. All of the cooling water passages of the engine can be drained through a plugged hole in the exhaust manifold. The exhaust decks and manifolds are doweled in place so that they will line up with other parts after removal or replacement. The exhaust manifolds are provided with openings at each cylinder for inspection and cleaning purposes. Plain flange covers are furnished for those openings for the side

on to the atmosphere, usually through one of the main engine exhaust valves. The exhaust deck castings and welded exhaust manifolds contain water jackets far the passage of cooling water from the welded exhaust nozzle, through which the water enters the engine. A drain plug is provided in the exhaust nozzle for draining any condensation in the nozzle, manifold, or exhaust opposite to the control side of the engine. The covers on the control side are quipped with pyrometers. These instruments indicate the temperature of the exhaust gases leaving the exhaust ports of each cylinder on a single selector indicator on the control board.

Figure 12-14. SALT WATER COOLING SYSTEM, GM 8-268 AND 8-268A.

Sub Diesel Home Page


SALT WATER TO OUTBOARD EXHAUST
VALVE AND MUFFLER
MAIN ENGINE SALT
WATER OVERBOARD
DISCHARGE
SALT WATER
THROUGH
AIR COOLERS
GENERATOR
AIR COOLERS
THROTTLING VALVE
EXHAUST PIPE
EXPANSION JOINT
INNER HULL
SEA CHEST BLOW
FRESH WATER
PUMP DISCHARGE
GAGE LINE
SALT WATER PUMP
SALT
WATER
STRAINER
THERMOMETER
(MERCURIAL)
SALT WATER PUMP
DISCHARGE TO FRESH
WATER COOLER
DRAIN
SALT WATER PUMP
SUCTION FROM SEA
FRESH WATER COOLER
OUTER HULL
DRAIN
FROM SEA

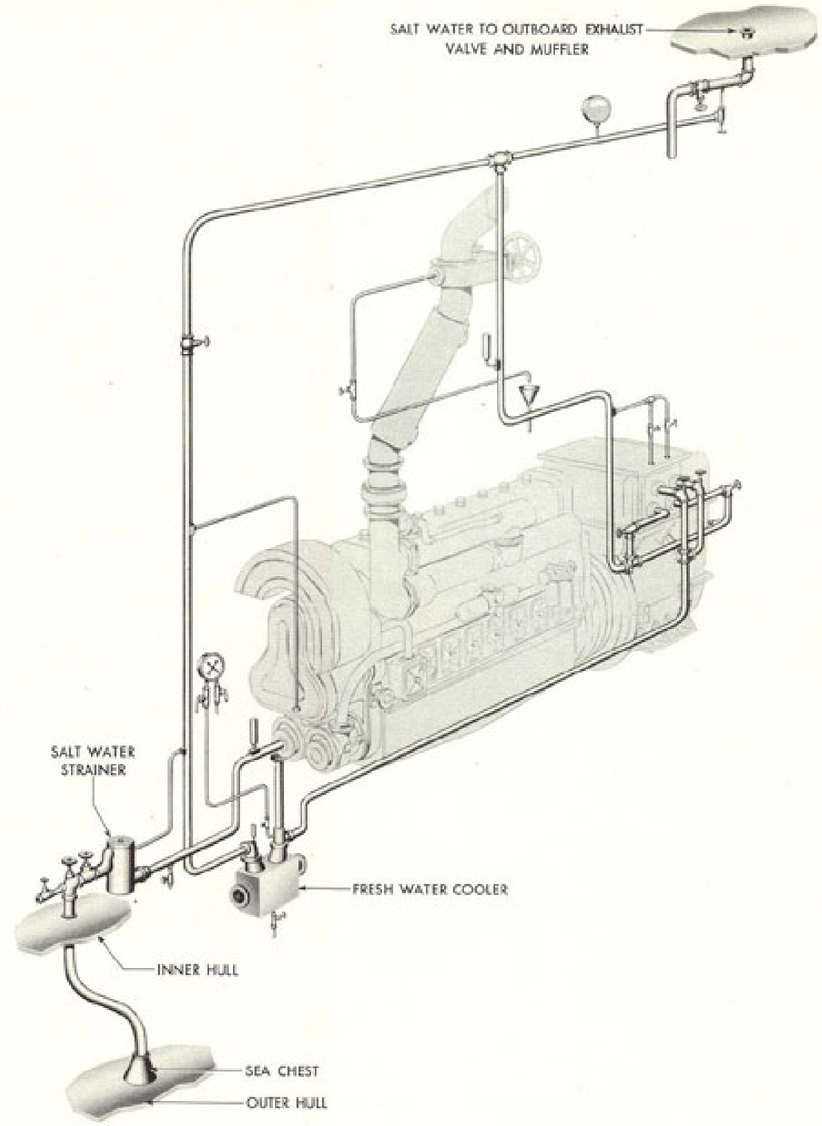
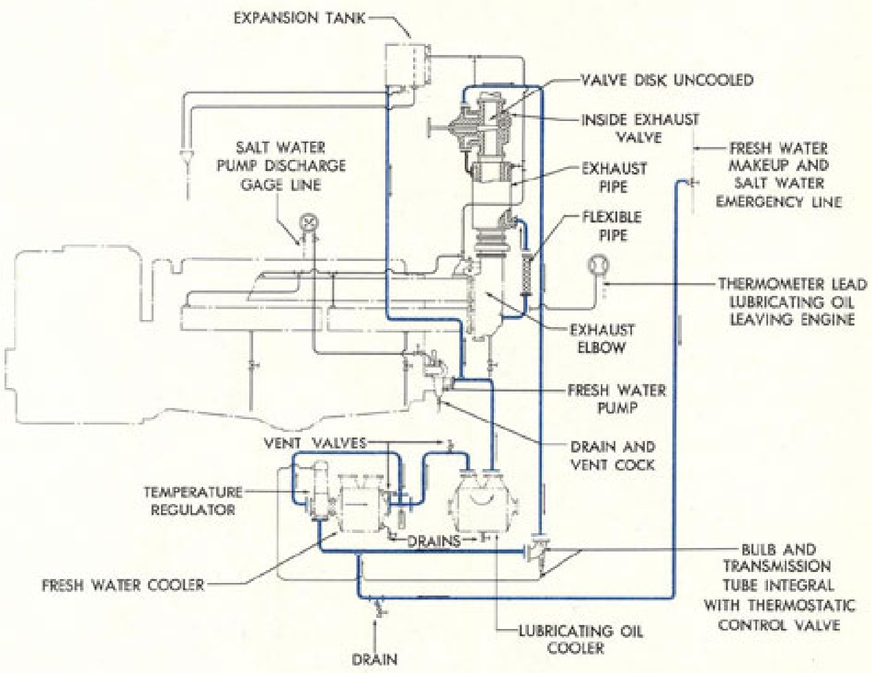
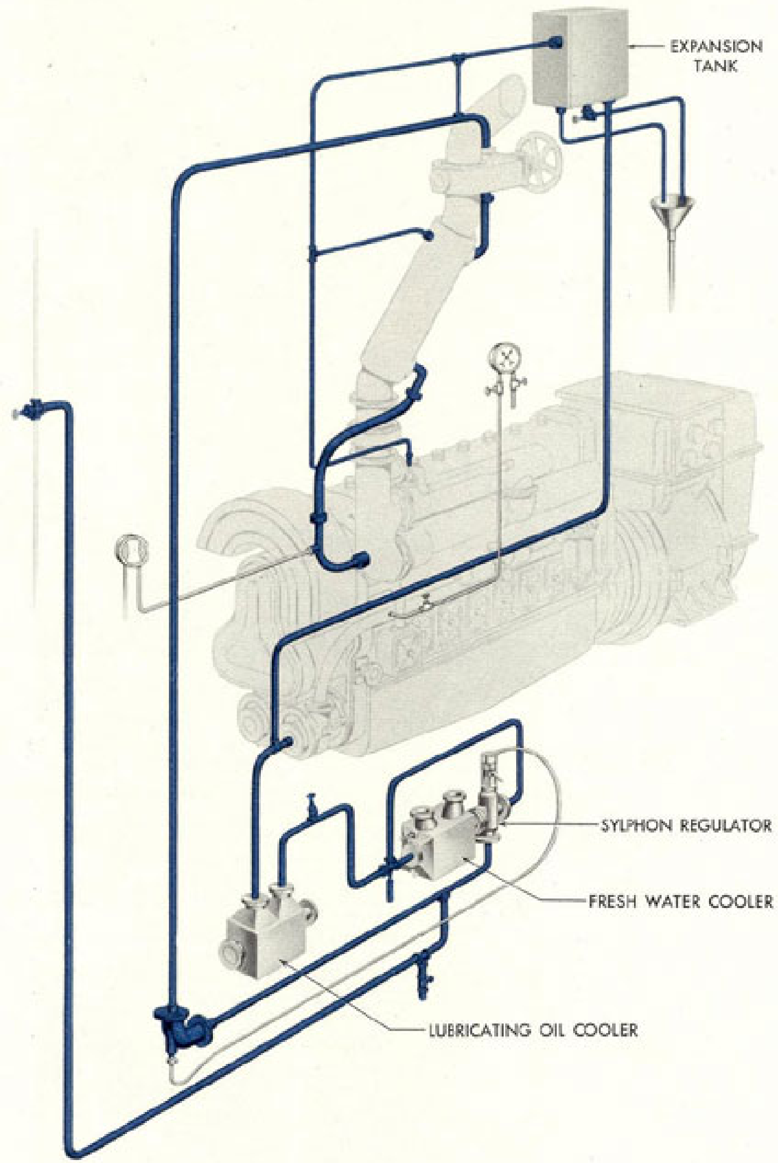

SALT WATER TO OUTBOARD EXHAUST
VALVE AND MUFFLER
SALT WATER
STRAINER
FRESH WATER COOLER
INNER HULL
SEA CHEST
OUTER HULL

Figure 12-15. FRESH WATER COOLING SYSTEM, GM 8-268 AND 8-268A.


EXPANSION TANK
VALVE DISK UNCOOLED
INSIDE EXHAUST VALVE
SALT WATER PUMP DISCHARGE GAGE LINE
EXHAUST PIPE
FRESH WATER MAKEUP AND SALT WATER EMERGENCY LINE
FLEXIBLE PIPE
THERMOMETER LEAD LUBRICATING OIL LEAVING ENGINE
EXHAUST ELBOW
FRESH WATER PUMP
DRAIN AND VENT COCK
VENT VALVES
TEMPERATURE REGULATOR
DRAINS
BULB AND TRANSMISSION TUBE INTEGRAL WITH THERMOSTATIC CONTROL VALVE
FRESH WATER COOLER
LUBRICATING OIL COOLER
DRAIN


EXPANSION TANK
SYLPHON REGULATOR
FRESH WATER COOLER
LUBRICATING OIL COOLER

13
REDUCTION GEARS

A. REDUCTION GEAR UNITS

13A1. Function and type. The main diesel engines are directly connected to the main generators which furnish power to the main motors or battery through the control cubicle. Two types of main drive installations are now in use in modern fleet type submarines. The older type which is at present used in about 95 percent of our submarines consists of four main motors arranged in pairs to drive each of the propeller shafts through a reduction gear. This type of installation uses a single control cubicle. The latest type of main drive installation consists of a split control cubicle and two large, slow-speed, double-armature motors which are directly connected to the propeller shaft. Each section of the split control cubicle is designed primarily to control propulsion on its particular side. It is possible, however, to tie the two sides of the split cubicle together and therefore use port engines on the starboard screw and vice versa.

This description of reduction gears is limited to the older type installation. Each reduction gear reduces the high main motor speed of approximately 1300 rpm to the propeller shaft speed of 280 rpm. The ratio of

The reduction gear assembly consists essentially of two main motor pinions forged and cut integral with the pinion shafts, one main gear or bull gear which is connected to the propeller shaft, and a lubricating oil pump gear which is geared to the inner pinion shaft. The forward ends of the pinion shafts are connected to their respective motors through flexible couplings. Each pinion shaft is supported by a cylindrical type bearing at each end.

The main gear is pressed and keyed to the gear shaft. The aft end of the shaft is coupled to the propeller shaft. On the forward end of the main gear shaft is mounted the collar of the main thrust bearing which absorbs the propeller thrust. The gear and shaft are carried on two sleeve bearings.

The sleeve bearings consist of steel shells lined with babbitt. The bearing shells are split

reduction is determined by the maximum efficiency obtainable from the propellers without loss of power at varying motor and propeller speeds.

The gears are single reduction, double helical type, a right- and left-hand helix being used to balance the fore and aft components of the tooth pressure. These helical gears produce a smoother action and avoid the tooth check of spur gears.

13A2. Description and operation. With the exception of minor differences in design, gear units produced by various manufacturers and installed on fleet type submarines today are similar. Specifications to which they are built will be found in the manufacturer's instruction book pertaining to the unit in question. The two units used on each ship are alike except that one is for port propulsion and the other for starboard propulsion. Facing aft, the port shaft rotates clockwise, and the starboard shaft rotates counterclockwise.

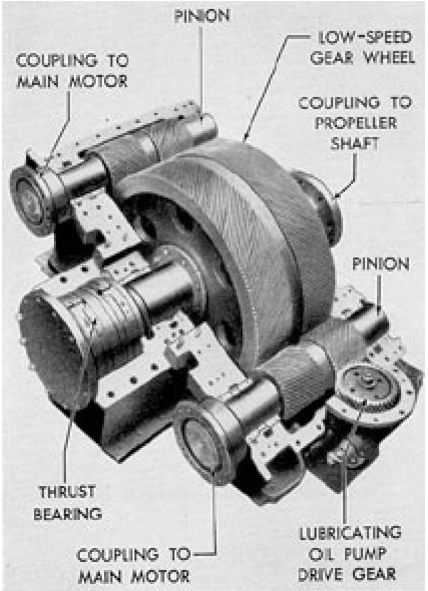


Figure 13-1. Reduction gear, top case removed.

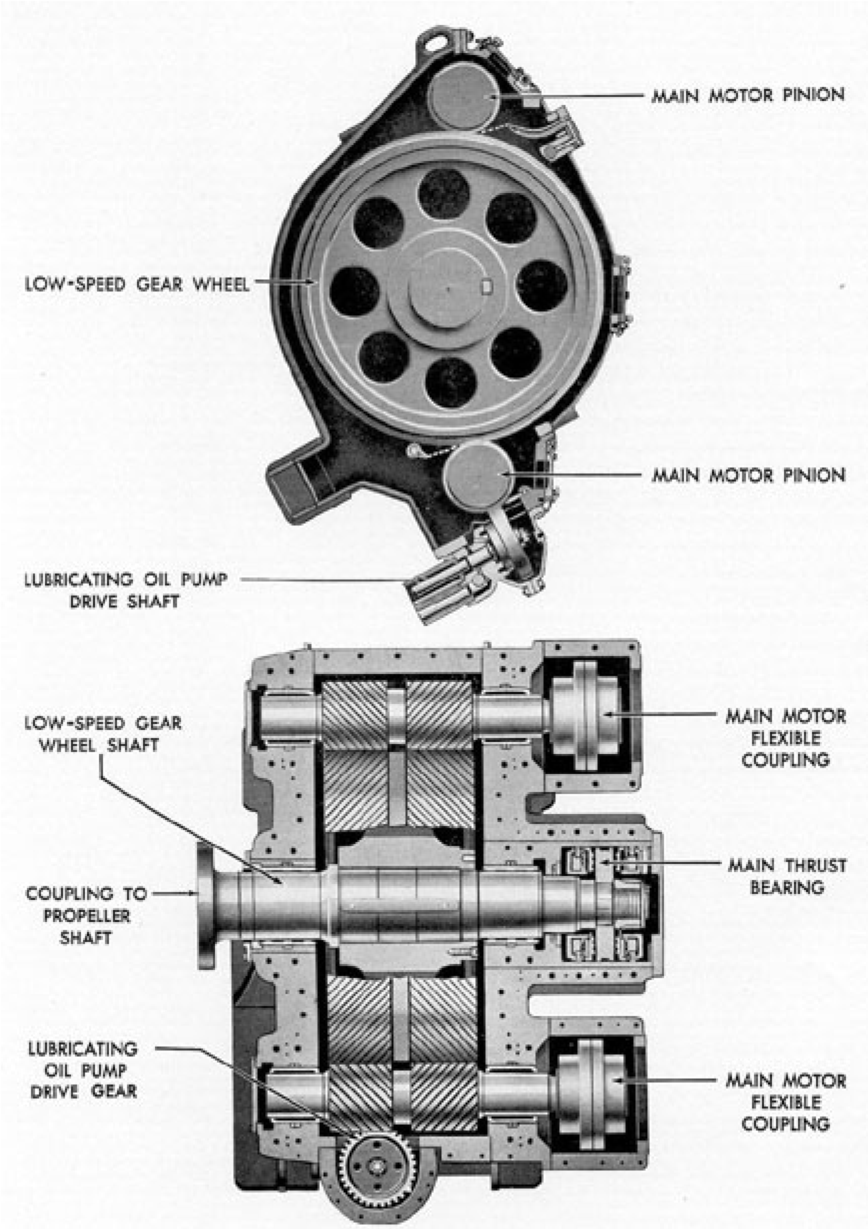


Figure 13-2. Sectional views of reduction gear.

and the two halves of each shell are held in alignment by dowels set in the lower half. Dowels in the bearing caps prevent the shells from rotating. The lubrication of the bearings is explained in Section 13A4f.

13A3. Flexible couplings between pinion and motor shafts. The couplings between the two main motor armature shafts and the pinion shafts of the reduction gear are of the enclosed flexible type. Each coupling consists essentially of

An accurate record of all repairs, adjustments, readings, and casualties should be kept in the machinery history.

b. **Unusual sounds.** A properly operating reduction gear has a certain definite sound which the trained operator can easily recognize. The cause of any unusual noises should be investigated, and the gears should be operated with caution until the source is located and remedied.

two hubs with external spur gears, and two sleeves with internal spur gears. The hubs are pressed on and keyed to their respective shafts. The floating sleeves fit around the hubs so that the spur gear teeth are permanently meshed. The floating sleeves are bolted together.

This type of coupling provides longitudinal flexibility between the driving and driven shafts and thereby permits the pinion to trail the main gear. Movement of the main gear is in turn limited by the clearance in the thrust bearing. The coupling permits a small amount of misalignment of the hubs to occur without causing operational difficulties. However, it is not advisable to operate continuously with the hubs out of alignment because the coupling is not intended to function as a universal joint. Continuous operation with the hubs out of alignment will result in excessive friction and gear teeth wear, and eventually will cause a breakdown.

The couplings are lubricated by a continuous stream of oil supplied by the main motor and reduction gear lubricating oil pump. Oil enters through a nozzle and after passing between the gear teeth is discharged through holes in the sleeve.

13A4. Maintenance. a. Machinery history. It is of great importance that the machinery history contain a complete record of the installation from the time of commissioning. Complete installation data as

c. Tooth contact. It is essential, for proper operation of the gears, that the total tooth pressure be uniformly distributed over the total area of the tooth faces. This is accomplished by accurate alignment, and adherence to the designed clearance limits. Alignment should be checked at the time the gear is installed, during each major overhaul, and after any casualty severe enough to threaten the alignment. Operating gears with faulty alignment are detrimental to the life and performance of the teeth. Continued quiet operation and good tooth contact are the best indications of proper tooth alignment.

d. Backlash. Backlash is measured by locking the main gear in its forward position and then moving each pinion just far enough forward and aft to make firm contact each way. The total lengthwise movement measured when doing this is the axial backlash. The backlash will increase with wear, and it can increase considerably without causing trouble. The actual longitudinal movement, as measured at the time the unit was built at the factory, should be found stamped on all pinion shafts except spares, and should be recorded in the machinery history. This measurement is the minimum allowable backlash.

e. Flexible couplings. The coupling backlash should be checked at regular intervals to see that it has not increased excessively. A dial indicator is used to measure the total backlash without dismantling the coupling. The one shaft is held stationary, and the dial indicator is

furnished by the contractor should be entered in the machinery index by prospective engineer officers at the contractor's yard. This should include the original bearing crown thickness or bridge gage readings, bearing clearances, thrust settings and clearances, and tooth clearances (backlash and root) of the gear wheel and pinion teeth. It is essential that these data be on hand when the alignment is subsequently checked.

mounted on the opposite or moving shaft with the indicator needle on some Dart of the coupling housing. By twisting the movable shaft back and forth without allowing the stationary shaft to move, the total backlash will be indicated on the dial indicator.

The backlash when found should be checked with the recorded initial backlash. If subsequent wear has increased the backlash to twice the original amount, replacement of the coupling should be considered.

Since the condition of the bearing surfaces depends upon the axial alignment of the shafts, regular inspection should include a check to see that proper alignment is maintained. To check the alignment, the flexible coupling must first be dismantled. To accomplish this, the manufacturer's instruction book should be consulted.

f. Bearings. All of the bearing caps may be removed for bearing inspection or replacement without disturbing the gear case. The pinions are light enough so that no trouble should be experienced when rolling out the lower halves of the pinion bearings once the shaft has been raised.

g. Bearing wear. The amount of wear of reduction gear bearings must not be allowed to become sufficiently great to cause incorrect gear tooth contact. The designed clearances, load diagrams, and methods of measuring bearing wear are given in the manufacturer's instruction book pertaining to the unit in question.

13A5. Special precautions. a. In case of churning or emulsification of the oil in the gear case, the gear must be slowed or stopped until the defect is remedied.

b. If for any reason, the supply of lubricating oil to the gears fails, the gears should be immediately stopped until the cause can be located and remedied.

c. When bearings are known to have been overheated, gears should not be operated, except in cases of extreme emergency, until bearings have been examined and the defects remedied.

d. If excessive flaking of metal from gear teeth occurs, the gears

When assembling, all bearing shells should be replaced in their original positions. Old cement should be cleaned off the mating surfaces of the bearing caps, end caps, and case, and a new coat of oilproof cement applied to these surfaces before reassembling. Do not permit the cement to contact the surface of the bearing. The dowel bolts should be tapped back into position before the bearing cap bolts are tightened.

Before starting the gear unit, sufficient oil should be pumped through the system by the standby pump to indicate pressures not less than 15 pounds on the two gages and to show steady flow through the thrust bearing sight flow indicator.

After starting the unit and securing the standby pump, the oil inlet temperature should not exceed 130 degrees F. Bearing temperatures should not exceed 180 degrees F, and the temperature rise should not exceed 50 degrees F. At full speed, lubricating oil pressure at the reduction gears should be at least 15 pounds. At any value above 25 percent of full speed, the pressure should not fall below 4 pounds. For continuous operation below 25 percent of full speed, the low limit pressure is 2 pounds.

Pressures and temperatures, as well as the flow through the thrust bearing flow indicator, should be observed at regular intervals during operation.

should not be adjusted, except in case of emergency, until the cause has been determined. Care should be taken, however, to prevent the entry of the metal flakes into the general lubricating system.

e. Unusual noises should be investigated at once, and the gears should be operated with caution until the cause is discovered and remedied.

f. No inspection plate, connection, fitting, or cover that permits access to the gear casing should be removed without specific authority of the engineer officer.

g. The immediate vicinity of an inspection plate joint should be kept free from paint.

h. When gear cases are open, precaution should be taken to prevent the entry of foreign matter. The openings should never be left unattended unless satisfactory temporary closures have been installed. Before replacing an inspection plate, connection, fitting, or cover, a careful inspection should be made by a responsible officer to insure that no foreign matter has entered or remains in the casing or oil lines.

i. Lifting devices should be inspected carefully before being used and should not be overloaded.

j. Naked lights should be kept away from vents while gears are in operation, as the oil vapor may be explosive.

B. MAIN MOTOR AND REDUCTION GEAR LUBRICATING SYSTEM

13B1. Description. Lubricating oil for the reduction gears and the main motors is contained in two sump tanks located beneath the reduction gears. Oil is supplied to each reduction gear unit and its bearings, as well as to the main motor bearings, by means of a pump attached to and driven by the reduction gears. The attached pump takes its suction directly from its sump tank and discharges oil directly into the reduction gear through a check valve, a strainer, a filter, and a cooler. The pump discharge line is also connected to the discharge side of the lubricating oil standby pump.

The standby pump is placed in operation in the event of failure of one of the attached pumps, and when the propeller shaft speed is below 34 rpm. The standby system is also used to prime the main motor and reduction gear bearings after a shutdown period.

The piping on the gear unit is arranged so that the oil flow divides, part of it going to the after bearings and inboard pinion spray box, and the remainder flowing to the forward bearings, outboard pinion mesh, and the flexible couplings.

All of the gear lubricating oil drains into the lower casing and is returned to the sump through a fitting connected to the bottom of the casing. A sounding rod may be inserted into the sump tanks for checking the oil level.

drops below the minimum pressure required. The alarm consists of a twin horn and warning light, both located in the maneuvering room.

13B2. Maintenance. Efficient lubrication of reduction gears is of the utmost importance. It is essential that oil at the designated working pressures and temperature be supplied to the gears at all times while they are in operation.

The proper grade of lubricating oil must be used. The oil must be so thin that the film will be squeezed from between the teeth, with resultant damage that may be beyond repair, nor so heavy that it will not flow through the restricted oil passages.

The lubricating system must be kept clean at all times. Particles of lint or dirt in the system are likely to clog the oil spray nozzles. The lubricating oil must be free from all impurities such as water, dirt, grit, and any particles of metal that may enter the system. Particular care must be taken to clean out metal flakes and fine chippings when new gears are wearing into a working fit. Magnets are fitted in lubricating oil strainers for this purpose.

The importance of taking immediate corrective measures when salt water is found in the reduction gear lubricating oil cannot be emphasized too strongly. The immediate location and sealing of the leak or removal of its source are not enough. Steps

A hand pump is provided for sampling the contents of the sump tanks. Before starting the machinery, samples should be taken from the tanks and examined for presence of water and dirt. When the hand pump brings up water, the pump should be operated until the water is removed. The engine should not be started until all of the water is removed. The hand pump is fitted with one suction line which takes a suction from either of the two sump tanks.

When filling the sump tanks from the filling line, the oil enters the sump tanks through the filling and transfer line. New oil may be transferred from the normal lubricating oil tank to the sump tank by means of the standby pump.

Low-pressure alarms are installed in the supply lines from the reduction gear to the main motors. The contact maker is set to close an alarm circuit when the lubricating oil pressure must also be taken to remove the contaminated oil from all steel parts. Several instances have occurred where, due to deferring this treatment, gears, journals, and couplings were so badly rusted and pitted that the gears had to be taken out by naval shipyard forces for reconditioning of teeth and journals. This condition can be reached in a week or less and may, result in burned-out bearings.

Frequent tests should be made to determine whether salt water is present in the oil, and the reduction gears should be inspected through the inspection plates for signs of salt water pitting. The oil level in the bottom of the gear case must not rise above the proper height predetermined for the particular installation. If the oil level is too high, the rotation of the gears will churn and aerate the oil, causing a sudden

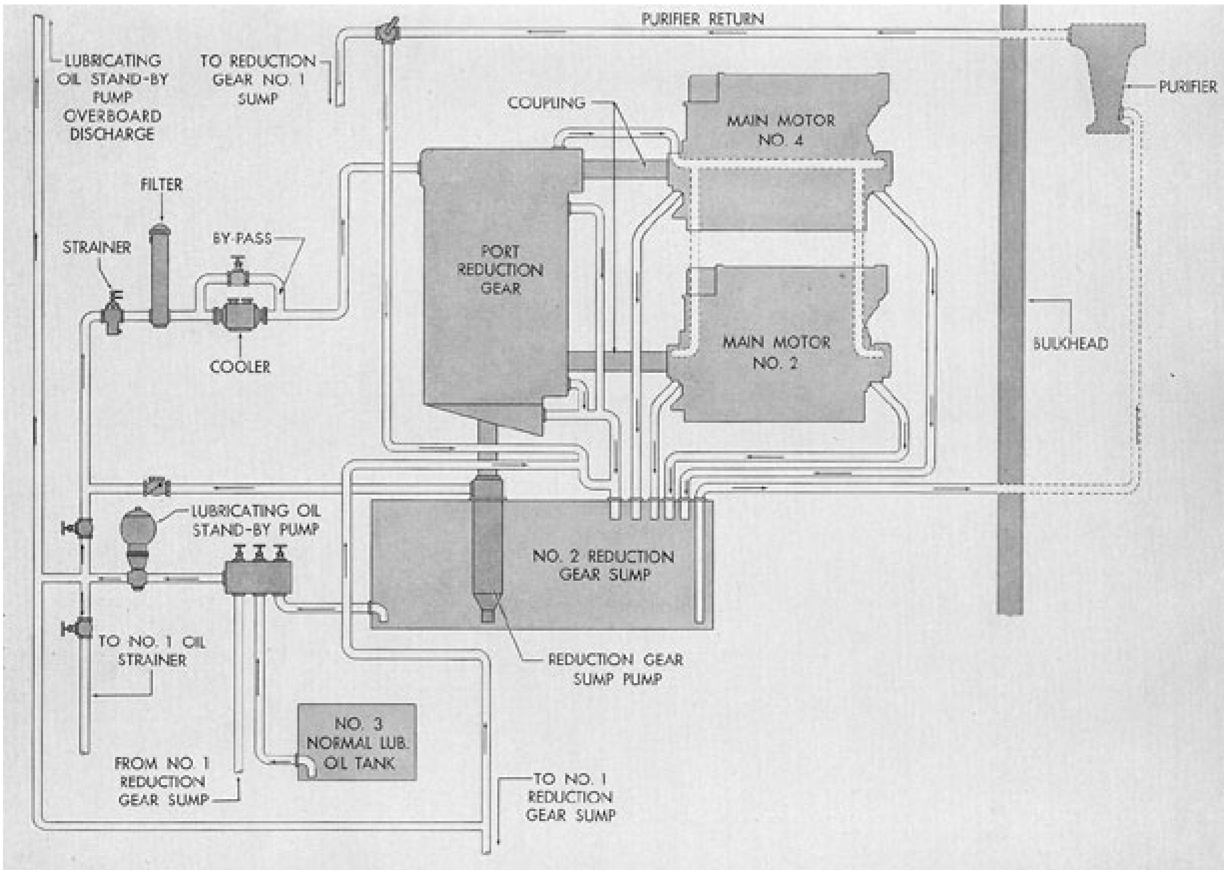


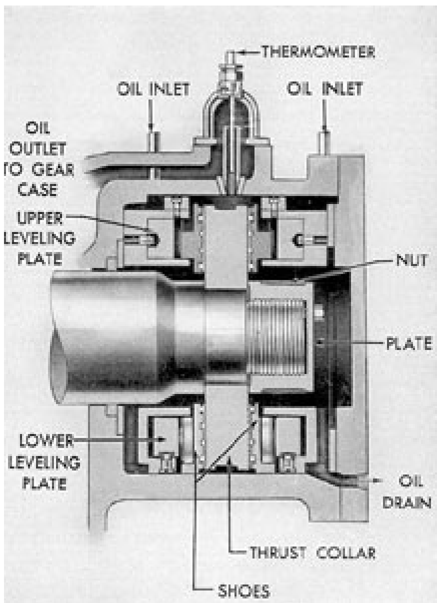
Figure 13-3. Schematic diagram of port main motor and reduction gear lubricating oil system.

increase in its temperature.

Spray nozzles to gears should be kept open at all times. No oil spray apparatus fitted for the lubrication of gears should be altered or rendered inoperative without authority from the Bureau of Ships.

C. PROPELLER SHAFT THRUST AND ADJUSTMENT

13C1. Description and operation. The thrust bearing on the forward end of the lowspeed gear shaft is manufactured by the Kingsbury Machine Works. This thrust bearing restricts axial movement of the propeller shaft in both the ahead and astern directions. The principal components of the bearing are a rotating thrust collar, which is keyed to the gear shaft, and stationary shoes with their load-equalizing supports or leveling plates. Hardened steel pivots or



rocking levers in the back of each shoe contact the leveling plates and allow slight titling to equalize the load.

The shoes are the bearing members in this type of bearing. They are supported in a manner that permits them to tilt and form a wedge shaped oil film between the shoe surface and the collar. The total end play permitted by the bearing is determined by the thickness of a spacer which rests against the end cover. This end play is fixed by the manufacturer at 0.015 to 0.030 inch.

The reduction gear oil pump supplies oil under pressure at a rate of approximately 3 gallons per minute. This quantity should be sufficient to limit the normal temperature rise between the oil inlet and outlet to about 15 degrees F. The oil pressure required is comparatively low, because the passages within the bearing are large. There are two oil inlets, one at each end of the bearing, and a single outlet as shown in Figure 13-4.

The line admitting oil to the bearing contains a needle valve that may be operated to obtain the desired flow. With the valve closed, sufficient oil will be delivered through a drilled hole in the valve seat for ordinary running conditions.

13C2. Maintenance. During normal operation, the thrust bearing will require no attention

Figure 13-4. Cross section of reduction gear thrust bearing.

except to see that the necessary circulation of clean, cool oil is maintained.

Since the bearing surfaces, when running, are completely separated by oil, there is practically no wear, and therefore, no take-up is provided except by shimming.

During the general overhaul period, the thrust bearings should be disassembled and thoroughly cleaned. Cleaning cloths that deposit lint should not be used. A coarse stone, a scraper, or a file should not be used on the collar surfaces.

D. PROPELLERS

13D1. General. Propellers used on modern submarines are of the four-blade solid construction type. There are two propellers on each ship, referred to as the starboard screw and the port screw. A knowledge of the design of the propeller is not important from the viewpoint of submarine operating personnel. It is enough to say that the designer has adequately designed the propeller to give optimum operating characteristics under all conditions of submarine operation, both surface and submerged. It is necessary, however, that submarine personnel have a knowledge of the terms used in describing a propeller so as to be able to discuss the subject of propeller operation more intelligibly. More important still, they should have some knowledge of the upkeep and maintenance of propellers, so as to keep them in the best possible operating condition.

13D2. Nomenclature. Terms used in describing a propeller and relative to propeller operation are as follows:

The pressure face is the after face of the propeller blade. It is customary to design the blade section by using this face for datum line. This is the driving side of the blade which pushes the water astern when the propeller is in action.

The suction face is the forward face of the blade. As this face is under a relatively low pressure, small irregularities in the surface

Disk area is the area of a circle whose diameter is equal to the propeller diameter.

PA/DA represents the ratio of the projected area to the disk area.

Developed area. The helicoidal (curved) surface of a propeller blade can be represented only approximately by a plane area. The developed area therefore approximates the sum of the actual areas of the pressure faces of all of the blades. Note: For convenience all areas are measured from the maximum hub diameter. This introduces a slight error due to the fact that the hub is not cylindrical.

Mean width ratio (MWR) is the ratio of the average width of the developed blade to the diameter of the propeller.

Pitch ratio is equal to the pitch divided by the diameter of the propeller.

Cavitation. When a propeller turns at high speed, the resulting high velocity between the propeller surface and the water, augmented by surface irregularities, tends to form a vacuum adjacent to the propeller. When the absolute pressure is reduced below the vapor pressure of the fluid, vapor pockets are formed, which break the continuity of flow and reduce the efficiency of the propeller. This phenomenon is called cavitation. When the cavitation bubbles collapse on the blade surfaces due to condensation, erosion of these surfaces results.

will cause cavitation. It is therefore important that this surface be maintained fair and smooth.

Diameter of a propeller is twice the distance from the shaft center to the extreme blade trip.

Pitch is defined as the distance the blade element would move in one revolution of operating in a solid medium. Unless otherwise defined, it is the designed pitch and equals the pressure face pitch of the blade section at the .7 radius. When the leading or trailing edge of the pressure face is not a true helix, the design pitch is considered to be the pitch of that part of the section which is a true helix.

Projected area is the area of the projection of the propeller blades upon a plane normal to the shaft axis.

True slip, or slip ratio, is equal to unity minus the ratio-of the speed of the water relative to the propeller in feet per minute divided by the pitch in feet times the rpm. When the speed of the ship through the water is used instead of the speed of the water relative to the propeller, the resulting slip ratio is known as apparent slip. The difference between these two velocities is due to the wake created by the ship's hull.

$S = I- V_a / (rpm \times pitch)$ where V_a = Velocity of water relative to the propeller in feet per minute.

Propeller numbering. Every propeller has been assigned a serial number for identification and to enable the Bureau of Ships to maintain a complete history. When referring to propellers,

the serial number, drawing number, and the nomenclature appearing on the drawing should be used.

13D3. Propeller inspection and maintenance. a. Inspection. Whenever the vessel is in drydock, the propeller should be inspected for possible damage. If there is reason to suspect that the propeller blades have been sprung or bent, and the fact is not obvious from a visual inspection, the pitch should be checked with a pitchometer. Whenever the propeller is removed, the tail shaft and hub bore should be inspected for corrosion and fractures.

All large, and sometimes small, propellers are brought to their final shape by chipping and grinding.

Propeller blade surfaces must be fair and free from humps and hollows. Many different blade thicknesses are used at numerous places on the blade and each must be accurate to less than 2 percent. Inaccuracies will set up forces that cause vibrations resulting in excess noise which is not acceptable on submarine installations.

Bent blades cause hydrodynamic irregularities which will cause vibrations and sometimes severe damage to struts and bearings.

Propellers are dynamically balanced to prevent vibrations. If an inspection and test show the need of removal of metal to obtain a balance, the metal must be taken from the pressure (after) face of the blades.

If an inspection shows small pieces broken off the blades or slight cracks, repair may be possible by the hot melt process. If the breaks or cracks appear to be so serious that either blade or hub strength is affected, expert consultants should be called in to survey the damage before any attempt is made to repair. If there is any question as to the suitability of the damaged propeller, it should be replaced. Pitting or erosion found during inspection should be considered from the viewpoint of cause and the elimination of the cause if possible. Fast runs or steady runs under high power in prolonged heavy weather will sometimes erode the backs of the blades. This erosion is due to cavitation and usually appears at the tips of the blades. Erosion under these conditions cannot be prevented, but pitting or erosion at any other point on the blades is usually the result of a fault that can be eliminated.

b. Propeller blade maintenance. The casting and machining of the propellers require extreme care to maintain the relationship between the engineering calculations of proper pitch, diameter, and area and the actual physical dimensions of the propeller.

Navy propellers are invariably made of cast solid manganese

Blade fillets located near the hub should be fair and should change uniformly, decreasing near the ends. There should never be any knuckles or sharp corners for the water flow to break over. Irregular contours always result in erosion.

Blade edges and tips must be maintained as sharp and clean as called for on the propeller drawing. A propeller can vibrate in many different ways, and each vibration is associated with a definite frequency. Forces that cause vibrations of a definite frequency are sometimes the result of blunt edges on the blades near the tip, and these vibrations result in a noisy or singing propeller. Propellers are dynamically balanced to prevent vibrations when in service.

Hub taper. The hub of the propeller is accurately bored out to receive the propeller shaft. One or more keys are used to insure a tight fit and to prevent movement between the shaft and the hub. These keys must fit uniformly and snugly in both shaft and hub, and no movement should be permitted. Loose keys work back and forth and may eventually result in the loss of a propeller.

Fairwater cap. The after end of the propeller tail shaft is sealed against water by a fairwater cap which is filled with hot tallow. Some ships have a separate nut behind the fairwater cap for holding the propeller hub on the tail shaft, and some fairwater caps have the nut integral with the cap. In either case, the cap and nut must be fitted so that there is no play. The nut is kept from working loose by a locking key. The faces

bronze. Usually small propellers and frequently large propellers (up to destroyer size) are machined to their true pitch. The tolerance allowed is from 1/2 to 2 percent, the amount depending on the application.

of the cap must be smooth and free of sharp corners or irregularities.

13D4. Propeller upkeep. It has been found that clean and properly operating propellers add measurably to the amount of speed obtainable from a given propulsion installation. It has therefore become the practice to clean the blades of both propellers and perform minor repair jobs on the blade tips at every refit and overhaul period of a submarine. It is much easier to do this when the vessel is in drydock, but the cleaning especially and some repair jobs may be accomplished by a diver when the ship is waterborne. The cleaning is usually done by an air operated cleaning tool with little or no difficulty. When the blades are cleaned, the diver should, in addition, make a careful inspection of the tips of the blades to check for irregularities, nicks, and bent sections. These should be corrected if the operations schedule permits.

13D5. Routine tests and reports. a. Whenever a ship is-docked the engineer officer (of the ship) should examine the propellers, and the result of the examination entered in the engineering log and in the ship's log.

b. As soon as practicable after docking a vessel, the naval shipyard or repair force should make a careful examination of the propellers, and any repairs found necessary should be undertaken immediately so that the undocking of the vessel will not be delayed.

c. The stenciled hub and blade data should be verified and recorded at each drydocking. Wear of bearings, adjustments made, general conditions found, and work performed should be recorded in the machinery history.

d. At each interim and regular naval shipyard overhaul docking, the hub cap should be removed from each propeller and the propeller nut examined.

e. When repairs are made to propellers, the activity performing the work should make a detailed report, including repairs effected, condition of propeller, location and extent of defects, data stamped on hubs and blades, and pitch measurements, if taken. Applicable forms should be submitted if major repairs affecting pitch and blade dimensions have been accomplished or if the propeller was balanced dynamically.

Previous Chapter Sub Diesel Home Page

Version 1.10, 22 Oct 04